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TECHNICAL NOTE

A hybrid solar desiccant cooling system

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1. INTRODUCTION

Developed countries use a significant amount of energy in cooling buildings. In most cases this cooling is done by refrigeration machines using a form of high-grade energy (e.g. electricity). A significant portion of the electricity consumed in the commercial sectors of Australia and the United States is used in providing air conditioning[1, 2].

Several studies[3, 4] have investigated alternative methods of providing cooling using lower-grade energy (normally thermal energy at temperatures that can be met by solar collectors). The system proposed by Dunkle[3] uses a desiccant to reduce the moisture content of air and then evaporatively cools it. Systems of this type, which use a desiccant dehumidifier to convert latent cooling load into sensible load and then meet this load using evaporative coolers, are known generically as open-cycle desiccant cooling systems.

Jurinak and Beckman[5] have investigated the applicability of a number of such systems in meeting residential cooling loads by dynamic simulation using the computer program TRNSYS[6]. This was done for a number of climate types with the desiccant regenerated by a solar air-collector/rock-bed storage system. The results showed that to compete with cooling by conventional vapour compression refrigeration, the systems need components which are at least as effective as any yet built. Their advantage, however, is their use of low-grade heat at temperatures between 60° and 100°C, which can be provided by a conventional solar energy system or from waste heat. Due to the greater amount of energy consumed for cooling in the commercial sector, and the variation in the type of loads to be met in commercial buildings, Close and Sheridan[7] investigated how well these systems performed in such buildings and found results similar to those of Jurinak and Beckman.

A significant advantage of these cycles, which was evident in both of these studies, is their ability to operate in an indirect evaporative cooling mode at times of low latent load and low relative humidity. In hot, dry climates such as Phoenix, Arizona and Port Hedland in northwest Australia, thermal coefficients of performance (COP = cooling energy provided/energy input) calculated on a monthly basis can reach 19 (July in Port Hedland). Such high values are obtained because indirect evaporative cooling has an infinite thermal COP. This figure would be reduced when parasitic power is accounted for, but still gives an indication of the potential of systems which can easily switch into the indirect evaporative cooling mode. Since the main restriction on the performance of these systems is the inability of the evaporative cooler to meet the cooling load without overburdening the desiccant dehumidifier, there appears to be merit in considering cycles of similar form which provide cooling without adding moisture.

This paper considers a cycle which combines indirect evaporative cooling with a dehumidifier to meet latent loads and a vapour compression refrigeration unit to meet sensible load. The cycle has been studied for two climate types and two different loads representative of those likely to be met in commercial buildings. Also considered is the combination of this cycle with a solar energy system which regenerates the desiccant dehumidifier.

2. SYSTEM DESCRIPTION

The system considered is shown in Fig. 1(a), and its representation on a psychrometric chart is shown in Fig. 1(b). The working fluid of the system can be traced through the diagrams as follows. Fresh air at ambient state 1 is mixed with the return air from the space at state 2 to give state 3. This air is dehumidified and heated to state 4 using a desiccant dehumidifier and is subsequently cooled to state point 5 by an indirect evaporative cooler, which in the case studied was a plate heat exchanger (PHE) device as described by Pescod[8]. Further cooling to the required entry state 6 is performed by the evaporator coil of the vapour compression (VC) unit. The desiccant dehumidifier has been assumed to be of the rotary wheel type and is regenerated by heat from the condenser of the vapour compression unit, plus an auxiliary heater. In the solar cycle the regeneration is done by the solar energy components, and ambient air is used to remove the condenser heat. This means that the condenser operates at a lower temperature than in the case where it is used to regenerate the desiccant, and as a result the vapour compression unit operates more efficiently (i.e. COP_{VC} rises).

When the cooling load is predominately sensible, indirect evaporative cooling, using the PHE, is used. If the PHE is unable to meet the full load, it is augmented by the vapour compression unit.

3. SYSTEM MODEL

The system was simulated using program TRNSYS. Several component models were combined to form a new TRNSYS module. These component models are described below.

The PHE was modelled using the method of analysis described by Pescod. This assumes a constant effectiveness which is defined as

$$\eta_{PHE} = \frac{T_4 - T_5}{T_4 - T_{[illegible]}}$$

The vapour compression unit was modelled by a fit to

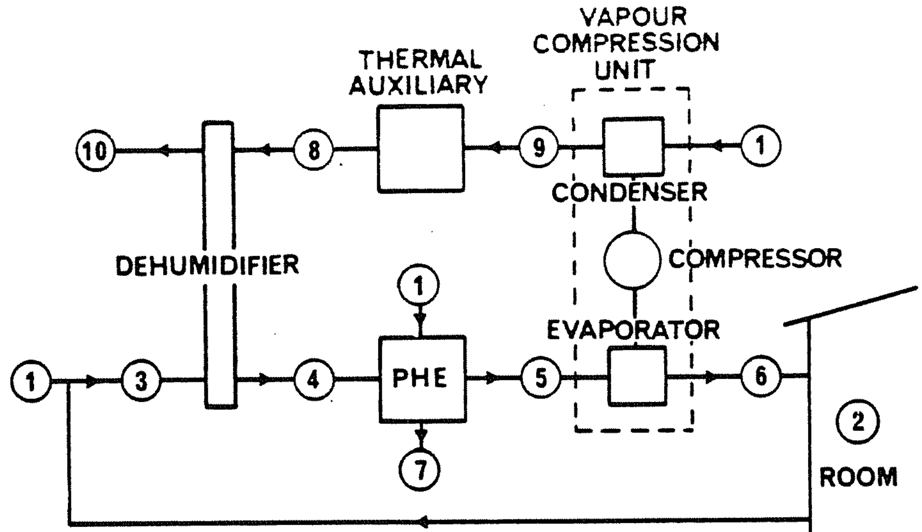

Fig. 1(a). Hybrid cycle: schematic.

the coefficient of performance (COP_{VC}) in terms of the independent variable ($T_{COND} - T_{EVAP}$). This fit was to data supplied by a manufacturer[9] and is considered typical for such units.

The desiccant dehumidifier was modelled using the theory developed by Banks. Close and Maclaine-Cross[10. 11]. This method develops an analogy between heat transfer in a rotary sensible heat exchanger and heat and mass transfer in a dehumidifier. Two potentials (analogous to temperature in heat transfer only systems) F_1 and F_2. which are complex functions of temperature and humidity ratio. determine the process path of the dehumidifier on a psychrometric chart. The performance of the dehumidifier is determined by two effectivenesses (η_{F_1} and η_{F_2}) which, given its inlet states. allow the calculation of the outlet states of the dehumidifier.

In solving the set of equations to determine the state points. the computer model first solves for point 6. given the room state and using the slope of the cooling load line and a criterion which states that the maximum enthalpy change in the conditioned space should be 15 kJ kg^{-1} (this corresponds to the accepted figure for mass flow per unit cooling capacity of 400 cfm ton^{-1} of refrigeration). If this criterion cannot be met because the load line crosses the saturation line before reaching the required enthalpy difference. the point of intersection between the load and saturation lines is chosen. This means that the air entering the space is capable of meeting the sensible and latent loads. but is saturated. and the fan speed must be increased to give a flow greater than that considered desirable for refrigerated air conditioning systems.

Once point 6 is known. the humidity ratio that the dehumidifier must reach is determined and. given the effectiveness η_{F_1} and η_{F_2}. points 4 and 8 (the required inlet condition of the regenerating stream) can be found. Point 5 may be found from the PHE equation. and so the cooling that the vapour compression unit must provide is known. as are its condenser and evaporator temperatures. From

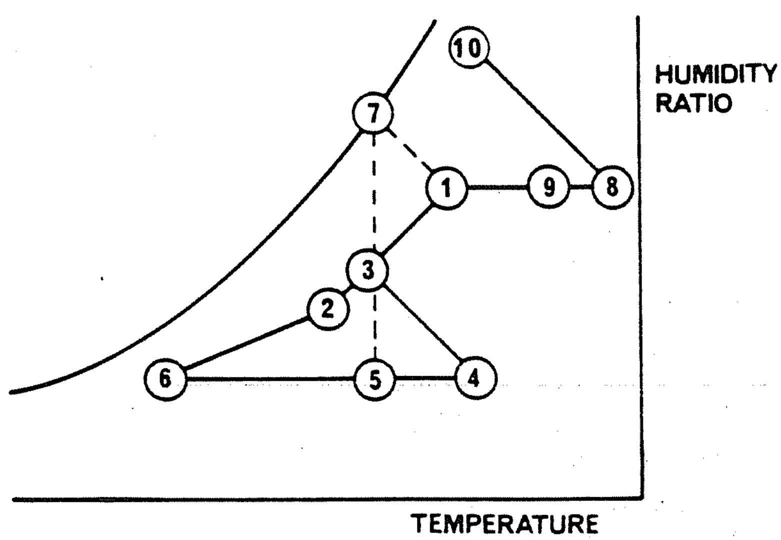

Fig. 1(b). Hybrid cycle: psychrometrics.

the calculated value of COP_{VC} and the energy accepted by the evaporator, the energy rejected in the condenser can be found and, as a result, state point 9 is known. If the temperature of point 9 is greater than that of point 8, the condenser can fully regenerate the desiccant wheel, and no auxiliary thermal energy is needed. Otherwise, the auxiliary energy is calculated from the temperature difference between points 8 and 9.

If possible to meet the cooling load using the PHE only, this is done, and the calculation is of the inlet to the conditioned space using the PHE equation. The hierarchy of control modes is given in Table 1.

When solar energy is used to regenerate the dehumidifier, a standard model of a combination solar air-collector/rock-bed storage system is used to find state point 9.

4. LOAD MODEL

The building studied was a three-story office block which included some energy conservation measures (notably no infiltration was considered). It was modelled using standard TRNSYS subprograms which are based on the ASHRAE "response factor method" [12]. This method allows for the dynamic lag in the response of walls, roofs, etc., to heat inputs. Thus, the cooling load exhibits the time dependence likely to be experienced by a real building subjected to time-varying heat inputs. Outside air for ventilation purposes was introduced through the cycle, and lighting, equipment and occupant loads were estimated from ASHRAE figures and included as time-dependent inputs. Two internal moisture gains were modelled: one was typical for an office building: the other resulted in a high latent load as might be found in a building with a number of moisture-generating appliances.

5. SYSTEM/LOAD COMBINATIONS STUDIED

The systems described were studied for two locations in Australia: these were Darwin, which has a hot, humid climate, and Port Hedland, which has a hot, dry climate. Both locations were considered for a summer month (January) and a winter month (July). Two sets of effectiveness were studied, and these are shown in Table 2. (Note that referring to high effectivenesses in the dehumidifier means high η_{F2} and low η_{F1}.)

The solar collector parameters used were $F_R(\tau\alpha) = 0.49$ and $F_R U_L = 3.49$ W m^{-2}K^{-1}, which are consistent with a double-glazed flat-plate solar collector.

In the high sensible fraction load, the sensible load was normally over 90% of the total load, while in the high latent load the sensible fraction of the load was typically 30–50% at times of high load. In both cases the ventilation flow rate was 7.1 L s^{-1} per person (15 cfm per person).

6. RESULTS AND DISCUSSION

The main aim of the study was to investigate changes in energy consumption resulting from the use of the hybrid system instead of a normal vapour compression unit. The study also compared the energy consumption of these two systems with a third option in which a combination of a PHE and vapour compression unit is used. Due to the difference in the quality of energy forms being used to run the systems, the comparisons are made on the basis of energy used and primary energy used. In doing this it is assumed that electricity has a thermal energy value of 12 MJ kWh^{-1}.

Tables 3 and 4 summarize the results obtained for the hybrid cycle with high-effectiveness components, and Table 5 shows the results for low-effectiveness components in Darwin.

From these tables it is evident that the hybrid cycle can use less energy than a normal vapour compression unit. Considering the case of a high-effectiveness hybrid cycle meeting the high sensible load in Darwin, we can see that the average energy saving when summer and winter conditions are considered is 24%. For the same type of load in Port Hedland, the energy saving is 40%. The savings in resource energy are at least equal to, and, typically, higher than, this (27% and 40%, respectively, for the cases quoted). (It should be noted that in all the calculations the extra parasitic power requirements of the hybrid cycle have not been considered.) The energy savings are a result of the potential for indirect evaporative cooling which has been used in the cycle to meet sensible load through the PHE, or latent load when the PHE is used in conjunction with the desiccant dehumidifier. In most of the cases studied the PHE and desiccant dehumidifier can meet the latent load by using only the energy rejected in the condenser of the vapour compression unit for regeneration of the dehumidifier. When the cooling required is mainly sensible, the PHE can meet a substantial part of it for both climates considered. In particular, in situations such as those in Port Hedland in winter, the PHE can give large energy savings. Over 80% of the cooling needed in this case is provided by the PHE, and the cycle operates in the indirect evaporative cooling mode for 86% of the time it is running.

Table 1. Control modes

Mode	Condition	Action Taken
0	No load	No cooling needed.
1	Warm and Dry	Indirect evaporative cooling using the PHE.
2	Hot and Damp	Indirect evaporative cooling using the PHE, plus auxiliary cooling by the vapour compression unit.
3	Hot and Wet	Full hybrid cycle operation using dehumidifier.
4	Very Hot and Dry	Sensible load which is met by the vapour compression unit with the aid of the PHE.

Table 2. Effectivenesses of components in systems studied

Parameter	Effectiveness	
	High	Low
η_{PHE}	0.85	0.70
η_{F1}	0.05	0.07
η_{F2}	0.95	0.80

The advantage given by the PHE can also be used by combining it with a vapour compression unit. Such a system will give energy savings of 20% and 35% (for Darwin and Port Hedland respectively) for the cases previously quoted of high sensible load and high effectiveness.

If solar energy is used instead of condenser energy to regenerate the desiccant dehumidifier, further energy savings are possible. The savings will depend on the size of the solar energy system associated with the hybrid cycle, but, for the "typical" size chosen here, the savings of a solar/hybrid cycle in the two situations quoted above would be 30% for Darwin and 42% for Port Hedland. Since the solar energy replaces thermal energy needed in the dehumidifier, the solar/hybrid cycle is more promising in higher latent load and higher ambient humidity cases. This is evident from Tables 3 to 5, which show that the advantage of using solar energy in Darwin is greater than using it in Port Hedland.

The effect of climate on the performance of the hybrid cycle can be gauged by comparing the results from Darwin and Port Hedland for different times of the year. Port Hedland has a hotter and drier climate than Darwin, and as a result the hybrid cycle provides greater energy savings there. The cycle spends more time in the indirect evaporative cooling mode, and the lower ambient humidity ratios mean that the desiccant dehumidifier can be regenerated more easily.

Using less effective components also has a significant effect on the hybrid system performance. Considering the case of the cycle meeting a largely sensible load in Darwin,

Table 3. Energy use in the hybrid cycle in Darwin

Month	Load Energy and Type (GJ)	Cycle	Energy Used (GJ)	Primary Energy Used (GJ)	System COP	System Primary Energy COP
January	167.4	Hybrid	59.6	190.7-	2.81	0.88
	High Sensible	Hybrid + Solar	53.5	170.1	3.13	0.98
		VC + PHE	60.6	202.4	2.76	0.83
		VC	62.0	207.1	2.70	0.81
	267.1	Hybrid	213.9	371.7	1.25	0.72
	High Latent	Hybrid + Solar	114.7	231.2	2.33	1.16
		VC + PHE	94.0	314.2	2.84	0.85
		VC	95.6	319.4	2.79	0.84
July	131.5	Hybrid	25.7	81.0	5.12	1.62
	High Sensible	Hybrid + Solar	23.0	76.5	5.72	1.72
		VC + PHE	27.0	90.2	4.87	1.46
		VC	47.9	160.0	2.75	0.82
	231.2	Hybrid	116.1	171.0	1.99	1.35
	High Latent	Hybrid + Solar	39.0	85.0	5.93	2.69
		VC + PHE	51.6	172.4	4.40	1.34
		VC	80.9	270.3	2.86	0.86

Table 4 Energy use by the hybrid cycle in Port Hedland

Month	Load Energy and Type (GJ)	Cycle	Energy Used (GJ)	Primary Energy Used (GJ)	System COP	System Primary Energy COP
January	188.8 High Sensible	Hybrid	62.0	207.1	3.05	0.91
		Hybrid + Solar	59.9	200.4	3.15	0.94
		VC + PHE	67.0	223.7	2.82	0.84
		VC	68.4	228.5	2.76	0.83
	288.5 High Latent	Hybrid	115.3	285.3	2.50	1.01
		Hybrid + Solar	67.8	210.5	4.26	1.37
		VC + PHE	102.6	355.2	2.81	0.81
		VC	112.2	374.9	2.57	0.77
July	89.6 High Sensible	Hybrid	1.4	4.4	64.00	20.36
		Hybrid + Solar	1.3	4.2	68.92	21.33
		VC + PHE	1.5	5.0	50.73	17.92
		VC	30.9	103.1	2.90	0.87
	189.2 High Latent	Hybrid	29.8	32.1	6.35	5.89
		Hybrid + Solar	0.9	3.0	210.22	63.07
		VC + PHE	7.0	23.5	27.03	8.05
		VC	62.6	209.0	3.02	0.91

Table 5. Energy use by the hybrid cycle with low effectiveness components in Darwin

Month	Load Energy and Type (GJ)	Cycle	Energy Used (GJ)	Primary Energy Used (GJ)	System COP	System Primary Energy COP
January	167.4 High Sensible	Hybrid	68.7	211.5	2.44	0.79
		Hybrid + Solar	58.8	180.9	2.85	0.93
		VC + PHE	61.3	204.8	2.73	0.82
	267.1 High Latent	Hybrid	274.3	463.3	0.97	0.58
		Hybrid + Solar	156.0	279.8	1.71	0.95
		VC + PHE	94.8	316.8	2.82	0.84
July	131.5 High Sensible	Hybrid	30.9	95.8	4.26	1.37
		Hybrid + Solar	27.0	82.3	4.86	1.60
		VC + PHE	30.8	120.8	4.27	1.09
	231.2 High Latent	Hybrid	116.1	171.0	1.99	1.35
		Hybrid + Solar	55.6	114.3	4.16	2.02
		VC + PHE	56.7	189.2	4.08	1.22

Table 6. Energy use in the hybrid cycle for a high sensible load in Darwin in July

Effectivenesses			Energy Used	Primary Energy Used
η_{PHE}	η_{F1}	η_{F2}	(GJ)	(GJ)
0.85	0.05	0.95	25.7	81.0
0.85	0.07	0.80	27.5	83.5
0.70	0.05	0.95	29.2	93.5
0.70	0.07	0.80	30.9	95.8

Table 7. Effect of load ratio on hybrid system performance (for high effectiveness components in Darwin in July)

Sensible/Total Load	COP	COP Based on Primary Energy
0.90	5.11	1.62
0.78	4.47	1.64
0.51	3.22	1.57
0.51	1.99	1.35

the energy savings over a vapour compression system drop from 25% to 12% when the effectivenesses drop from the high to low values used. The effect does not appear as significant, however, as in desiccant open-cycle cooling systems where the same reduction in effectivenesses results in COPs being more than halved[5]. The effect of varying the effectiveness of the PHE appears to be more significant than that of varying the dehumidifier effectivenesses, as is shown in Table 6. To a large extent this is due to the time spent in the indirect evaporative cooling mode, which means that the PHE is in use more often than the dehumidifier.

Using a lower effectiveness for the PHE also has an effect on the combination vapour compression/PHE cycle. For the case quoted above, the energy savings over a vapour compression system drop from 20% to 16% when the effectiveness is lowered. The high effectiveness used for the PHE is, however, technically feasible (this may not be so for the high dehumidifier effectiveness considered).

The other variable which has an effect on the hybrid system performance is the ratio of sensible load to total load. From the results presented in Tables 3 and 4, it can be seen that the higher latent load results in lower COPs. This is further evidenced by Table 7, which shows the hybrid systems performance for two intermediate load ratios.

The hybrid system copes better with higher sensible load to total load ratios. With high latent load extra thermal energy is needed to regenerate the dehumidifier, because more moisture must be removed from the air. For the two intermediate load ratios given, the ratio of extra cooling load met to extra thermal energy supplied is 1.36. The energy added is low-grade thermal energy, and so the result is better when primary energy is considered. This can be seen from the last column in Table 7.

As has been stated previously, none of the calculations to date have included parasitic power. This will be used mainly in the fans circulating air through the cooling system and space. A survey done by Kowalczewski[13] found that fan energy was approximately 10% of the cooling load, which means it is 23% of the energy input to a system with a COP of 3. Thus changes in parasitic power can have a significant effect on the overall COP of an air conditioning system. Due to the lack of data on the pressure drops in desiccant dehumidifiers, no attempt has been made to estimate the increase in parasitic power due to extra components in the hybrid system. It is also likely that the hybrid system will operate with a higher mass flow rate than vapour compression systems which normally cool to lower temperatures and hence have lower flow rates. Increases in parasitic energy could have a significant affect on the viability of hybrid systems in energy terms, and, since the extra energy needed is high-grade energy, it could also reduce their significant advantage in saving primary energy.

7. CONCLUSIONS

An air conditioning cycle which incorporates a desiccant dehumidifier to help meet the latent load has been investigated for a range of conditions and loads. The results indicate that, for the two climate types considered and the high sensible load, the hybrid cycle uses 25–40% less energy over January and July than a conventional vapour compression unit.

Other major conclusions of the study are listed below.

1. Combining an indirect evaporative cooler with a vapour compression unit can give significant energy savings. This is mainly because of the large percentage of the time spent in the indirect evaporative cooling mode in hot, dry climates.
2. The hybrid cycle saves more energy in hot, dry climates than it does in hot, humid climates where it may, in fact, use more energy than a vapour compression unit.
3. Although component effectivenesses affect the hybrid cycle performance, the effect is not as drastic as in other desiccant cooling systems.
4. The cycle saves energy compared to a vapour compression unit when the load to be met has a high sensible fraction, but may use more energy if the load has a high latent fraction.
5. Solar energy can be combined with the hybrid cycle to further save energy, especially in meeting loads with higher latent fractions.

NOMENCLATURE

COP coefficient of performance of the system
COP_{VC} coefficient of performance of the vapour compression unit
$F_R(\tau\alpha)$ solar collector gain characteristic
$F_R U_L$ solar collector loss characteristic in W m^{-2} K^{-1}
T_{COND} temperature of outlet air from condenser of vapour compression unit in K
T_{EVAP} temperature of outlet air from evaporator of vapour compression unit in K
T_N temperature of state point n on Fig. 1(b) in K
T_n^* wet bulb temperature of state point n in K
F_1, F_2 dehumidifier "potentials" (functions of temperature and humidity ratio)
η_{F1}, η_{F2} dehumidifier effectivenesses
η_{PHE} PHE effectiveness

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