Experiment 5

HEAT PUMP AND AIR COOLER

1. OBJECTIVE

- Demonstrate the performance of the equipment in both heating and cooling modes.
- Evaluate the coefficient of performance (COP) when operating as a heat pump.
- Evaluate the coefficient of performance (COP) when operating as an air cooler.

2. INTRODUCTION & THEORETICAL BACKGROUND

An air cooler and a heat pump essentially comprise the same cycle components, however, it is their objectives that differentiate between them. In an air cooler, or a refrigerator, the heat extracted from the air, Q_L (absorbed by the refrigerant in the evaporator) is the required cooling effect.

However, the heat rejected to the circulating water in the condenser, Q_H although necessary, is not the objective.

On the other hand, the heat pump utilizes the heat absorbed from a low-temperature source to maintain a heated space at a higher temperature, thus Q_H is the required heating effect.

The theoretical model with which such a device is compared in order to evaluate its performance is the reversible simple refrigeration cycle shown in Figure 1.

Such a cycle takes in heat isothermally from a reservoir at temperature T_L and rejects heat isothermally to another reservoir at temperature T_H. The intervening processes are adiabatic reversible expansion and compression processes. (See Figure 1)

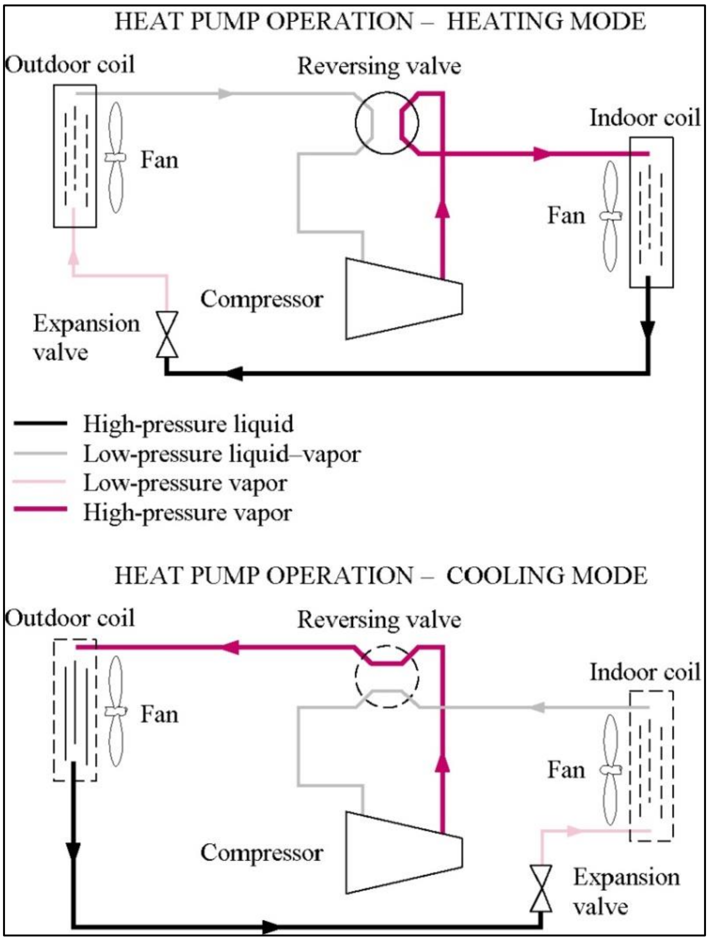

Fig. 1: **Reversible Simple Refrigeration Cycle**

The coefficient of performance of the machine when operating as an air cooler is

$$\boldsymbol{COP_R = \frac{Q_L}{E_C} = \frac{Q_L}{Q_H - Q_L}} \tag{1}$$

And when operating as a heat pump is

$$\boldsymbol{COP_{HP} = \frac{Q_H}{E_C} = \frac{Q_H}{Q_H - Q_L}} \tag{2}$$

However, the numerical values will be less than those corresponding to the ideal reversible engine. For an ideal reversible engine the heat transfer ratio Q_L/Q_H can be replaced by the ratio of the absolute temperature of the two reservoirs as follows:

Thus for a reversible air cooler, the COP is

$$\boldsymbol{COP_R = \frac{T_L}{T_H - T_L}} \tag{3}$$

And for a reversible heat pump the COP is

$$\boldsymbol{COP_{HP} = \frac{T_H}{T_H - T_L}} \tag{4}$$

Note that in all cases, $COP > 1$ and $COP_{HP} = COP_R + 1$

2. APPARATUS

The apparatus consists of two separate units, the air conditioner and the control console. The two units are connected, by electrical cables, thermocouple wires and nylon water pipes. (See Figure 2)

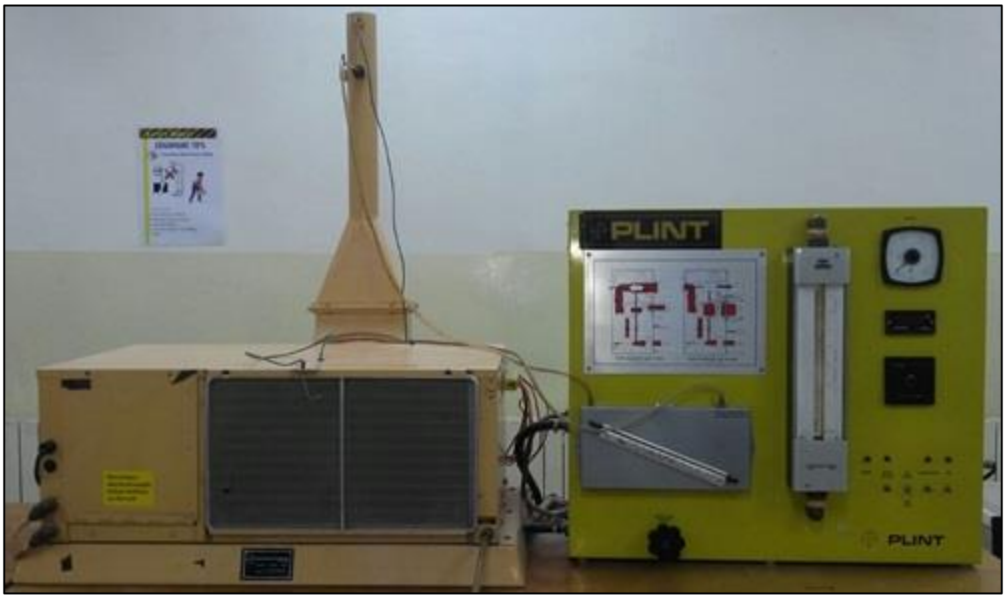

Fig. 2: Apparatus - General View

The air conditioner unit is completely self-contained and consists of a hermetically sealed *refrigeration system driven by a 0.4 kW motor, a refrigerant-to-water heat exchanger, a refrigerant-to-air heat exchanger, reversing valve, fan and motor, condensate collector and electrical controls. The air to be conditioned enters by way of the finned, refrigerant-to-air heat exchanger, passes through the centrifugal fan which is driven by a motor immersed in the air flow. The air is then discharged to a duct of circular cross-section carrying a pitot tube and a thermocouple. When the air is being cooled, and the relative humidity is high enough, moisture is deposited on the heat exchanger and drained off to a measuring vessel.

The control console unit carries all the electrical switches and fuses, a wattmeter and a multi-point digital temperature indicator. It also carries a flowmeter for measurement of water passing through the conditioner, an inclined manometer for use with the Pitot tube for air flow measurement and a vessel containing a 2 kW immersion heater. The latter is sometimes necessary when the conditioner is operating as a heat pump extracting heat from the circulating water since, if the temperature of the water on entry to the conditioner falls below about 10°C, there is a likelihood of freezing taking place.

*refrigerant: R22, Chlorodifluoromethane $CHClF_2$,The boiling point of R22 is -40.8 °C

The following instruments are provided:

1. Wattmeter for measurement of electrical power input to compressor and to fan.
2. Multipoint digital temperature indicator.
3. Whirling psychometer for measurement of relative humidity of air entering and leaving the conditioner.
4. Pitot-static tube and inclined manometer for measurement of air flow.
5. Cooling water flowmeter.
6. Graduated collecting vessel for condensate.
7. Thermocouples for temperature measurement
8. This apparatus uses water as a source and air as a sink in heating mode, and water as a sink and air as a source in cooling mode.

3. PROCEDURE

1. Before starting the machine ensure that the cooling water is turned on and regulated to give a flow of about 4 L/min. If the temperature of the cooling water entering the apparatus is less than 10°C and the machine is to operate as a heat pump, the water must also be switched on to ensure that freezing does not take place in the refrigerant-to-water heat exchanger.
2. Select cooling or heating as desired and switch on the compressor and fan. It takes between thirty minutes and one hour for temperature conditions in the apparatus to stabilize. It is suggested that a set of readings should be taken every ten minutes and continued until two successive readings show a change in air and water temperature of not more than 0.3°C.
3. The wattmeter shows the total electrical input to both the fan and the refrigerator compressor. The input to the fan alone may be measured by momentarily switching off the fan to measure the power to the compressor motor, then subtracting this reading from the reading for both the fan and the compressor.
4. The relative humidity of the air entering the conditioner should be measured by means of the sling hygrometer provided. Ensure that the reservoir in the hygrometer is filled with water and that it is whirled for a sufficiently long period (about thirty seconds) to give steady readings.
5. The relative humidity can be then used to find the humidity ratio ω, with the help of the psychometric chart, given at the end of the experiment.

6. When operating as a heater there will be no moisture deposited in the conditioner, since the relative humidity falls as the air passes through the machine.
7. When operating as a cooler there may or may not be deposition of moisture depending on the relative humidity of the air entering the machine. It may take a considerable time, as much as two to three hours, for the rate of flow of condensate to reach a stable value and, for this reason; it may be desirable when a cooling test is to be made to start up the apparatus several hours in advance of the laboratory period. There is some intrinsic variation in condensate flow rate and the measuring period should be as long as possible.

Record the following temperatures:

T_1 Air at inlet

T_2 Air at discharge

T_3 Circulating water at inlet

T_4 Circulating water at discharge

		Heat Pump	Air Cooler
T_5	Compressor	Discharge	Inlet
T_6	Compressor	Inlet	Discharge
T_7	Refrigerant-to-water heat exchanger	Discharge	Inlet
T_8	Refrigerant-to-water heat exchanger	Inlet	Discharge
T_9	Refrigerant-to-air heat exchanger	Inlet	Discharge
T_{10}	Refrigerant-to-air heat exchanger	Discharge	Inlet

The air flow is measured by means of a pitot tube mounted in the center of the discharge duct. The pressure of air at this point is effectively equal to that of the atmosphere, P_a If H mm H_2O is the velocity head measured by the Pitot tube, the mass flow rate is given by:

$$\dot{m}_a = 0.00105\sqrt{\frac{H \times P_a}{T_2}} \quad kg/s \tag{5}$$

Where H: Manometer reading in mm H_2O

P_a: Atmospheric pressure in N/m^2

T_2: Air temperature at discharge in K

Part A
Heat Pump

1. THEORY

The machine arrangement when operating as a Heat Pump is shown in Figure 1. The Figure shows the theoretical circuit of apparatus in heating mode, drawing energy from the circulating water and delivering it to the air. The compressor delivers refrigerant under pressure and at high temperature to the refrigerant-to-air heat exchanger, where heat is transferred to the air and the refrigerant condenses in the process. The refrigerant then passes through a restriction tube to the low pressure side of the circuit and to the refrigerant-to-water heat exchanger where it evaporates, taking up heat from the circulating water. It then returns to the compressor. (See Figure 1 & 2)

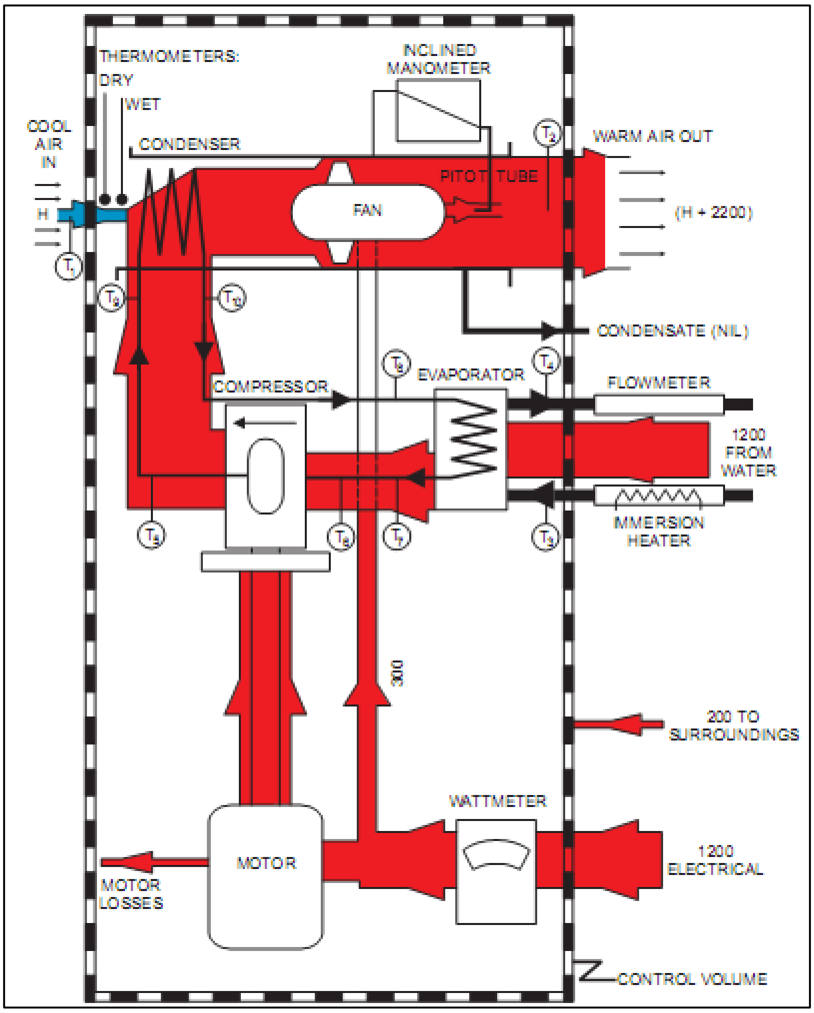

Fig. 1: Heat Pump - Schematic Layout

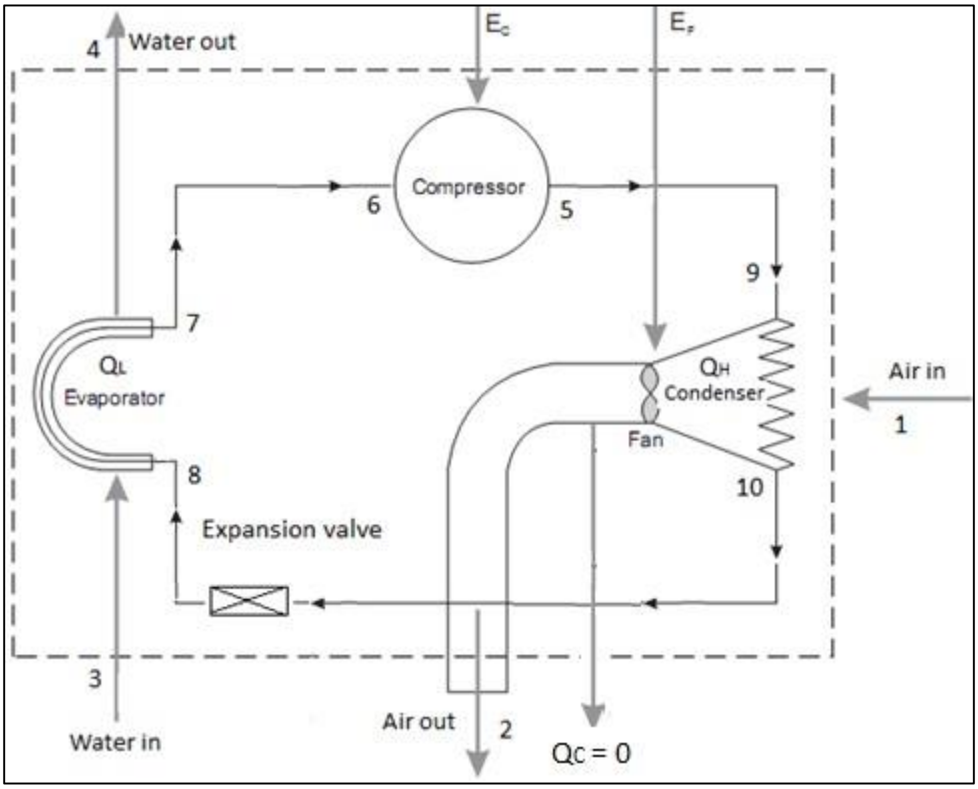

Fig. 2: Energy Flow Diagram for Heat Pump

The steady state steady flow (SSSF) equation for the system, see Figure 2 is the same as for the air cooler and may be written as:

$$Q_H - E_C - E_F = Q_L$$

Where

$$Q_L = \dot{m}_w\left(h_{f4} - h_{f3} \right) = \dot{m}_w C_{Pw}(T_4 - T_3)$$

$$Q_H = \dot{m}_a(h_2 - h_1) + \dot{m}_a(W_2 h_{v2} - W_1 h_{v1})$$

$\dot{m}_a$ = Mass flow rate of dry air (exchange heat with condenser)

$\dot{m}_w$ = Mass flow rate of water (exchange heat with evaporator)

h_1 *and* h_2 = Enthalpy of dry air at inlet and exit (from psychometric chart)

W_1 *and* W_2 = Humidity ratio of air at inlet and exit (from psychometric chart)

h_{v1} *and* h_{v2} = Enthalpy of water vapor in air at inlet and exit

($h_{v1} = h_{g1}$ *and* $h_{v2} = h_{g2}$ From steam table)

$E_C =$ Electrical input to the compressor

$E_F =$ Electrical input to the fan

The coefficient of performance may be defined in two different ways:

The external coefficient of performance

$$COP_{HP,\ E} = \frac{\dot{m}_a(\,h_2 - h_1\,) + \dot{m}_a(\,W_2 h_{v2} - W_1 h_{v1}\,)}{E_C + E_F} \qquad (1)$$

And the corresponding value for an ideal machine operating between the same mean temperatures is:

The max coefficient of performance

$$COP_{HP,\ max} = \frac{(T_1 + T_2)/2}{(T_1 + T_2)/2 - (T_3 + T_4)/2} \qquad (2)$$

Where the temperatures in these equations are in Kelvin

The internal coefficient of performance

Again, the performance of the basic heat pump is not identical with the overall performance since the energy supplied to the circulating fan is not chargeable to the heat pump and appears in the discharge air in which the fan and motor are immersed. This leads to the internal coefficient:

$$COP_{HP,\ I} = \frac{\dot{m}_a(\,h_2 - h_1\,) + \dot{m}_a(\,W_2 h_{v2} - W_1 h_{v1}\,) - E_F}{E_C} \qquad (3)$$

This may be compared with an ideal performance based upon the temperature difference across the refrigerator circuit

The ideal max coefficient of performance

$$COP_{HP,ideal\,max} = \frac{T_{10}}{T_{10} - T_8} \qquad (4)$$

Where the temperatures in these equations are in Kelvin

In a real machine such as the present one, the coefficient of performance falls short of the ideal for a number of reasons, the most important are:

- Electrical and mechanical losses in both the fan and the compressor.
- The imperfection (irreversibility) of the refrigeration cycle itself.
- The necessity for temperature differences between refrigerant and air, and between refrigerant and water. As a result of which the refrigerant cycle operates between substantially wider temperature limits than those applicable to the water and air forming the source and sink.

2. OBSERVATIONS

Table 1: DATA OBSERVED

Experiential Temperatures			
Location		**Tem p. ℃**	**Value**
Air in	Inlet Dry Bulb Temp.	$T_{1,db} = T_1$	
	Inlet Wet Bulb Temp.	$T_{1,wb}$	
Air out	Exit Dry Bulb Temp.	$T_{2,db} = T_2$	
	Exit Wet Bulb Temp.	$T_{2,wb}$	
Water	inlet	T_3	
	outlet	T_4	
Compressor	outlet	T_5	
	inlet	T_6	
Evaporator	outlet	T_7	
	inlet	T_8	
Condenser	inlet	T_9	
	outlet	T_{10}	
Condensate Temp.	at Discharge	T_C	
Manometer Reading	H	mm H_2O	
Mass Flow Rates			
Circulating Water	$\dot{m}_w$	l/min	
Condensate at Discharge	$\dot{m}_C$	ml/min	
Dry Air	$\dot{m}_a$	kg/S	
Power Consumed			
Total Power	E_T	kW	
Compressor Power	E_C	kW	
Fan Power	E_F	kW	

3. RESULTS & DISCUSSION

Table 2: ENTHALPIES

Item	Symbol	Unit	Value
Dry air entering conditioner	h_1	kJ/kg	
Dry air leaving conditioner	h_2	kJ/kg	
Water vapor entering conditioner	h_{v1}	kJ/kg	
Water vapor leaving conditioner	h_{v2}	kJ/kg	
Condensate	h_C	kJ/kg	
Humidity ratio of air at inlet	W_1	kg/kg	
Humidity ratio of air at exit	W_2	kg/kg	
Mass flow rate of dry air	$\dot{m}_a$	kg/s	

Table3: SUMMARY OF RESULTS

Item	$(COP_{AC})_{I}$	$(COP_{AC})_{E}$	$(COP_{AC})_{max}$	$(COP_{AC})_{I,\,max}$
Value				

1. All the results were recorded and tabulated under the results table.
2. Calculate the COP for the Heat Pump showing your work on psychometric chart and steam table attached and compare it with the ideal theoretical value.
3. Given the diameter of the discharge duct of the heat pump to be D = 0.073 m, derive the mass flow rate of air in kg/s as per the formula below.

$$\dot{m}_a = 0.00105\sqrt{\frac{H \cdot P_a}{T_2}} \quad kg/s$$

   And the velocity correction $U = 0.96U$
4. Discuss any discrepancy and the possible causes of errors in this experiment.
5. Write your own opinions about the results. What might be? Discuss about whether the results are acceptable or not?

Part B
Air Cooler

1. THEORY

The machine arrangement when operating as an Air Cooler is shown in Figure 1. The figure shows the theoretical circuit in cooling mode. The direction of flow is now reversed. The refrigerant passes from the compressor to the refrigerant-to-water heat exchanger, where it gives up heat to the cooling water, subsequently passing through the reducing valve to the refrigerant-to-air heat exchanger, where it evaporates, and extracting heat from the air. When the apparatus acts in cooling mode, the air is sometimes cooled to below the dew point and condensate is deposited.

(See Figure 1& 2)

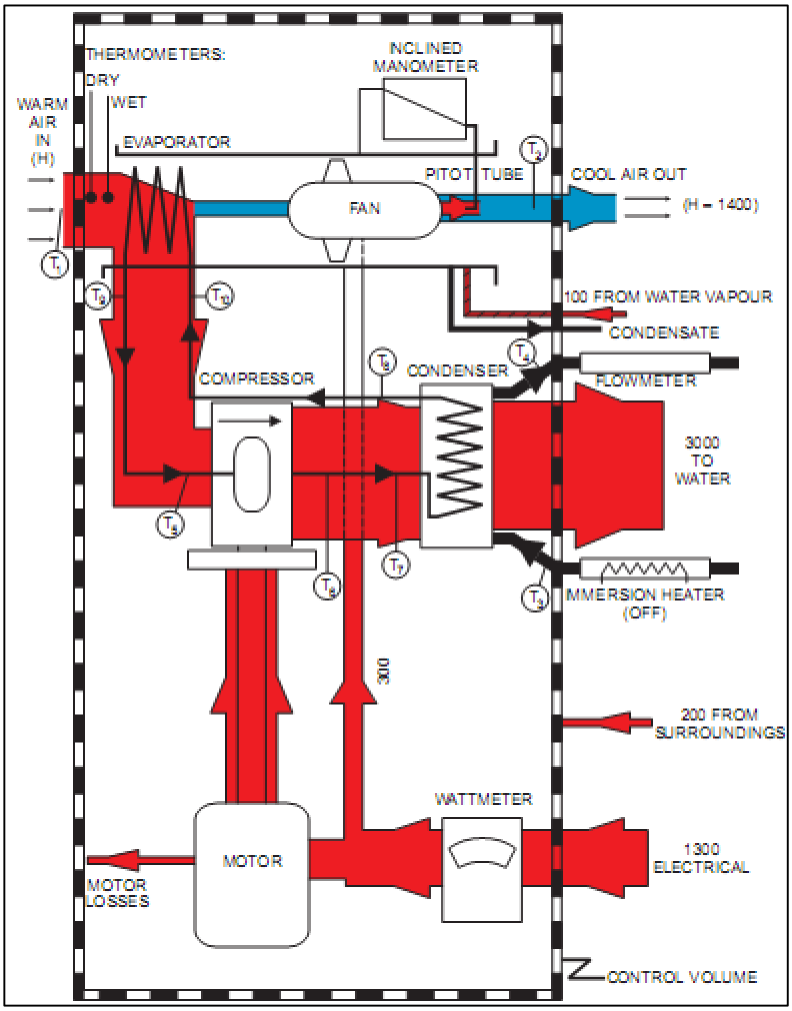

Fig. 1: Air Cooler - Schematic Layout

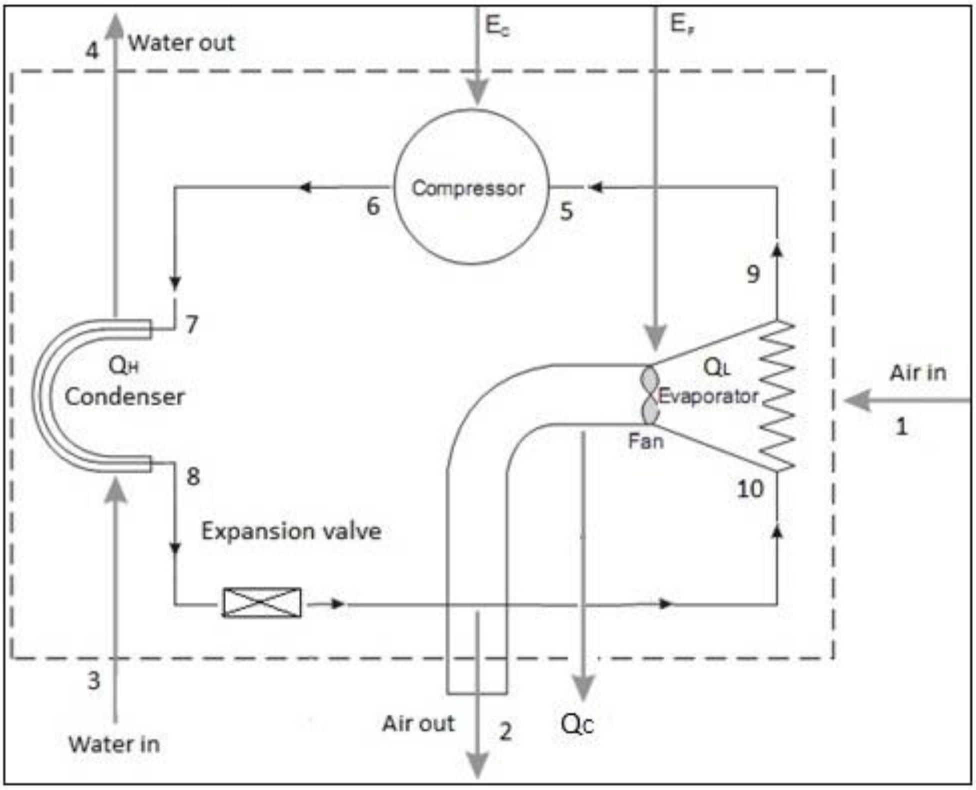

Fig. 2: Energy Flow Diagram for Air Cooler

The steady state steady flow (SSSF) equation for the system may be derived from the energy flow diagram Figure 2, as

$$Q_H - E_C - E_F = Q_L$$

Where:

$Q_H = \dot{m}_w(h_{f4} - h_{f3}) = \dot{m}_w C_{Pw}(T_4 - T_3)$

$Q_L = \dot{m}_a(h_1 - h_2) + \dot{m}_a(W_1 h_{v1} - W_2 h_{v2}) - Q_C$

$Q_C = \dot{m}_C C_{Pw} T_C$

$\dot{m}_a$ = Mass flow rate of dry air (exchange heat with evaporator)

$\dot{m}_w$ = Mass flow rate of water (exchange heat with condenser)

$\dot{m}_C$ = Mass flow rate of condensate water from the air stream

h_1 *and* h_2 = Enthalpy of dry air at inlet and exit (from psychometric chart)

W_1 *and* W_2 = Humidity ratio of air at inlet and exit (from psychometric chart)

h_{v1} *and* h_{v2} = Enthalpy of water vapor in air at inlet and exit

($h_{v1} = h_{g1}$ *and* $h_{v2} = h_{g2}$ From steam table)

T_C = temperature of condensate water

E_C = Electrical input to the compressor

E_F = Electrical input to the fan

The coefficient of performance may be defined in two different ways:

The external coefficient of performance

$$COP_{AC,\ E} = \frac{\dot{m}_a(h_1 - h_2) + \dot{m}_a(W_1 h_{v1} - W_2 h_{v2}) - \dot{m}_C C_{Pw} T_C}{E_C + E_F} \quad (1)$$

And the corresponding value for an ideal machine operating between the same mean temperatures is:

The max coefficient of performance

$$\boldsymbol{COP_{HP,\ max} = \frac{(T_1 + T_2)/2}{(T_3 + T_4)/2 - (T_1 + T_2)/2}} \quad (2)$$

Where the temperatures in these equations are in Kelvin

The internal coefficient of performance

The performance of the basic refrigerator is not identical with the overall performance since the energy supplied to the circulating fan is not chargeable to the refrigerator and appears in the discharge air in which the fan and motor are immersed. This leads to the internal coefficient:

$$\boldsymbol{COP_{AC,\ I} = \frac{\dot{m}_a(h_1 - h_2) + \dot{m}_a(W_1 h_{v1} - W_2 h_{v2}) - \dot{m}_C C_{Pw} T_C - E_F}{E_C}} \quad (3)$$

The ideal max coefficient of performance

This may be compared with an ideal performance based upon the temperature difference across the refrigerator circuit

$$\boldsymbol{COP_{HP,ideal\ max} = \frac{T_{10}}{T_8 - T_{10}}} \quad (4)$$

Where the temperatures in these equations are in Kelvin

In a real machine such as the present one, the coefficient of performance falls short of the ideal for a number of reasons, the most important are:

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Compressor	inlet	T_5	
	outlet	T_6	
Condenser	inlet	T_7	
	outlet	T_8	
Evaporator	outlet	T_9	
	inlet	T_{10}	
Condensate Temp.	at Discharge	$\mathbf{T_C}$	
Manometer Reading	H	mm H_2O	
Mass Flow Rates			
Circulating Water	$\dot{m}_w$	l/min	
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Dry Air	$\dot{m}_a$	kg/S	
Power Consumed			
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3. RESULTS & DISCUSSION

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Water vapor leaving conditioner	h_{v2}	kJ/kg	
Condensate	h_C	kJ/kg	
Humidity ratio of air at inlet	W_1	kg/kg	
Humidity ratio of air at exit	W_2	kg/kg	
Mass flow rate of dry air	$\dot{m}_a$	kg/s	

Table3:COP RESULTS

Item	$(COP_{AC})_{I}$	$(COP_{AC})_{E}$	$(COP_{AC})_{max}$	$(COP_{AC})_{I,\,max}$
Value				

1. All the results were recorded and tabulated under the results table.
2. Calculate the COP for the Heat Pump showing your work on psychometric chart and steam table attached and compare it with the ideal theoretical value.
3. Given the diameter of the discharge duct of the heat pump to be D = 0.073 m, derive the mass flow rate of air in kg/s as per the formula below.

$$\dot{m}_a = 0.00105\sqrt{\frac{H \cdot P_a}{T_2}} \quad kg/s$$

And the velocity correction $U = 0.96U$

4. Discuss any discrepancy and the possible causes of errors in this experiment.
5. Write your own opinions about the results. What might be? Discuss about whether the results are acceptable or not?

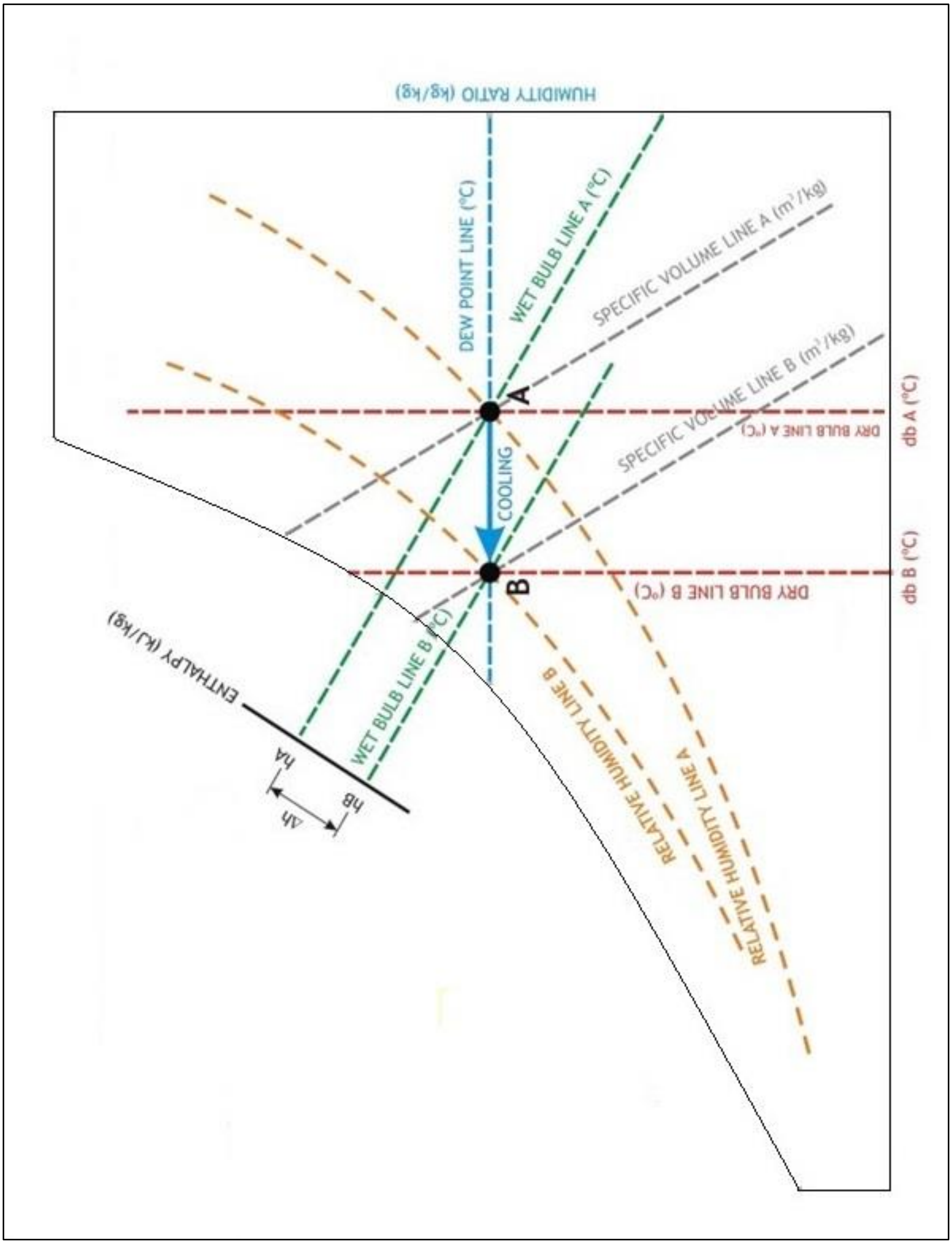
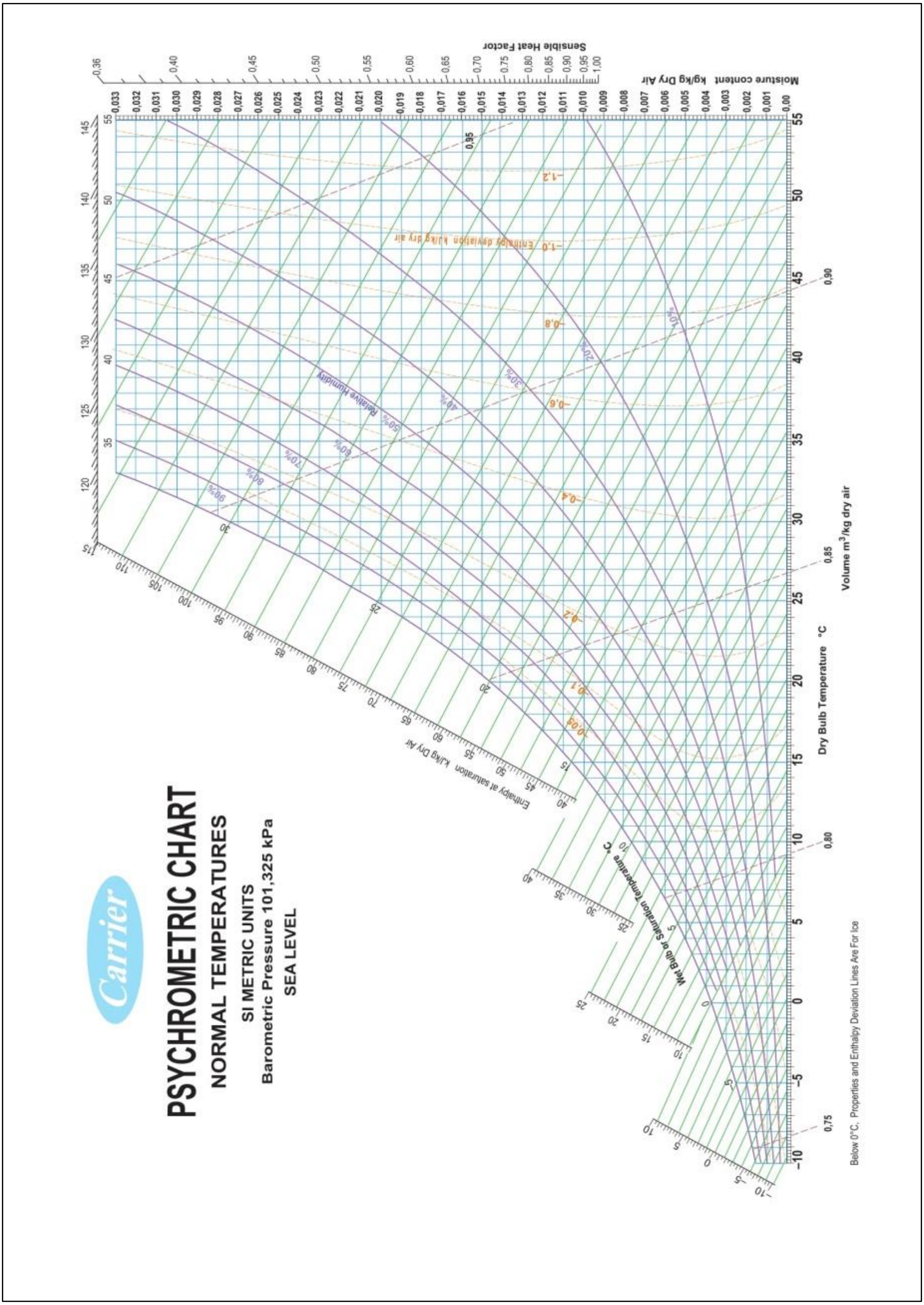


Fig. 2: Psychometric Chart (Basics)


Carrier
PSYCHROMETRIC CHART
NORMAL TEMPERATURES
SI METRIC UNITS
Barometric Pressure 101,325 kPa
SEA LEVEL
Dry Bulb Temperature °C
Volume m³/kg dry air
Moisture content kg/kg Dry Air
Sensible Heat Factor
Enthalpy at saturation kJ/kg Dry Air
Wet Bulb or Saturation Temperature °C
Relative Humidity
Enthalpy deviation kJ/kg dry air
Below 0°C, Properties and Enthalpy Deviation Lines Are For Ice

SATURATED STEAM - TEMPERATURE TABLE

		Spec. vol. m 3/kg		Enthalpy kJ/kg	
T ^oC	P bar	Sat liq. vf X1000	Sat. Vap. vg	Sat. liq. hf	Sat vap. hg
0.01	0.0061	1.0002	206.1	0.01	2501
4	0.0081	1.0001	157.2	16.79	2509
5	0.0087	1.0001	147.1	21	2511
6	0.0093	1.0001	137.7	25.21	2512
8	0.0107	1.0001	120.9	33.61	2516
10	0.0123	1.0001	106.4	42.01	2520
11	0.0131	1.0007	99.86	46.19	2522
12	0.0140	1.0007	93.79	50.4	2523
13	0.0150	1.0007	88.13	54.59	2525
14	0.0160	1.0007	82.85	58.8	2527
15	0.0170	1.0007	77.93	62.99	2529
16	0.0182	1.0013	73.34	67.17	2531
17	0.0194	1.0013	69.05	71.36	2533
18	0.0206	1.0013	65.04	75.57	2534
19	0.0220	1.0013	61.30	79.76	2536
20	0.0234	1.002	57.79	83.94	2538
21	0.0249	1.002	54.52	88.13	2540
22	0.0264	1.002	51.45	92.32	2542
23	0.0281	1.0026	48.58	96.5	2544
24	0.0298	1.0026	45.89	100.7	2545
25	0.0317	1.0032	43.36	104.9	2547
26	0.0336	1.0032	41.00	109.0	2549
27	0.0357	1.0032	38.78	113.2	2551
28	0.0378	1.0038	36.69	117.4	2553
29	0.0401	1.0038	34.73	121.6	2554
30	0.0425	1.0045	32.90	125.8	2556
31	0.0450	1.0045	31.17	130.0	2558
32	0.0476	1.0051	29.54	134.1	2560
33	0.0503	1.0051	28.01	138.3	2562
34	0.0532	1.0057	26.57	142.5	2563
35	0.0563	1.0057	25.22	146.7	2565
36	0.0595	1.0063	23.94	150.8	2567
38	0.0663	1.007	21.60	159.2	2571
40	0.0738	1.0076	19.52	167.5	2574
45	0.0959	1.010	15.26	188.4	2583
50	0.1235	1.012	12.03	209.3	2592
55	0.1576	1.015	9.569	230.2	2601
60	0.1994	1.017	7.671	251.1	2610
65	0.2503	1.020	6.197	272.0	2618
70	0.3119	1.023	5.042	293.0	2627
75	0.3858	1.026	4.131	313.9	2635
80	0.4739	1.029	3.407	334.9	2644

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