INDIRECT EVAPORATIVE COOLING

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About the Author

On graduating in Mechanical Engineering from Monash University in 1968, Bill Ellul worked with CSIRO for 14 years, in air conditioning and solar energy research before joining the Victorian Gas and Fuel Corporation and State Electricity Commission of Victoria, working on biogas utilisation, brown coal combustion, power production and cogeneration. He formed engineering consultants Ecopower Pty Ltd in 1989 and in 1997 acquired Rotary Heat Exchangers Pty Ltd, the company incorporated in 1968 as a result of scientific research in heat exchangers by CSIRO and Monash University.

He received three Australian Energy Awards in the fields of biogas utilisation, cogeneration and waste energy recovery and has published several technical papers in scientific journals in Australia and internationally.

Abstract

Much has been written and studied about Evaporative Cooling (EC) and Indirect Evaporative Cooling (IEC) in Australia since the 1960's, a period of nearly 60 years, with numerous publications in international scientific journals and AIRAH. Scientific papers and experimental work by Australia's CSIRO and others led the Australian chapter of R&D on this subject as well as spurring the HVAC industry in the areas of energy recovery and IEC.

In this paper the concepts and theory of EC and IEC are discussed and a mathematical computer model is developed and used to analyse three air flow IEC orientation options, termed 2, 3 and 4 Port IEC (PIEC).

The computer model combines the basic components of an IEC and the theoretical and empirical equations defining the behaviour of moist air. Excel basic macro programming, using iterative solution methods, was used to solve the equations and the results are presented in Tables and as cycles on the psychrometric chart for the different forms of IEC.

A new approach providing a better understanding of IEC, its behaviour, advantages and disadvantages, is presented, introducing the novel concept of the idealised dew point temperature cooler.

The ability of these systems to reduce air conditioning energy usage, particularly in extreme temperature conditions and low humidity, is examined. IEC offers the possibility of reducing peak electricity usage and greenhouse gas emissions particularly in extreme heat events in moderately humid climates, by substantially increasing COP.

1 INTRODUCTION

Indirect Evaporative Cooling has been researched since the 1960's, a period of nearly 60 years. Scientific papers and experimental work by Australia's CSIRO scientists and engineers, Dunkle, Banks and Ellul [1,2, & 3] and Pescod [4], led the Australian chapter of R&D on this subject as well as spurning the Australian HVAC industry in the areas of energy recovery and IEC.

This paper presents a new approach and a better understanding of IEC by mathematical modelling three different forms IEC can take. It examines their advantages and disadvantages and how they can reduce air conditioning energy consumption with substantially higher Coefficient of Performance (COP) than conventional systems.

A recent 2018 International Energy Agency, Green House Gas Control Technologies GHGT-14 conference in Melbourne predicted that the increased demand on air conditioning alone by mid-century could adversely impact climate change, by pushing up the world's temperature by more than 0.5C, where coal is the dominant form of power and called for the development of lower energy use air conditioning systems.

This paper shows how IEC offers the possibility of reducing peak electricity usage and greenhouse gas emissions, particularly in extreme heat events in moderately humid climates.

2 DIRECT EVAPORATIVE COOLING (EC)

Over the past several centuries and probably for many thousands of years, man has used water as a cooling medium, arguably as a refrigerant. The principle of the cooling effect of a wet rag was used to cool and preserve some perishable foods and was patented 130 years ago in Australia as the Coolgardie safe. The human body is designed to use the same principal to cool itself by producing sweat on the skin surface to allow cooling via evaporation.

Science has taught us that the heat required to evaporate water (refrigerant R718 H_2O), changing its phase from liquid to vapour, termed "latent heat" is a very large 2454 kJ/kg. This is exactly the mechanism of cooling of the environment by evaporating water from a surface utilising this large amount of latent cooling effect. The latent heat is removed from the surroundings thus cooling occurs.

Water has by far the greatest latent heat of evaporation of all the other refrigerants used in air conditioning and therefore promises to be by far the best option for economical cooling, even though it is used in an open 100% fresh air cycle, where it is dissipated to the environment, but with no harmful environmental effects. EC has the potential to increase COP for air conditioning far beyond present practice, in low to moderate humidity climates.

Processes involving dry air/water vapour mixtures such as EC are best represented on a psychrometric chart, with Cartesian co-ordinates dry bulb temperature t, and absolute moisture content w, as shown in Figures 1 to 4.

The sketch in Figure 1 shows how a room is cooled by direct EC with the room supply air 2 at temperature t2, mixing with room air to achieve room temperature tR at point R. The EC is said to be 100% efficient when it reaches its ideal saturation point, where it achieves its coldest temperature, the wet bulb temperature, twb. In practice the EC achieves a slightly higher temperature with efficiency less than 100%. This trajectory line 1 to 2 is the wet bulb line for every air state along this line. The trajectory, on the chart, of the air state 1 to 2, and 2 to R, is termed a cycle.

EC results in an increase of both absolute w and relative humidity RH% of the air being cooled and the refrigerant water is dissipated in the process. The increased humidity of the room supply air, both in relative and absolute terms, as well as their water consumption are major impediments for these systems, restricting their use to dry climates.

The cooling effect q from EC can be calculated from the air flow and temperature difference (t1 - twb1) which can then be used to calculate its COP.

Triangles A and B shown in Figure 1 indicate EC for two ambient temperatures t1 and t2 at constant w, where t2 > t1. From simple geometry triangle B is larger than triangle A so that we can deduce that

(t2 - twb2) > (t1 - twb1)
q2 > q1
$COP_B > COP_A$.

This shows that for EC, unlike conventional heat pump air conditioning, the higher the ambient to be cooled, for any constant w, the greater the cooling effect and the higher the COP with no limit on ambient temperature. In this paper, for comparison purposes, the maximum COP for EC, if its efficiency is 100%, is denoted COPEC and is presented in Tables 1 using a P_{in} of 1.5 kW.

As COP is inversely proportional to P_{in} the values given may increase substantially for systems designed to optimise the fan/flow energy efficiency having lower values of P_{in}.

3 COOLING WITHOUT THE ADDITION OF WATER

Figure 1 also shows point 1 being cooled without the addition or removal of water vapour. On the chart point 1 moves to the left on constant moisture content w1 horizontal line as water is neither added nor removed. If cooling continues without the addition of water, it reaches the 100% saturation point where the dew point temperature tdp, is reached and liquid water will start to condense from the air causing it to reduce its moisture content w1.

The evaporator cooling coil of a conventional refrigerated air conditioning system or heat pump, can both cool and dehumidify moist air using a finned-tube radiator type heat exchanger. In this case the air flow is cooled by the recirculating refrigerant flowing inside the heat exchanger and evaporatively cooling as it evaporates inside the tubes, to a temperature generally much lower than tdp of the air being cooled. It can cope with high ambient humidity air.

Unlike EC with water, the refrigerant is conserved and is recycled inside the cooling system. The author presented a numerical model depicting this process, Ellul [5], to calculate the trajectory of this type of moist air cooling on a psychrometric chart.

4 IEC AND ORIENTATION OPTIONS

The undesirable aspect of direct EC is that it results in the humidification of the cooled supply air to the building. This becomes even more critical for human comfort if the ambient hot air is already in a humid state. For this reason satisfactory EC is restricted to dry climates.

We can achieve cooling, without adding moisture and widen the application of comfort EC by combining it with an air to air sensible heat exchanger (HE) with the EC to separate the moisture laden EC air from the supply side air to the building. There are several ways this can be done. This paper explores three different airflow arrangement options as given in Figures 2,3 & 4 and compares some of their advantages and disadvantages. The concept of a 2 Port, 3 Port and 4 Port IEC (PIEC) is here introduced to refer to the total number of air inlets and outlets in a particular unit to describe the different IEC flow options being analysed.

The idealised maximum limit case of 100% efficient EC and HE is also analysed as this provides an instructive upper limit for comparison purposes, giving a better understanding of IEC.

Figure 2 shows the IEC arrangement where the air inlet to the EC is maintained the ambient air 1 and the room air is simply recirculated through the cooled side of the HE. When HE performance is maximum, t4 = t1 and t2 = t3, with the corresponding cycle shown as a rectangle. Room air is cooled on the dry, indoor side of the HE, while air on the wet side is heated 3 - 4 and exhausted at 4.

The COP performance of this system will be identical to the simple EC shown in Figure 1 as the cooling effect q is proportional to the same (t1 - twb1) in both cases. In other words the limit to cooling in this arrangement is twb1. However the main advantage over simple EC is that the room is not being humidified thus providing better comfort conditions.

Figure 2 shows the room R starting at t1 such that

tR = t1 = t4 and t2 = t3

and tR decreasing to match the room load such that

tR = t4 < t1

The rectangle reduces in size , while the COPEC remains constant based on (t1 - twb1).

5 THE 4PIEC WITH BALANCED HE AIR FLOW

We can achieve far better results, with lower than twb1 cooling if we consider the different air flow arrangement options shown in Figures 3 & 4. These take advantage of an iterative feedback cooling phenomenon which occurs because the drier air cooled by the HE is fed back to the EC, which in turn cools it further. This cooling sequence forces state point 2 towards the tdp which is much cooler than twb. An electrical analogy of this iterative feedback effect is the high pitched sound produced when microphone and speaker are too close.

A numerical computer model was developed to simulate the performance of these IEC options, based on the empirical equations for moist air given by Dunkle and Noris [1]. These were solved numerically by an iteration process using Excel macro basic programming. The computer modelling shows that in this arrangement, an iterative cooling feedback loop is formed which converges to a stable low temperature limit. This limit is shown to increase in temperature when less efficient component performance is used.

The model was used to analyse the performance of all the options presented, in order to determine such aspects as temperature, humidity and COP. The modelling results are presented in Tables 1 & 2. Balanced air flow is assumed in the HE of all the 4PIEC options with equal wet and dry flows through the HE.

The EC is placed on the exhaust air side. This physical separation ensures that the moisture laden cooled air is exhausted from the building through the wet side of the sensible HE, so that sensible cooling of the fresh air is achieved without introducing any moisture to the room. The return or room air R, feeding back to the EC, is progressively cooled, resulting in the above mentioned advantageous feedback iterative cooling.

The resulting cycle in Figure 3, shows the first two iterations as elongating rectangles with 2 cooling lower than twb1, with the added advantages of reducing both w as well as water evaporation as the rectangle elongates.

ΔtR = (tR – t2) = (tR – tS) represents the room load and (t1 – t2) represents the total cooling effect, as this is a 100% fresh air cooler.

6 4PIEC/2PIEC MODELLED PERFORMANCE

The computer model was used to analyse 4PIEC & 2PIEC performance presented in the following tables for the following supply air flow rate and two ambient conditions, 50 C and 40 C, with a dry w of 7 corresponding to a tdp of a low 8.7 C. P_{in} was increased to 2.4kW to account for the increased power usage caused by introducing the HE.

The case where ΔtR = 0 is here termed the idealised 2PIEC, as this transforms the unit to 2 Ports, in and out. In Tables 1a & 1b the EC and HE performance were also idealised at 100%:

QS = 1000 l/s
w1 = 7 g/kg ; tdp = 8.7C
P_{in} = 2.4 kW

t1 = 50 C; ηHE = ηEC = 1; COPEC = 21.6; tdp = 8.7C; twb = 23.3C

ΔtR C	**t2 = tS C**	**tR C**	**q**	**COP**
0 (2port)	8.7	8.7	50.2	20.9
1	9.9	10.9	48.7	20.3
5	17.6	22.6	39.4	16.4
10	25.7	35.7	29.5	12.3
15.7	34.4	50.0	19.0	7.9

TABLE 1a. 4PIEC/2PIEC IDEALISED PERFORMANCE

t1 = 40 C; ηHE = ηEC = 1; COPEC = 15.9; tdp = 8.7C; twb=20.4

ΔtR C	**t2 = tS C**	**tR C**	**q**	**COP**
0 (2port)	8.7	8.7	37.9	15.8
1	9.9	10.9	36.5	15.2
5	17.6	22.6	27.4	11.4
10	25.7	35.7	17.3	7.2
11	27.2	38.2	15.6	6.5
11.7	28.3	40.0	14.2	5.9

TABLE 1b. 4PIEC/2PIEC IDEALISED PERFORMANCE

In the above two table for the idealised 2 Port case, supply air reaches tdp maximising COP. As expected, COP also increases with ambient temperature. The effect of increasing ΔtR is to reduce the cooling effect and COP in both tables.

The high limit in ΔtR is reached when tR reaches t1. This would be equivalent to operating the cooler outside the building so that the room return air temperature equates to the ambient temperature.

An important advantage shown in these idealised results is that irrespective of how high the ambient temperature for a constant tdp or w, the cooled supply air temperature remains the same low value if ΔtR is maintained. This counters the present experience with conventional air conditioning and is a welcome relief considering the expected increase in frequency of extreme hot events due to climate change.

In the following tables the effect of realistic component efficiencies is introduced. The efficiencies of both HE and EC are reduced by only 10% to a high achievable 90%, in order to evaluate the effect of inefficiencies on the cooler.

t1 = 50 C ; ηHE = ηEC = 0.9; COPEC = 19.4; tdp = 8.7C

ΔtR C	t2 = tS C	tR C	q	COP
0 (2port)	16.1	16.1	46.0	19.1
1	17.7	18.7	43.6	18.1
5	23.7	28.7	33.5	14.8
10	31.0	41.0	25.6	10.8
13.9	36.1	50.0	18.8	7.8

TABLE 1c. 4PIEC/2PIEC PERFORMANCE

t1 = 40 C; ηHE = ηEC = 0.9; COPEC = 14.3; tdp = 8.7C

ΔtR C	t2 = tS C	tR C	q	COP
0 (2port)	14.4	14.4	34.5	14.4
1	16.0	17.0	32.4	13.5
5	22.1	27.1	24.1	10.1
10	29.4	39.4	14.3	6.0
10.2	29.7	40.0	13.9	5.8

TABLE 1d. 4PIEC/2PIEC PERFORMANCE

The results shown in Tables 1c & 1d show an approximate 20% drop in performance expected from a drop in component efficiencies of only 10%, suggesting the importance of utilising high efficiency components.

7 THE IDEALISED 2 PORT IEC

As discussed in the preceding section, we see that in the idealised performance Tables 1a & 1b, that an anomaly occurs when ΔtR = 0, which is also reflected in Figure 3. This represents the idealised 2PIEC arrangement, as 2, S and R is the same point. From these results we see that the corresponding cycle shown in Figure 3, reduces to just one straight line from 1 to tdp with the same line depicting the wet exhaust stream 3 to 4.

This mode is only presented to provide an understanding of the process and to examine the extreme boundaries of IEC. In this case, we do not have a 3rd or 4th Port. There is no cooling supply air for the building and no heat load to the cooler. This is the zero room load condition ΔtR = 0, giving the unachievable maximum values for cooling and COP which as expected increase with increasing t1 for a constant w1.

This process is also accompanied by a progressive reduction in evaporated water usage as state point 3 is being forced downward along the saturation line, until in the limit no water evaporation is required. Ambient air 1 enters the cooler and exits 4 at exactly the same condition, so theoretically no EC or water usage, in the limit is required.

The modelling therefore shows that for the IEC options presented, the low limit of the ambient dew point temperature tdp cannot be achieved in practice, as it requires unattainable ideal conditions i.e. when both HE and EC perform ideally with 100% efficiency, (ηHE = ηEC = 1) and no room load. Another outcome of these results is the importance of high efficiency components.

There was a reduction in water usage in all the 4PIEC models, compared to direct EC.

8 3PIEC

The final arrangement studied is that of 3PIEC shown in Figure 4. Here only one fan is required blowing fresh air through the high flow dry side inlet port, thus pressurising the IEC. This allows for flow 3 to separate and escape into the room forming the supply air port, while the balance air flow is sent through the wet side of the HE, prior to being exhausted as a humidified air stream. This increases fan energy and water usage, due to the higher pressure drops, caused by the much higher dry flow through HE and EC for the same supply air flow.

The HE is significantly unbalanced in air flow, as the dry side flow QD needs to be much larger than the wet side flow QW, to provide sufficient supply flow QS. This increases the temperature effectivity on the low flow, exhaust or wet side ηtW, but lowers the temperature effectivity on the high flow, dry supply side ηtD. The unbalanced flow thermodynamic relationships which apply, irrespective of the type of HE used, are derived in the Appendix. This airflow unbalance, reduces the length of (t1 - t2) and increases the length of (t4 - t3), of the cycle shown in Figure 4. The ability for the supply air 3 to approach tdp has therefore been reduced. P_{in} was increased to 3.4 kW, to account for the increase power usage to accommodate the increased air flow, resulting in reduced COP.

The computer model was used to analyse the performance presented in the following Tables 2a to 2e, with ambient air temperatures 30C, 40C and 50C. For comparative purposes the following conditions were used, maintaining P_{in} = 3.4kW, w1 = 7 g/kg_w and tdp = 8.7C and the inlet air fan dry flow side as a constant 2000 l/s. The effects of reducing supply air flow and component efficiencies are also analysed in Tables 2b to 2e.

QD = 2000 L/s
QS = 1000 L/s
ηtD = 0.5; ηtW = 1; ηEC = 1

t1 C	t2 C	t3 = tS C	w3	t4 C	q	COP
50	34.3	18.7	13.5	50.0	38.1	11.2
40	28.4	16.6	11.8	40.0	28.4	8.4
30	22.2	14.4	10.2	30.0	19.0	5.6

TABLE 2a. IDEALISED 3PIEC PERFORMANCE

QD = 2000 L/s
QS = 900 L/s
ηtD = 0.5; ηtW = 0.909; ηEC = 1

t1 C	t2 C	t3 = tS C	w3	t4 C	q	COP
50	34.3	18.7	13.5	50.0	34.3	10.1
40	28.3	16.6	11.8	40.0	25.6	7.5
30	22.2	14.4	10.2	30.0	17.1	5.0

TABLE 2b. 3PIEC PERFORMANCE

QD = 2000 L/s
QS = 900 L/s
ηtD = 0.4; ηtW = 0.727; ηEC = 1

t1 C	t2 C	t3 = tS C	w3	t4 C	q	COP
50	37.6	18.9	13.7	41.5	34.0	10.0
40	31.0	17.6	12.6	33.9	24.5	7.2
30	24.0	15.1	10.7	25.9	16.3	4.8

TABLE 2c. 3PIEC PERFORMANCE

QD = 2000 L/s
QS = 600 L/s
ηtD = 0.5; ηtW = 0.714; ηEC = 1

t1 C	t2 C	t3 = tS C	w3	t4 C	q	COP
50	34.3	18.7	13.5	50.0	22.8	6.7
40	28.3	16.6	11.8	40.0	17.0	5.0
30	22.2	14.4	10.2	30.0	11.4	3.4

TABLE 2d. 3PIEC PERFORMANCE

QD = 2000 L/s
QS = 600 L/s
ηtD = 0.4; ηtW = 0.571; ηEC = 1

t1 C	t2 C	t3 = tS C	w3	t4 C	q	COP
50	37.6	18.9	13.7	36.7	22.7	6.7
40	31.0	17.6	12.6	30.4	16.4	4.8
30	24.0	15.1	10.7	23.6	10.9	3.2

TABLE 2e. 3PIEC PERFORMANCE

Table 2a represents the idealised 100% efficiency case with 50% unbalance, so that the supply flow QS of 1000 L/s equates to those in Table 1. Tables 2b to 2e examines two other unbalance ratios with supply air reductions, while maintaining QD flow. The proportion of unbalance flow was used to calculate the reduction in ηtW and ηtD as shown in the Appendix.

Since ΔtR plays no part in this arrangement this option has an advantage when ΔtR cannot be controlled and when used as an open air cooler.

3PIEC arrangement lowers COP dependant on fan efficiency and increases water usage. There is also the disadvantage that supply w is being humidified, although not to the higher extent expected from a direct EC.

9 DISCUSSION COOLING PERFORMANCE RESULTS

The air flow is assumed to be 1000 l/s balanced flow in all 4PIEC examples, and is the maximum supply air flow for 3PIEC in Table 2a.

For all systems examined the energy input P_{in} is predominantly fan power which is not markedly affected by air conditions. The results given in these Tables can be scaled by changing the values of P_{in} and air flow to reflect any change in fan/flow design.

As expected, all Tables show that as ambient conditions rises from 30C to 50C, COP increases dramatically as a result of the much higher expected resultant evaporative cooling effect. For a refrigerated heat pump, COP can generally vary from 2.5 to about 5 for extremely efficient systems and does not rise with increasing ambient temperature. More commonly it is in the range 2.5 to 4. The model results show that theoretically we can achieve much higher values of COP with IEC. This is dependent on ambient temperatures being high, preferably over 30C and absolute humidity low. This promises substantial energy reductions of 200% to 300% with 100% fresh air cooling, with no limitation on the extreme temperature in low humidity climates. This is a much needed result in the efforts to fight climate change.

Where ambient humidity conditions rise above the ideal for IEC, it is possible to increase the COP of the combined system, if it is integrated together with a conventional dehumidifying heat pump.

The idealised 2PIEC where both HE and EC are assumed to be 100% efficient is of course impossible to achieve in practice, but the exercise helps enhance our understanding of the IEC and gives us an upper limit on COP and shows how water usage can be minimised. These results show that in order to achieve high COP it is important to use highly efficient HE, EC and fans.

The results indicate that the 4PIEC may have significant advantages over the 3PIEC in achieving higher COP and lower energy usage if ΔtR can be maintained below 5C with high efficiency components, otherwise the 3PIEC has the advantage, but with an increase in water usage. Appropriate system design and component selection will be vital.

10 CONCLUSIONS

The modelling analysis presented in this paper shows that Indirect Evaporative Cooling can provide low energy use air conditioning especially in extreme temperature environments, for dry low dew point temperature conditions exceeding the benefits of direct Evaporative Cooling systems and conventional refrigerated heat pump systems. Unlike refrigerated systems, COP for IEC increases dramatically as extreme temperatures rise.

Compared to direct EC, IEC can result in better performance and lower building humidity conditions, thus providing better comfort conditions as well as increasing the applicability of EC to regions where direct EC is unacceptable.

The results indicate that the 4 Port IEC may have significant advantages over the 3 Port IEC in achieving higher COP with lower water usage and therefore lower energy usage, if the room load temperature rise ΔtR can be maintained below 5C. This will require appropriate system design and component selection. In order to achieve good IEC performance, the high performance of both HE and EC are critical factors. The 3 Port IEC has an advantage when the room load cannot be controlled such as when used as an open air cooler, but will require a higher water usage.

The use of appropriate indirect evaporative cooling systems will result in increased air conditioning COP, as ambient temperature rises, resulting in reduced energy usage and operating cost, as well as assisting in the climate change battle, by reducing GHG emissions.

11 NOMENCLATURE

$COP = q/P_{in}$	Coefficient of Performance
$COPEC = q/P_{in}$	Coefficient of Performance of EC with 100% efficiency
Cp [kJ/kg C]	specific heat of moist air
P_{in} [kW]	total power required to run cooler
q [kW]	cooling effect; heat flow
Q [L/s]	air flow rate
R	return air, room condition
S	supply air
tR [C]	room air temperature
twb [C]	wet bulb air temperature
tdb, t [C]	dry bulb air temperature

tdp [C]	dew point temperature
w [$g_w/kg_{dry\ air}$]	air moisture content
$\Delta tR = (tR - t2) = (tR - tS)$	room load temperature difference
$\eta tD = (t1 - t2)/(t1 - t3)$	temperature effectivity on dry side of HE, refer Figures 3 & 4
$\eta tW = (t4 - t3)/(t1 - t3)$	temperature effectivity on wet side of HE, refer Figures 3 & 4
ηHE	temperature effectivity of balanced HE where ηtD = ηtW
ηEC	efficiency of EC
ρ [kg/m^3]	air density

12 APPENDIX

In order to examine the effect of unbalance flows on the HE we apply conservation of energy to equate the cooling on the wet side to the heating on the dry side of the HE, giving the following relationships where the air density and specific heat are constants:

$$qD = QD\ \rho\ Cp\ (t1 - t2) = qW = QW\ \rho\ Cp\ (t4 - t3)$$

Therefore: $qD/qW = QD(t1 - t2)/QW(t4 - t3)$

$\eta tD = (t1 - t2)/(t1 - t3)$; $\eta tW = (t4 - t3)/(t1 - t3)$

and: $\eta tD/\eta tW = (t1 - t2)/(t4 - t3) = QW/QD$; If $QD > QW$ then $\eta tW > \eta tD$

(Note for balanced flow QW = QD so that ηtD = ηtW = ηHE)

<u>For 3PIEC</u>, from Figure 4 $QS = (QD - QW)$

Therefore: $QS/QD = 1 - \eta tD/\eta tW$ & $\eta tD/\eta tW = (1 - QS/QD))$ & $\eta tW = \eta tD/(1 – QS/QD)$

The following Table presents the derived HE performance values used in the 3PIEC Tables. The two cases ηtD = 0.5 & 0.4 were selected, as large flow unbalance can result in high ηtW.

For QD = 2000 L/s

QS	QW	QS/QD	ηtD/ηtW	ηtW Assuming ηtD = 0.5	ηtW Assuming ηtD = 0.4
1000	1000	0.5	0.5	1	0.8
900	1100	0.45	0.55	0.909	0.727
600	1400	0.3	0.7	0.714	0.571

13 ACKNOWLEGEMENTS

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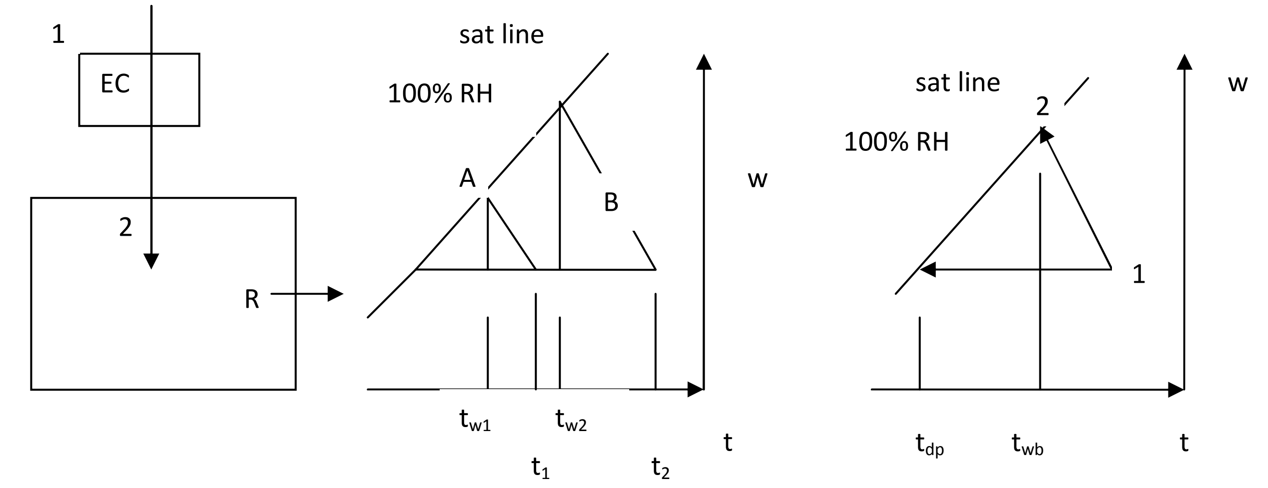

FIGURE 1 DIRECT EVAPORATIVE COOLING EC 1 - 2

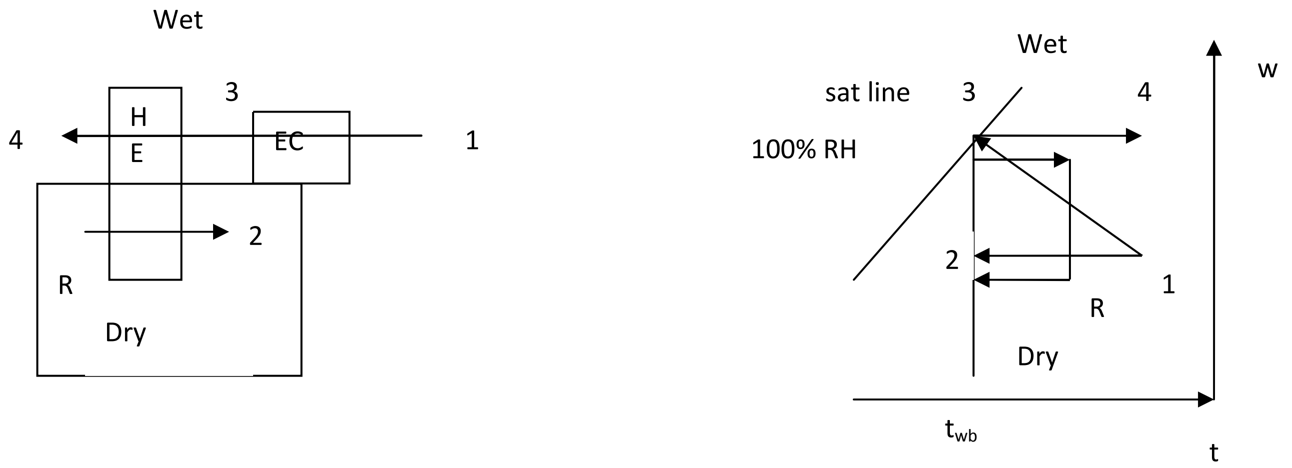

FIGURE 2 INDIRECT EVAPORATIVE COOLING - IEC

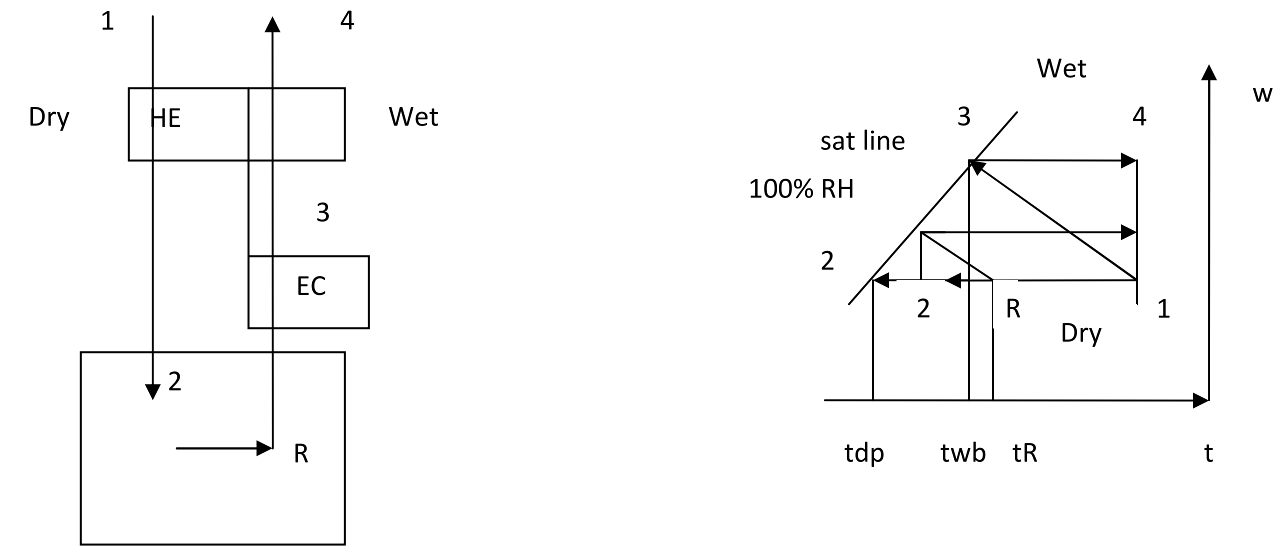

FIGURE 3 INDIRECT EVAPORATIVE COOLING - 4PIEC

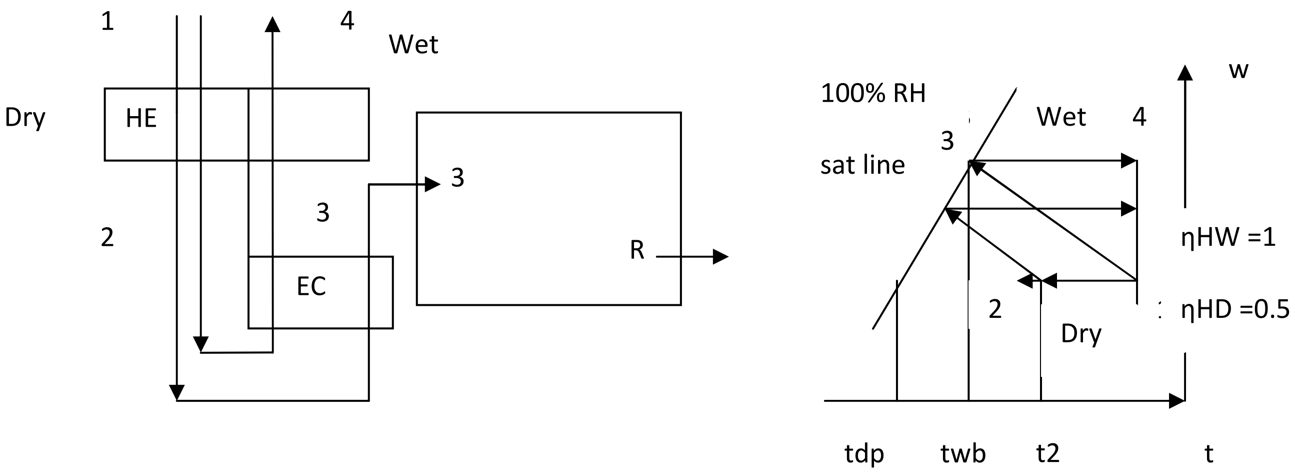

FIGURE 4 INDIRECT EVAPORATIVE COOLING - 3PIEC