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Nomenclature

A_c	Cross section for air flow
ARI	American refrigeration institute.
$COP_{c.a}$	Cooling coefficient of performance with respect to air .
$COP_{c.t}$	Total cooling coefficient of performance.
$COP_{c.w}$	Cooling coefficient of performance with respect to water.
COP_h	Heating coefficient of performance.
$C_{p.w}$	Specific heat enthalpy at constant pressure of water.
$C_{p.a}$	Specific heat enthalpy at constant pressure of air.
DBT	Dry bulb temperature.
h_p	Enthalpy at any point or state P
lpm	Litres per minute.
$\dot{m}_a$	Mass flow rate of air in evaporator
$\dot{m}_{cw}$	Mass flow rate of water in condenser heat exchanger.
$\dot{m}_{dshw}$	Mass flow rate of water in de-superheator heat exchanger.
$\dot{m}_{ew}$	Mass flow rate of water in evaporator heat exchanger.
$\dot{m}_r$	Mass flow rate of refrigerant .
$\dot{Q}_a$	Rate of heat absorption by air
$\dot{Q}_{c.w}$	Rate of heat absorption by condenser water.
$\dot{Q}_{ds.w}$	Rate of heat absorption by de-superheator water.
$\dot{Q}_{e.w}$	Rate of heat absorption by evaporator water
$\%Rh_{IN}$	Inlet percentage relative humidity of air.
$\%Rh_{OUT}$	Outlet percentage relative humidity of air.
Δt_{con}	Temperature difference between inlet and outlet condenser water
$t_{c.sat}$	condensing temperature of saturated refrigerant
$t_{c.r.o}$	Condenser refrigerant outlet temperature
$t_{ds\ wi}$	De-superheator water inlet temperature
$t_{ds\ wo}$	De-superheator water outlet temperature
$t_{e.a.i}$	Evaporator air inlet temperature
$t_{e.a.o}$	Evaporator air outlet temperature
$t_{e.r.o}$	Evaporator refrigerant outlet temperature
$t_{e.sat}$	Evaporating temperature of saturated refrigerant
Δt_{evap}	Temperature difference between inlet and outlet evaporator water
$t_{e.w.i}$	Evaporator water inlet temperature
$t_{e.w.o}$	Evaporator water outlet temperature
W_{com}	Power input to the compressor.

1 Introduction

A heat pump is a machine or device that moves heat from one location (the 'source') to another location (the 'sink' or 'heat sink') using mechanical work.
Different types of heat pumps include ‘air source heat pump (extracts heat from outside air)’, ‘geothermal heat pump (extracts heat from the ground or similar sources)’.

Multi Utility Heat pumps are a unique kind of heating system, because they can do the work of both a furnace and an air conditioner. Thus, there's no need to install separate systems to heat and cool your home. Heat pumps can also work extremely efficiently, because they simply transfer heat, rather than burn fuel to create it.

Multi Utility heat pump studied during the project is Water to Water and Air to Water Heat pump. It is a Vapor compression unit with R22 as a working fluid. It can produce simultaneously cold water, hot water and can cool the ambient air. Waste heat can be recovered using novel patented Tube-Tube Heat Exchangers. These vented double wall tube and tube heat exchangers (Indian Patent # 205 362) enable recovery of the waste heat in a reliable and cost effective manner.

During the course of the project, testing of Multi Utility Heat Pump (MUHP) is carried out during which Coefficient of Performance (COP) is obtained as a function of various operating parameters. In the experiment, water temperature at evaporator outlet, water temperature at condenser outlet was varied by changing mass flow rate of water and subsequent COP is calculated.

Multi utility heat pump are very effective since they generate multiple utilities, have low operating cost and small payback period. Also in the era of scarce electricity, MUHP is attractive option.

2 List of components

2.1 Danfoss compressor.

General Characteristics

Model number (on compressor	SM185S4RC
Code number for Single pack	SM185-4RI
Code number for Industrial pack	SM185-4RM
Drawing number	8551119a
Suction and discharge connections Suction connection Discharge connection	Rotolock 2-1/4 J Rotolock 1-3/4 J Rotolock
Oil sight glass Oil equalisation connection Oil drain connection LP gauge port Swept volume Displacement @ Nominal speed	Threaded 3/8”” flare SAE 1/4”” NPT Schrader 249.9 cm3/rev 43.5 m3/h@2900 rpm
Net weight Oil charge	100 kg 6.2 litre, Mineral - 160P
Maximum system test pressure Low Side / High side	25 bar(g) / 32 bar(g)
Maximum differential test pressure	24 bar
Maximum number of starts per	12
Refrigerant charge limit	13.5 kg
Approved refrigerants	R22

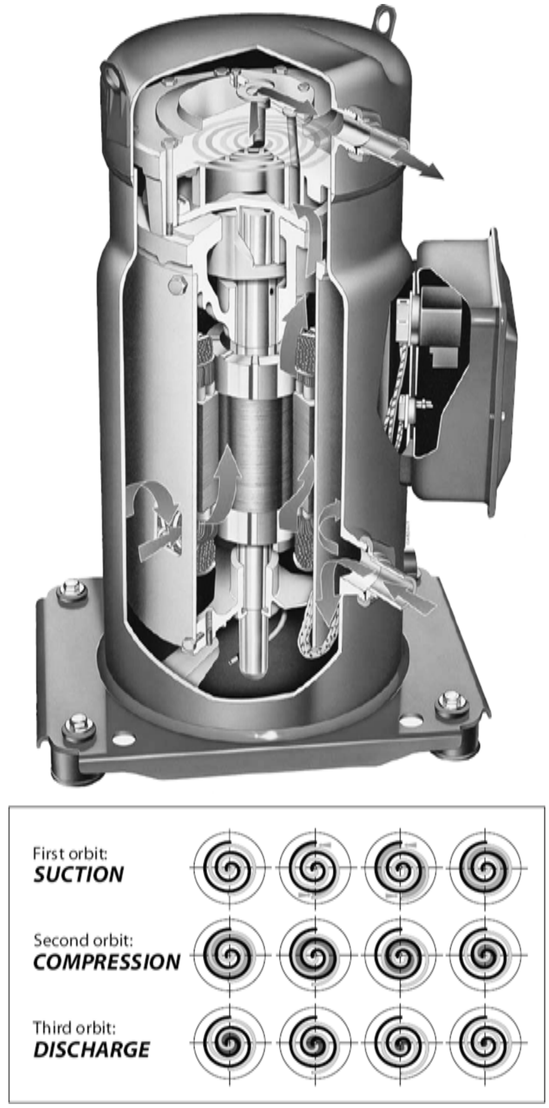


Figure 2.1Danfoss Compressor [2]

Electrical Characteristics

Nominal voltage	380-400V/3/50Hz
Voltage range	340-440 V @ 50Hz
Winding resistance (between phases) +/- 7% at 25°C	0.77 C
Maximum Must Trip current (MMT)	35 A
Locked Rotor Amps (LRA)	175 A
Motor protection	Internal thermostat , ext. overload protector needed

In a Danfoss Performer* scroll compressor, the compression is performed by two scroll elements located in the upper part of the compressor above the motor (see figure 2.1). Suction gas enters the scroll elements where compression takes place. The center of the orbiting scroll traces a circular path around the center of the fixed scroll. This movement creates symmetrical compression pockets between the two scroll elements. Low pressure suction gas is trapped within each crescent-shaped pocket as it gets formed; continuous motion of the orbiting scroll serves to seal the pocket, which decreases in volume as the pocket moves towards the center of the scroll set increasing the gas pressure. Maximum compression is achieved once a pocket reaches the center where the discharge port is located; this stage occurs after three complete orbits. Compression is a continuous process:

when one quantity of gas is being compressed during the second orbit, another quantity is entering the scrolls and yet another is being discharged all at the same time.

2.2 Sight glass.[3]

Sight glasses are used to indicate the condition of the refrigerant in the liquid line of the plant, the flow in the oil return line from the oil separator and the moisture content in the refrigerant. They are equipped with sensitive indicators that reflect a color, depending on the moisture content in the refrigerant. The values under "green/dry" are to be considered as perfect condition meaning full protection against harmful effects from moisture. In other words, the filter drier is working perfectly. If the green color starts to fade, the color change has begun and the indicator should therefore be watched more carefully. If the color changes to yellow it is a clear signal that the capacity of the filter drier is exceeded and should be replaced as soon as possible.

Moisture content ppm = parts per million					
25 C			43 C		
Green/dry	Intermediate colour	Yellow/wet	Green/dry	Intermediate colour	Yellow/wet
<150	150-300	>300	<250	250-300	>500

Features

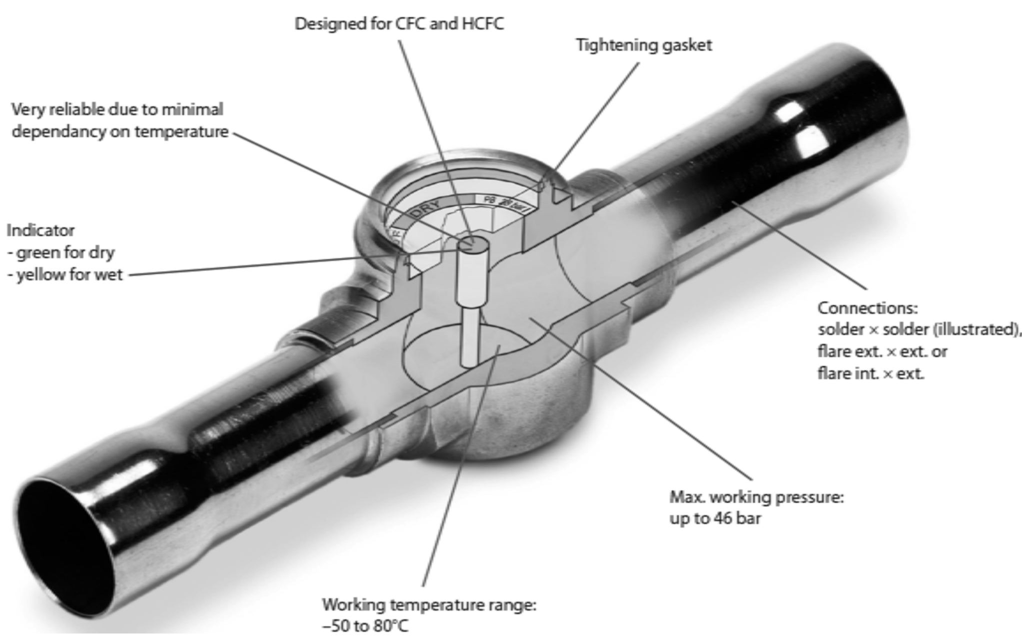


Figure 2.2 Sight glass [3]

2.3 Filter Drier [4]

Figure 2.3 Filter Drier [5]

Eliminator liquid line filter driers protect refrigeration and air-conditioning systems from moisture, acids, and solid particles. With these contaminants eliminated, systems are safer from harmful chemical reactions and from abrasive impurities.

There are two types of Eliminator cores. Eliminator type DCL, with a solid core of 80% Molecular Sieve and 20% activated alumina, is the drier of choice for systems with HCFC and CFC refrigerants and mineral or alkyl benzene oils. Type DCL driers are well suited for systems that operate at high condensing temperatures and require high drying capacity.

2.4 Thermostatic Expansion valve

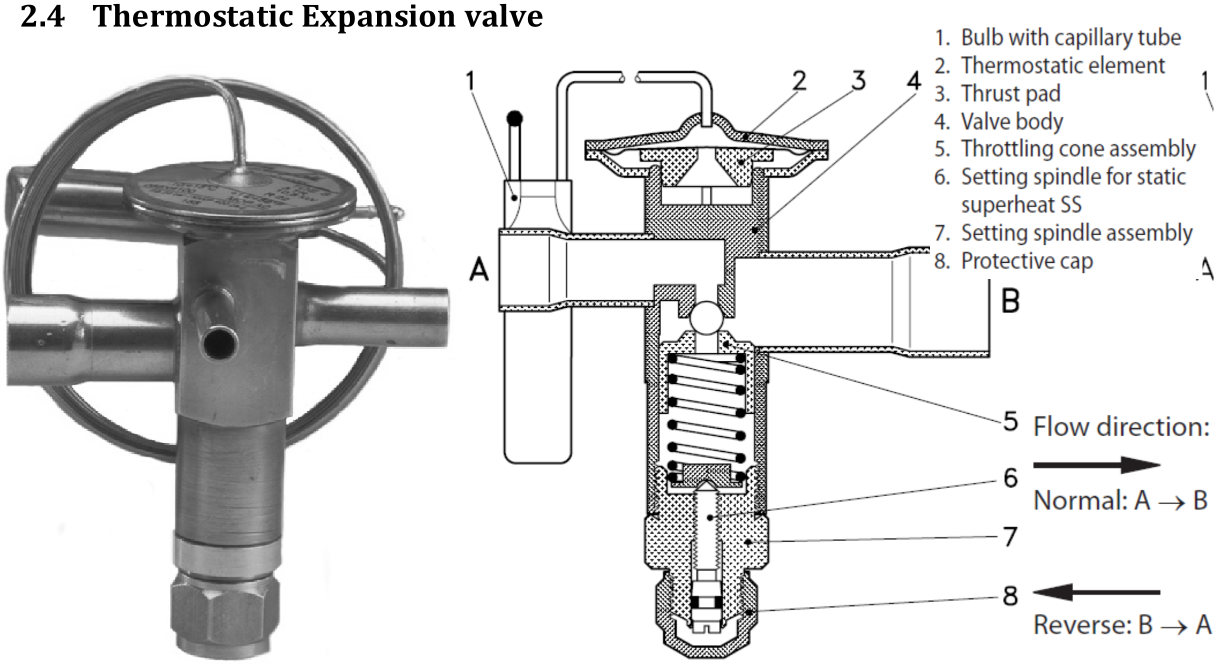


Figure 2.4 Thermostatic expansion valve[5]

A thermostatic expansion valve is a component in refrigeration and air conditioning systems that controls the amount of refrigerant flow and the superheat at the outlet of the evaporator. This is accomplished by use of a temperature sensing bulb filled with a similar gas as in the system that causes the valve to open against the spring pressure in the valve body as the temperature on the bulb increases. As temperatures in the evaporator decreases, so does the pressure in the bulb and therefore on the spring causing the valve to close.

3 Objectives

To evaluate following Parameters:

1. Air cooling capacity
2. Water cooling capacity
3. Cooling COP
4. Heating capacity
5. Heating COP

4 Theory

4.1 To determine air cooling load.

Ambient air is cooled as it is passed over evaporator coils. To measure the air cooling load we must estimate the enthalpy of air at evaporator inlet and at evaporator outlet. For measuring mass flow rate of air, we need to get its volumetric flow rate, which in turn depends upon the velocity. We take velocity measurements at different locations and take average velocity.

$$\dot{m}_a = v_{avg} A_c \tag{1}$$

From of inlet and outlet air DBT and %Rh we can get the enthalpy of air from psychometric charts.

$$\dot{Q}_a = \dot{m}_a (h_{in} - h_{out}) \tag{2}$$

4.2 To determine Water cooling load

Tap water is cooled as it is passed through evaporator. To estimate water cooling load:

$$\dot{Q}_{e.w} = \dot{m}_{e.w} C_{p.w} (t_{e.w.i} - t_{e.w.o}) \tag{3}$$

4.3 To determine Condenser Water heating load

Condenser Heat is recovered by the water as a result of which water is heated. This takes place at refrigerant saturation temperature .Part of the water from condenser outlet is taken out while rest is allowed to pass through SHR –WH where it is further heated.

$$Q_{c.w} = \dot{m}_{c.w} C_{p.a} (t_{c.w.o} - t_{c.w.i}) \tag{4}$$

4.4 To determine SHR –WH load:

Water from condenser outlet is passed through SHR –WH where heat is recovered from superheated refrigerant.

$$\dot{Q}_{ds.w} = \dot{m}_{ds.w} C_p (t_{ds.w.o} - t_{ds.w.i}) \tag{5}$$

4.5 To determine COPs of the system.

The COP gives us an idea about how good the pump is. With this ratio, we can compare two different pumps of different capacities. The total cooling COP measures the cooling effect in evaporator side .

$$Maximum\ COP_{c.t} = \frac{Total\ cooling\ capacity}{Power\ input} = \frac{\dot{Q}_{e.w} + \dot{Q}_a}{\dot{W}_{com}} \tag{6}$$

The total cooling COP is further divided into water cooling COP and air cooling COP which measures the cooling effect with respect to water and air respectively.

$$COPc.w = \frac{\dot{Q}_{e.w}}{\dot{W}_{com}} \tag{7}$$

$$COPc.a = \frac{\dot{Q}_a}{\dot{W}_{com}} \tag{8}$$

The heating COP measures the heating effect at the condenser side. Heating COP is given by

$$Maximum\ COP_h = \frac{Total\ heating\ capacity}{Power\ input} = \frac{\dot{Q}_{c.w} + \dot{Q}_{ds.w}}{\dot{W}_{com}} \tag{9}$$

5 Schematic

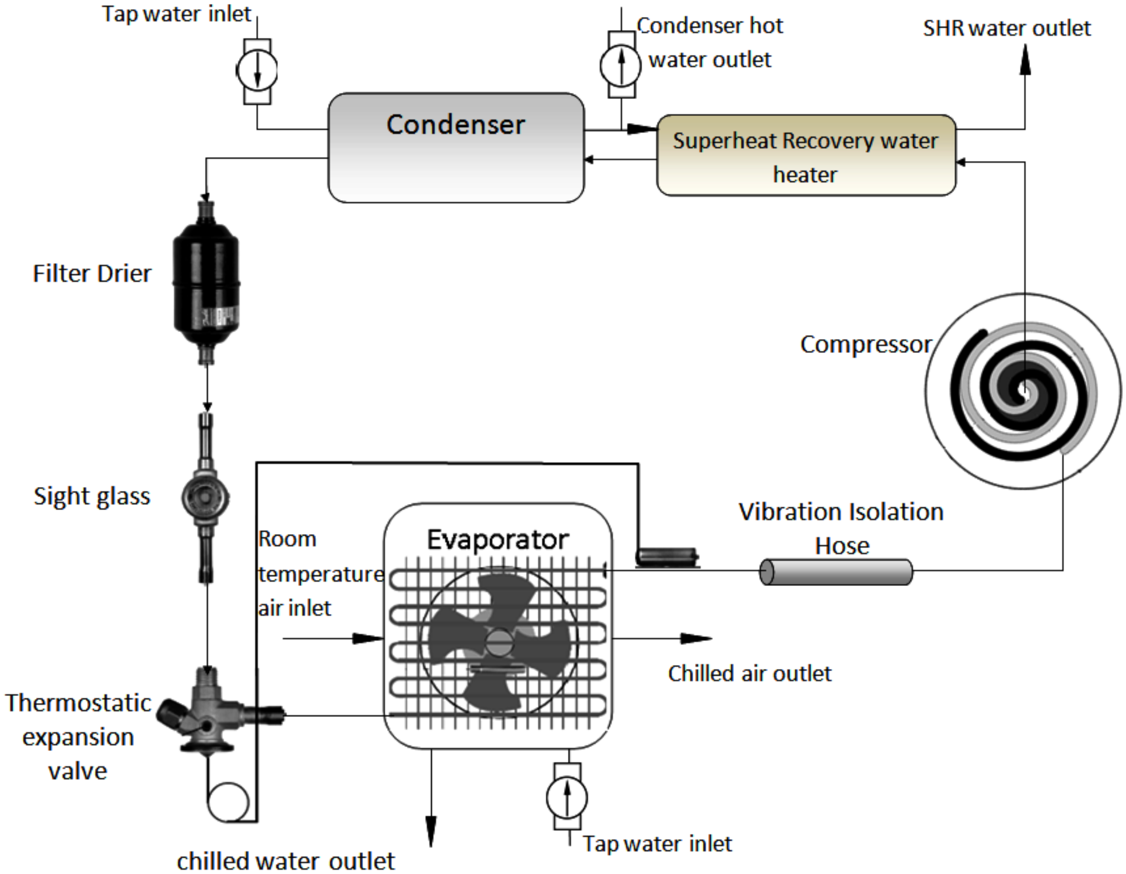


Figure 5.1 Schematic

Procedure for testing

1. Evaporator water inlet temperature and condenser water inlet conditions are kept constant.
2. Evaporator water outlet temperature is varied for given condenser water outlet temperature (hence at constant condenser pressure) by adjusting mass flow rate of evaporator of water.
3. Corresponding COP's are calculated.
4. The condenser water outlet temperature is varied keeping evaporator pressure constant by varying the mass flow rate of condenser water. step 3 is repeated.
5. Performance curves are generated for obtained values of COP as a function of evaporator water outlet temperature and Condenser water outlet temperature.

6 Sample calculations for Ideal cycle

Refrigerant R22 is the working fluid in an ideal vapor-compression refrigeration cycle that has a evaporating temperature of 1 C and condensing temperature of 60 C.As per ARI standards [1], evaporator outlet is superheated by 11.1 C and condenser outlet is sub cooled by 8.3 C. Difference between water temperature of at condenser inlet and outlet is 15 C and difference at the evaporator inlet and outlet is 5 C. Determine the heating capacity of condenser.

Known

a. $t_{e.sat}$=1 C
b. $t_{c.sat}$=60 C
c. $t_{e.r.o}$=12.1 C
d. $t_{c.r.o}$ =51.7 C
e. Δt_{con} = 15 C
f. Δt_{evap} = 5 C

To find

a. Heating capacity of condenser ($\dot{Q}_c$)
b. mass flow rate of condenser water $\dot{m}_{c.w}$
c. mass flow rate of evaporator water $\dot{m}_{e.w}$

Assumptions:

a. Each component of the cycle is analyzed as a control volume at steady state. The control volumes are indicated by dashed lines on the accompanying sketch.
b. Except for the expansion through the valve, which is a throttling process, all processes of the refrigerant are internally reversible.
c. Compressor and expansion valve operate adiabatically.
d. Kinetic and potential energy effects are negligible.
e. Heat exchanging is 100% efficient .

Analysis:

We first start with finding the quantities available in the data sheet of Danfoss manual [1]. Since the values in the tables are available at 0°C and 5°C, we interpolate to get the value at 1°C.

a. Cooling capacity = $\frac{1*39.257\text{kW} + 4*32.455\text{kW}}{5}$ = 33.815 kW
b. Power input = $\frac{1*15.216\text{kW} + 4*15.136\text{kW}}{5}$ =15.15 kW.
c. Mass flow = $\frac{1*945\text{kg/h} + 4*792\text{kg/h}}{5}$ =823 kg/h.

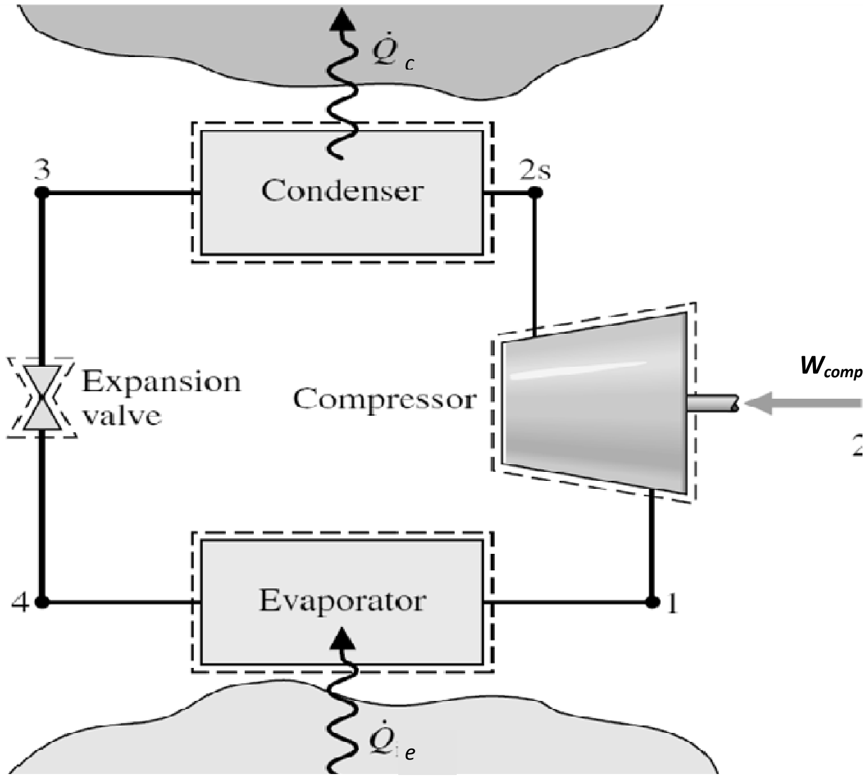
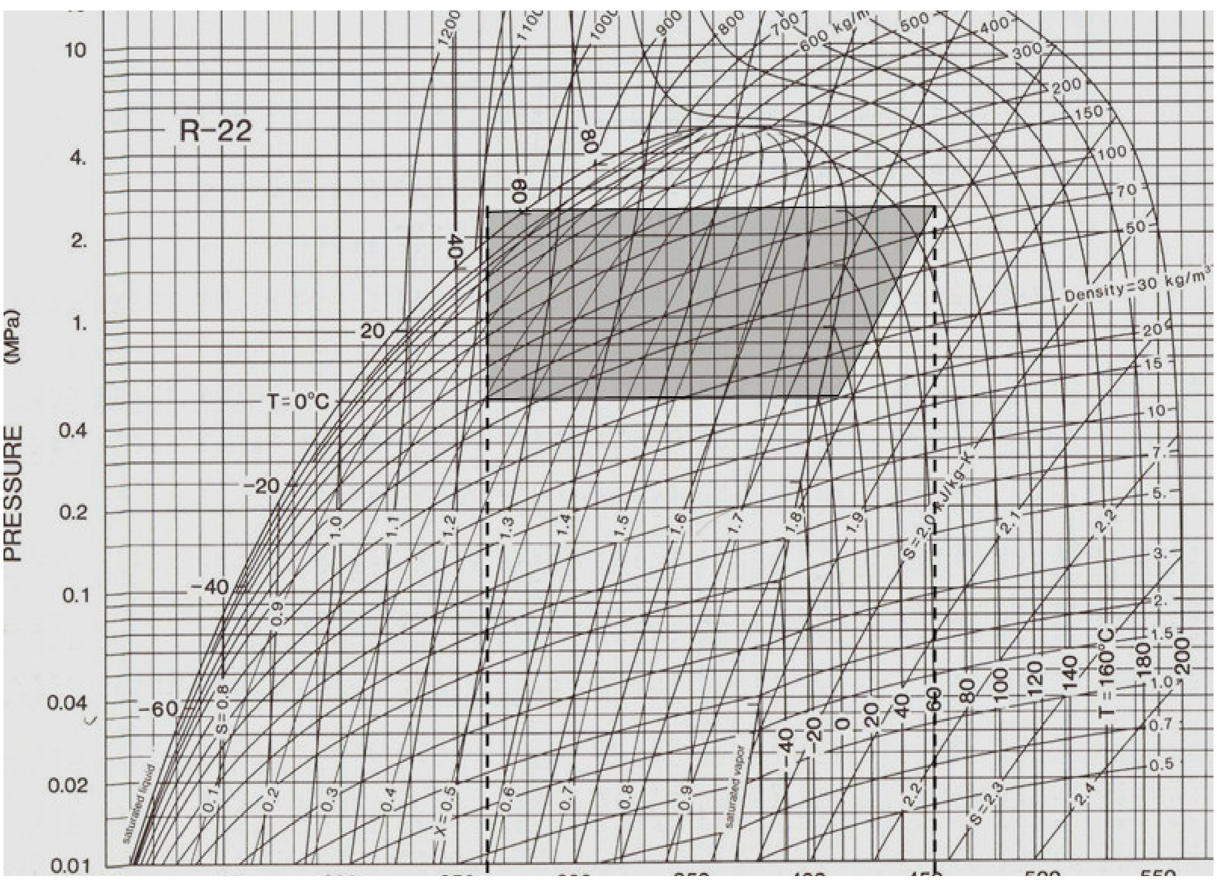


Figure 6.1 Schematic of cycle

Figure 6.2 P-h cycle for R22 [6]



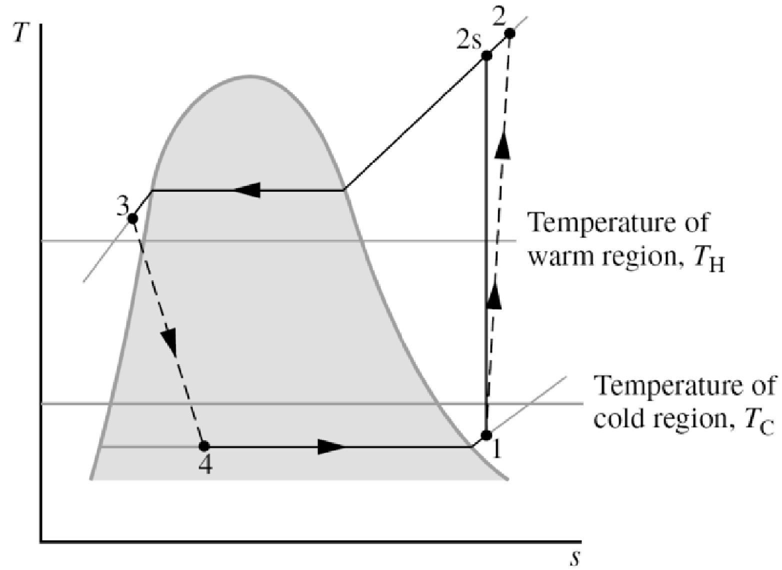


Figure 6.3 T-S diagram for cycle

From R22 p-h diagram we have,
Heating capacity = $\dot{m}_r$(h_{2s} – h_3) = (823/3600)kg/s×(455-265)kJ/kg = **43.44 kW**.

$$Maximum\ COP_{c.w} = \frac{cooling\ capacity}{Power\ input} = \frac{33.815\text{kW}}{15.152\text{kW}} = \mathbf{2.23}.$$

$$Maximum\ COP_h = \frac{heating\ capacity}{Power\ input} = \frac{43.436\text{kW}}{15.152\text{kW}} = \mathbf{2.87}.$$

Calculation of mass flow rate of condenser water(assuming 100% efficiency).

$\dot{m}_{c.w}C_{p.w}\Delta t_{con}$ = Heating capacity, putting Δt_{con} as 15 °C

$\dot{m}_{c.w} = \frac{43.436\text{kW}}{\frac{4.18\text{kJ}}{\text{kg K}}\times 15\text{K}}$ = **0.693 kg/s** = .0115 kg/min =1.925×10^{-4} kg/h

Calculation of mass flow rate of evaporator water(assuming 100%heat transferred to water).

$\dot{m}_{e.w}C_{p.w}\Delta t_{evap}$ = Cooling capacity, putting Δt_{evap} as 5 °C.

$\dot{m}_{e.w} = \frac{33.815}{4.18*5}$ = **1.62 kg/s** = 0.0195 kg/min =3.25×10^{-4} kg/h

Thus $\dot{m}_{e.w}$ will vary linearly with percentage heat transferred to water as opposed to air, from 0 to 0.693 kg/s.

7 Calculation of mass flow rate of air

7.1 Procedure

1. Air velocity was measured at different locations at evaporator inlet using anemometer.
2. Average of the readings was taken to calculate the mass flow rate

$$\dot{m}_a = \rho_a 2Av_{avg} = 1.2\frac{kg}{m^3} \times 2 \times 1.16m \times .69m \times 1.4m/s = 2.7kg/s$$

Air velocity measurement at area 1

1.8		1.7		1.8
	1.4		1.6	
1.2		1.5		1.3
	1.3		1.3	
1.1		1.5		1.1

Air velocity measurement at area 2

1.6		1.9		1.8
	1.5		1.7	
1		1.4		1.3
	1.2		1.6	
1.1		1.3		1.1

Figure 7.1 Air velocity measurement

8 Calibration of temperature

1. We took a standard thermometer and an ice bath.
2. We increased its temperature by 5 C by adding hot water in it and stirred it thoroughly.
3. We then noted down the reading on the thermometer and the corresponding reading on the temperature indicator starting from the ice bath.
4. We plotted the graph of indicator temperature vs. standard thermometer.
5. The temperature indicator showed on average 1.5 C less than standard thermometer.

standard thermometer(C)	temperature indicator (C)
0	-1.4
5	3.5
10	8.6
15	13.4
20	18.5
25	23.5
30	28.6
35	33.6
40	38.5
45	43.5
50	48.6
55	53.5
60	58.5

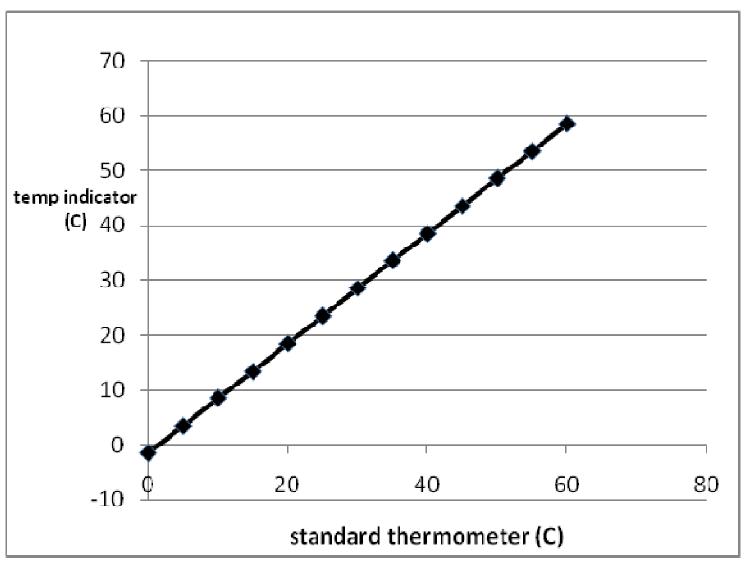


Figure 8.1 standard temperature vs. indicator temperature

9 Set 1

9.1 Aim

1. To study the effect of condenser pressure which in turn depend upon mass flow rate of condenser cooling water on COP_h and COP_c.
2. 2. To find maximum temperature of water out of sup heat recovery water heater.

9.2 Procedure

1. Start the condenser cooling water flow by opening a valve.
2. Put on the heat pump by switching on controller unit.
3. Allow the maximum water flow.
4. Allow all flow of condenser cooling water to pass through Superheat recovery water heater.
5. Take the readings in the time interval of each 5 minutes.
6. After 3 consecutive readings of all parameters remain same, steady state is reached.
7. Take the readings for $t_{c.w.i}$,$t_{ds.w.o}$,$t_{e.a.i}$,%Rh_{in},$t_{e.a.o}$,%Rh_{out}, P_e,P_c,Power input ,air velocity, water flow rate.
8. Now decrease the water flow rate and follow steps 4 to 6.
9. Repeat the procedure for 5 different water flow rates.
10. Ensure that water flow rate MUST NOT be reduced to such a level that condenser pressure (Gauge) exceeds 24 bars. In this case compressor trips.
11. Plot the variation of COP_h COP_c vs $t_{c.sat}$ .Also variation of COP_h vs. Condenser cooling water mass flow rate.
12. comment on the results obtained.

9.3 Observation Tables

		water conditions					
Sr.no.	Power(kW)	$t_{c.w.i}$(˚C)	$t_{ds.w.o}$(˚C)	$\dot{m}_{c.w}$(kg/s)	$t_{c.sat}$(˚C)	Q_{cond}(kW)	COP_h
1	14.99	29.1	46.1	0.68	48.98	48.60	3.24
2	15.07	29.0	48.6	0.53	51.28	43.39	2.88
3	15.58	29.0	51.9	0.43	54.54	41.12	2.64
4	16.37	28.8	54.3	0.39	57.21	41.60	2.54
5	16.78	29.1	58.8	0.34	58.44	41.88	2.50

	Air conditions								
Sr.no.	$t_{e.a.i}$(˚C)	%Rh_{in}	$h_{e\cdot a.i}$	$t_{e.a.o}$(˚C)	%Rh_{out} %	$h_{e.a.o}$	$t_{e.sat}$(˚C)	Q_e(kW)	$COP_{c.a}$
1	23.0	66.0	50	13.9	72.0	31.83	-1.14	47.18	3.15
2	20.6	63.0	46.56	13.5	74.4	31.57	-1.77	38.92	2.58
3	19.5	64.0	43.2	12.7	75.4	30.04	-1.77	34.17	2.19
4	18.7	66.5	43	12.8	74.9	30	-1.77	33.76	2.06
5	19.4	65.5	42.71	12.8	75.1	30	-1.14	33.00	1.97

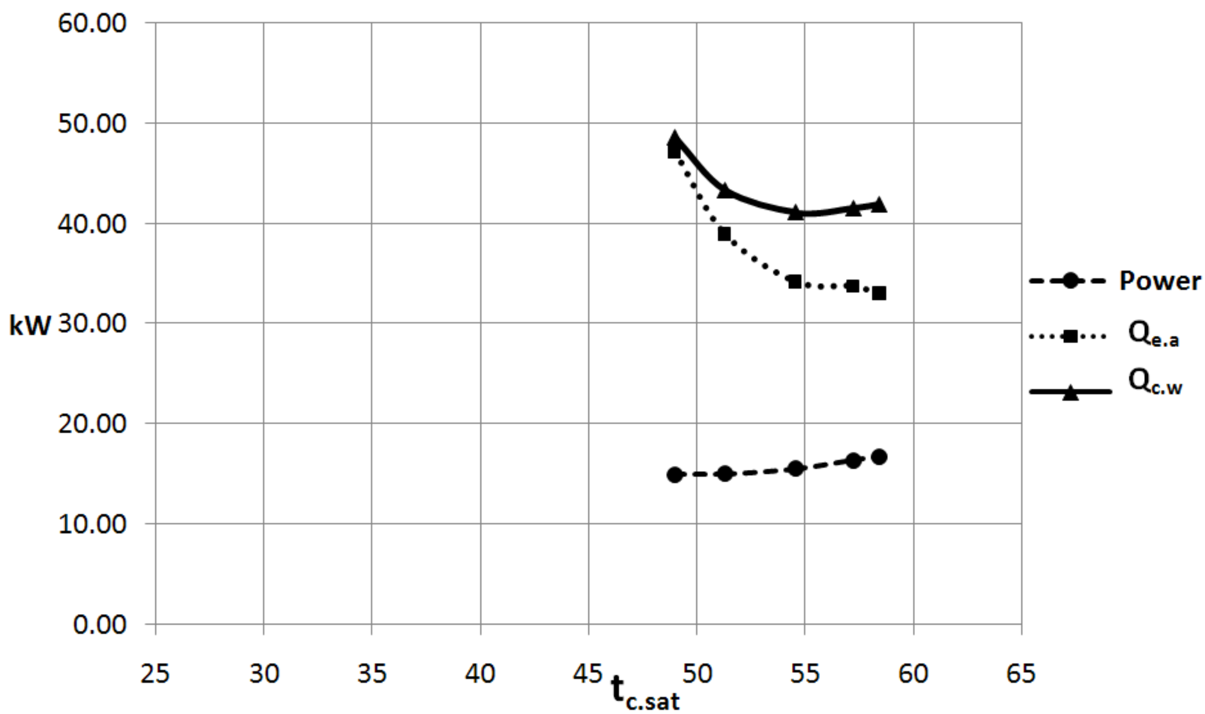
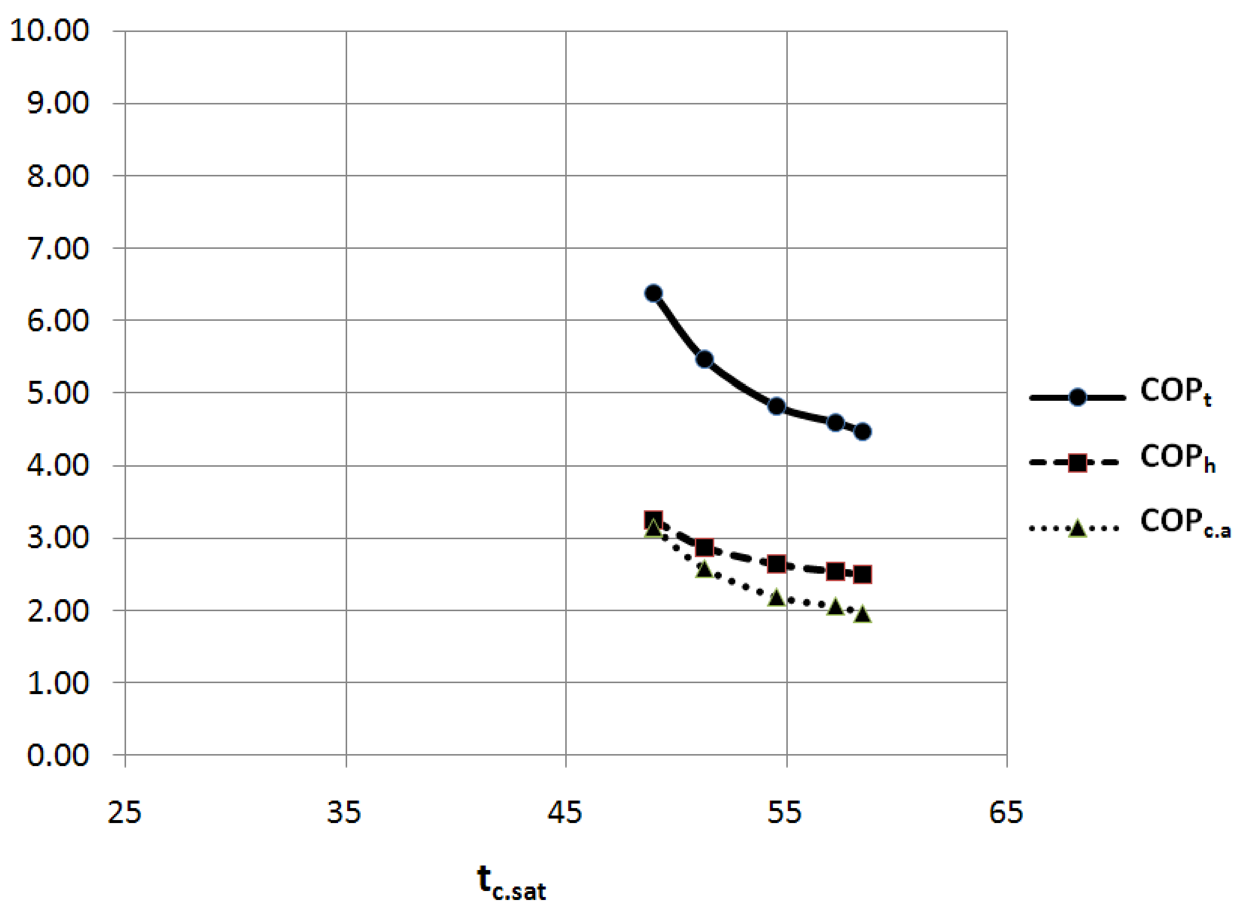

60.00
50.00
40.00
kW 30.00
20.00
10.00
0.00
25
30
35
40
45
50
55
60
65
$t_{c.sat}$
Power
$Q_{e.a}$
$Q_{c.w}$

Figure 9.1


10.00
9.00
8.00
7.00
6.00
5.00
4.00
3.00
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25
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45
55
65
$t_{c.sat}$
COP_t
COP_h
$COP_{c.a}$

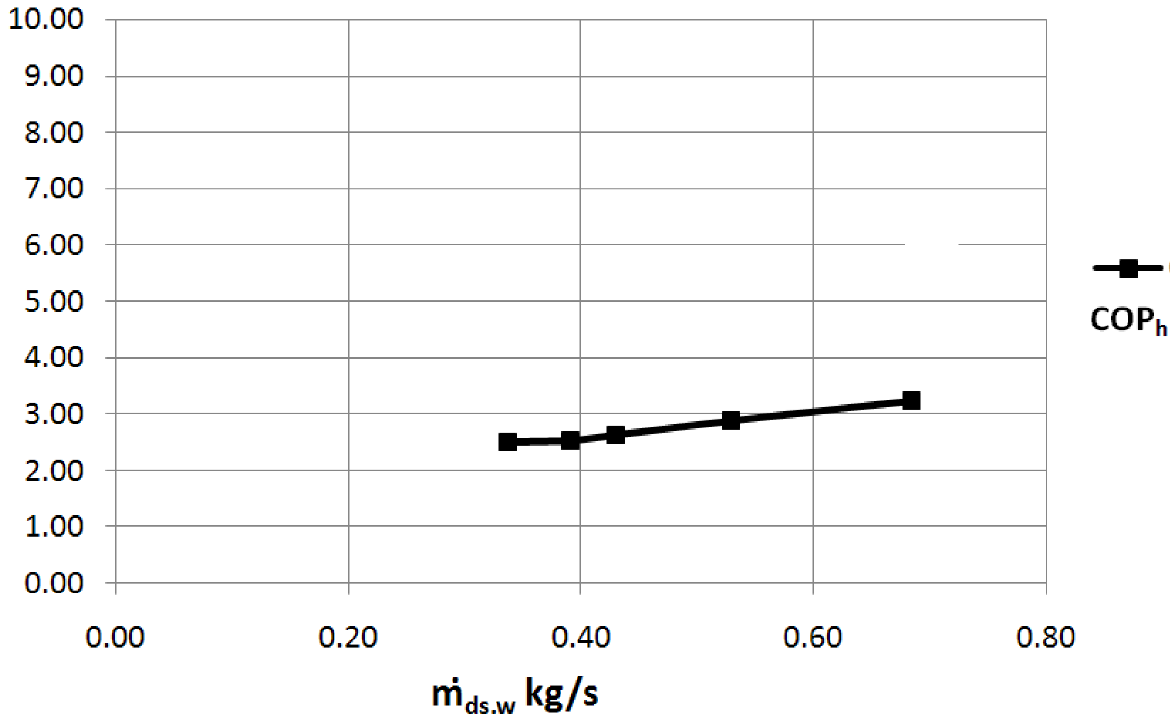


Figure 9.3

1. COP_h and COP_c decreases with increase in Condenser saturation temperature.
2. Power input increases with increase in Condenser saturation temperature.
3. COP_h increases with $\dot{m}_{ds.w}$

9.5 Conclusions

1. COP decreases with increase in condenser saturation temperature. This can be explained as below. As condenser temperature increases for same evaporator temperature, Power input to compressor increases as well as cooling capacity decreases.
2. Hence, the discharge pressure should be kept as low as possible depending upon the temperature of the cooling medium available.

10 Set 2

10.1 Aim

1. to study the effect of condenser pressure which in turn depend upon mass flow rate of condenser cooling water on COP_h and COP_c
2. To find maximum temperature of water out of condenser.

10.2 Procedure

1. Start the condenser cooling water flow by opening a valve.
2. Put on the heat pump by switching on controller unit.
3. Allow the maximum water flow.
4. Don't allow flow of condenser cooling water to pass through Superheat recovery water heater. All flow is taken out from condenser.
5. Take the readings in the time interval of each 5 minutes.
6. After 3 consecutive readings of all parameters remain same, steady state is reached.
7. Take the readings for $t_{c.w.i}$,$t_{c.w.o}$,$t_{e.a.i}$,%Rh_{in},$t_{e.a.o}$,%Rh_{out}, P_e,P_c,Power input ,air velocity, water flow rate.
8. Now decrease the water flow rate and follow steps 4 to 6.
9. Repeat the procedure for 5 different water flow rates.
10. Ensure that water flow rate MUST NOT be reduced to such a level that condenser pressure (Gauge) exceeds 24 bars. In this case compressor trips.
11. Plot the variation of COP_h, COP_c vs. $t_{c.sat}$ .Also variation of COP_h vs. Condenser cooling water mass flow rate.
12. comment on the results obtained.

10.3 Observations table

	Condenser water conditions						
sr no	$t_{c.w.i}$(˚C)	$t_{c.w.o}$(˚C)	$t_{c.sat}$(˚C)	$\dot{m}_{c.w}$(kg/s)	$Q_{c.w}$(kW)	COP_h	COP_c + COP_h
1	27.5	45	54.5	0.69	50.80	3.2	6.8
2	27.5	47.1	56.6	0.59	48.42	2.9	6.0
3	27.5	49.2	59.0	0.52	47.00	2.8	5.4
4	27.5	49.8	60.0	0.51	47.56	2.8	5.1

	Evaporator air conditions								
sr no	$t_{e.a.i}$(˚C)	%Rh_{in}	$h_{e.a.i}$	$t_{e.a.o}$	%Rh_{out}	$h_{e.a.o}$(kJ/kg)	$t_{e.sat}$(˚C)	$Q_{e.w}$(kW)	COP_c	Power(KW)
1	21.5	63.2	47.18	11.6	76.0	27.83	-1.14	57.42	3.6	16.00
2	20.5	63.2	43.76	11.4	73.5	26.88	-1.77	50.09	3.0	16.43
3	21.1	69.0	43.11	11.9	76.1	28.49	-1.77	43.39	2.6	16.75
4	21.2	68.0	43	12.1	76.5	29.22	-1.14	40.89	2.4	17.23

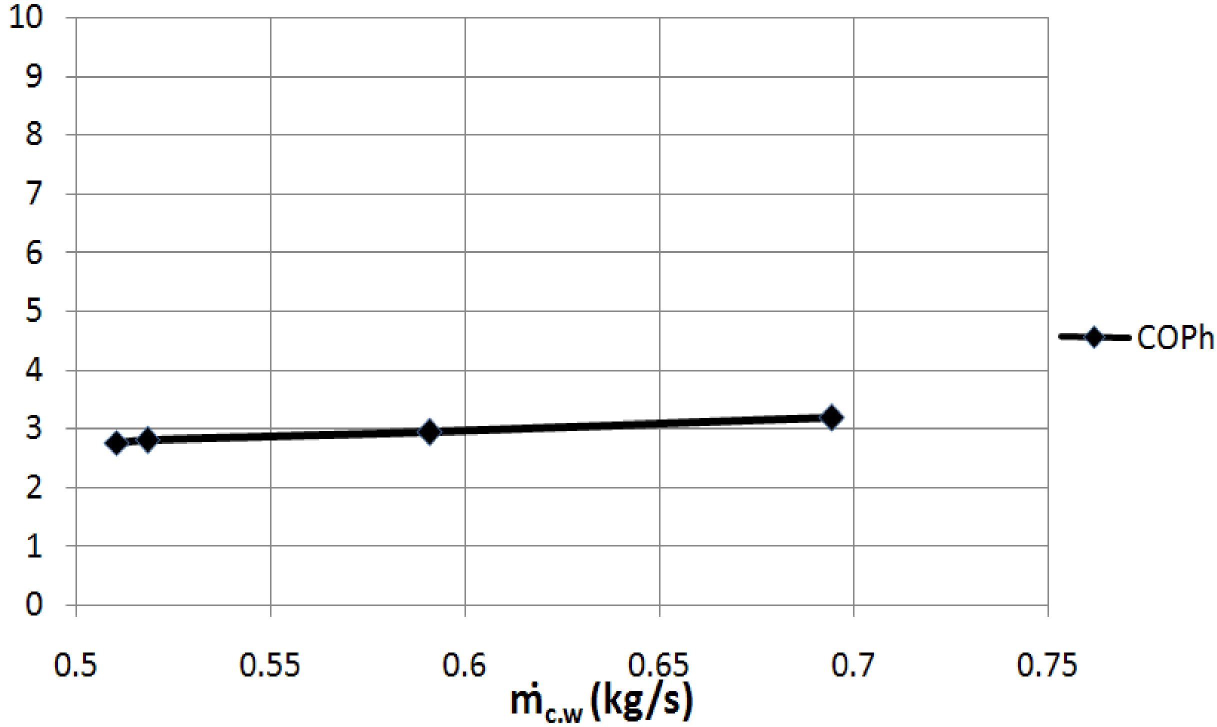


Figure 10.1 COPh vs. condenser water flow rate

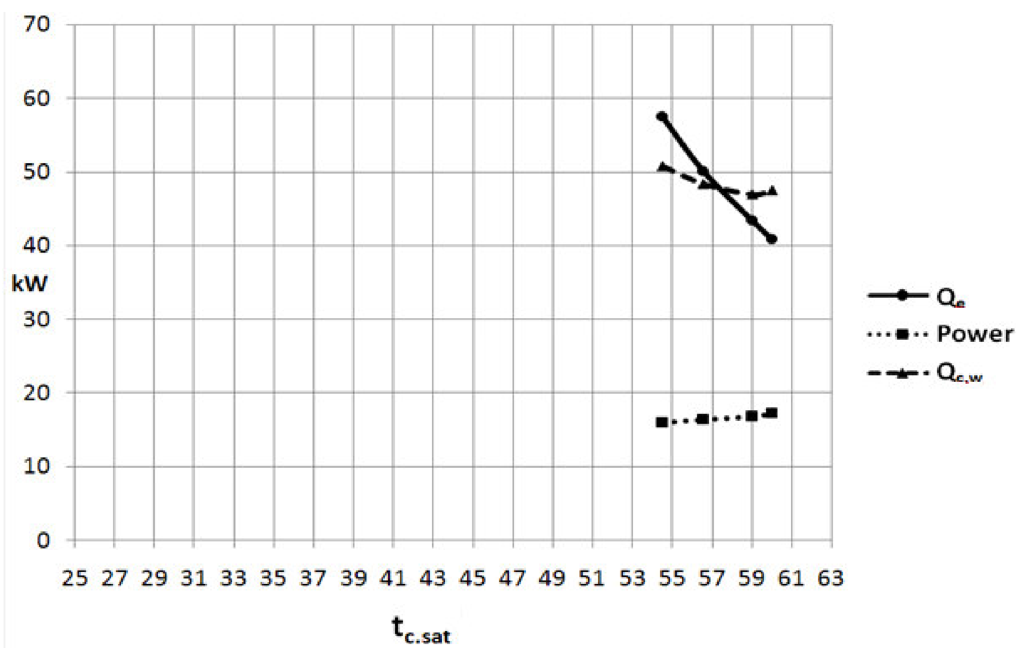


Figure 10.2

10.4 Observations

1. COP_h and COP_c decreases with increase in Condenser saturation temperature.
2. Power input increases with increase in Condenser saturation temperature.
3. COP_h increases with m_{cw}

10.5 Conclusion

1. COP decreases with increase in condenser saturation temperature. This can be explained as below. As condenser temperature increases for same evaporator temperature, Power input to compressor increases as well as cooling capacity decreases N
2. Note that according to energy balance, Condenser heat duty should be equal to evaporator heat duty plus compressor power input .But from observation chart this is nor so. This phenomenon can be explained on the basis of Loss of heat to air.
3. Hence, the discharge pressure should be kept as low as possible depending upon the temperature of the cooling medium available

11 Set 3

11.1 Aim:

to study the effect of mass flow rate of evaporator water on COP.

11.2 Procedure

1. Start the condenser cooling water flow and evaporator water flow by opening a valve.
2. Put on the heat pump by switching on controller unit.
3. Adjust the condenser cooling water flow to such a value that $t_{ds.w.o}$ is about 54 C
4. Note that all condenser water is passed through superheat recovery water heater.
5. Allow the steady state to be reached.
6. Note down $t_{e.w.i}$, $t_{e.w.o}$, $t_{e.a.i}$, $t_{e.a.o}$,%Rh_{in},%Rh_{out},P_e,P_c,Power consumed
7. Gradually increase the mass flow rate of water through evaporator keeping condenser cooling water flow rate constant.

11.3 Observations table

	Condenser water conditions						
Sr no.	$t_{c.w.i}$(˚C)	$t_{ds.w.o}$(˚C)	$\dot{m}_{c.w}$(kg/s)	$Q_{c.w}$(kW)	COP_h	Power	COP_t
1	27.3	54.6	0.45	50.81	3.0	16.99	6.0
2	27.3	54.5	0.45	50.85	3.0	16.84	6.0
3	27.4	54.3	0.45	50.20	3.0	16.71	5.9
4	27.6	54.7	0.45	50.57	3.1	16.45	6.1

	Evaporator air conditions							
Sr no.	$t_{e.a.i}$(˚C)	$\%Rh_{in}$	$h_{e.a.i}$	$t_{e.a.o}$(˚C)	$\%Rh_{out}$	$h_{e.a.o}$	$Q_{e.a}$(kW)	COPc air
1	22.6	56.0	46.93	13.3	72.1	30.55	42.53	2.5
2	22	59.0	46.71	13.5	74.0	31.45	39.62	2.4
3	21.3	61.0	45.76	14	73.0	32.3	34.95	2.1
4	21.6	63.0	47.36	14.9	74.0	34.6	33.13	2.0

	Evaporator water conditions					Evaporator conditions	
Sr no.	$t_{e.w.i}$(˚C)	$t_{e.w.o}$(˚C)	$\dot{m}_{c.w}$(kg/s)	$Q_{e.w}$(kW)	$COP_{c.w}$	Q_e(kW)	$COP_{c.t}$
1	27.3	14	0.17	9.27	0.5	51.80	3.0
2	27.3	15.4	0.21	10.49	0.6	50.12	3.0
3	27.4	15.6	0.27	13.12	0.8	48.07	2.9
4	27.6	16.2	0.33	15.88	1.0	49.02	3.0

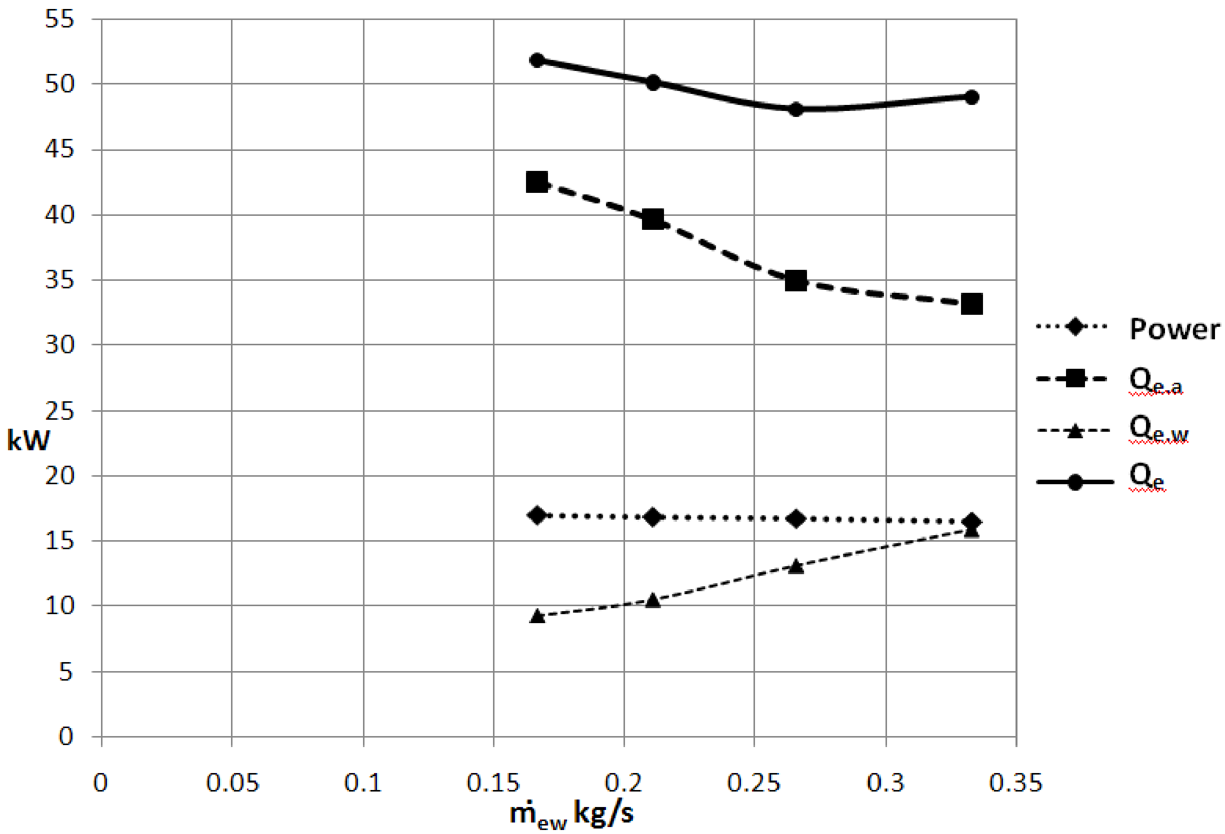
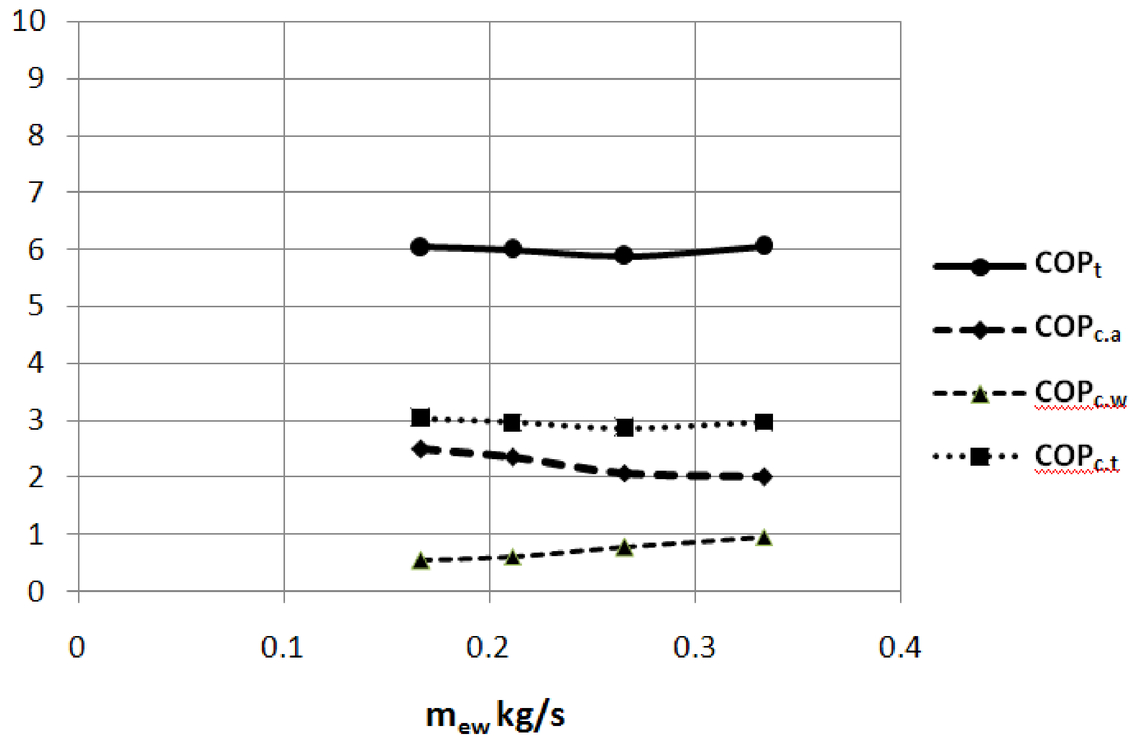


Figure 11.1



11.4 Observation

1. Pressure of the evaporator remains fairly constant due to comparatively small mass flow rate of water.
2. Evaporator heat duty is nearly constant.
3. As mass flow rate of evaporator water is increased, it takes more and more part of evaporator heat duty, and Power input remains constant hence COP for chilled water increases with its mass flow rate.
4. Air outlet temperature at evaporator increases with mass flow rate of chilled water.
5. Total COP is remaining constant.

11.5 Conclusions

1. In this set evaporator and condenser temperature are remaining fairly constant. As a result of which total COP remains almost constant.
2. Power input is a function of saturation temperatures of evaporator and condenser. As these conditions are not changing to large extent, power input is constant.
3. COP for chilled water increase because for a constant power input, chilling capacity increases as water flow rate at evaporator increases.
4. Air outlet temperature at evaporator increases with increasing mass flow rate of evaporator water because heat duty of evaporator to air decreases.
5. If large flow rates of water are available then, evaporator pressure will increase resulting in increase in COP.

12 Error analyses.

In general ,for water cooling and heating load,

$$\dot{Q}_w = \rho \dot{V} C_{p.w} \Delta t$$

$$\dot{V} = \frac{Volume\ through\ flow\ metre}{time}$$

$$\frac{\Delta(\dot{Q}_w)}{\dot{Q}_w} = \frac{\Delta(\rho)}{\rho} + \frac{\Delta(\dot{V})}{\dot{V}} + \frac{\Delta(C_{p.w})}{C_{p.w}} + \frac{2 \times \text{L. C of sensor}}{\Delta t} \tag{10}$$

$$\frac{\Delta(\dot{V})}{\dot{V}} = \frac{\Delta(Volume)}{Volume} + \frac{\Delta(time)}{time} \tag{11}$$

For air cooling load,

$$\dot{Q}_a = \rho_a v_{avg} A_c \Delta h$$

$$\frac{\Delta(\dot{Q}_a)}{\dot{Q}_a} = \frac{\Delta(\rho)}{\rho} + \frac{\Delta(v_{avg})}{v_{avg}} + \frac{\Delta(l)}{l} + \frac{\Delta(w)}{w} + \frac{\Delta(\Delta h)}{\Delta h} \tag{12}$$

The error in Δh has to be estimated using the error in temperature and %Rh measurements and psychometric charts.

For measuring the error in COP, we need the errors in the heat transfer and the compressor work input .Errors in heat transfer are already obtained. Errors in compressor can be obtained using the L.C of the power meter.

For heating COP,

$$\frac{\Delta(\dot{W}_{com})}{\dot{W}_{com}} = \frac{\text{L. C of power meter}}{\dot{W}_{com}}$$

$$COP_h = \frac{\dot{Q}_{c.w} + \dot{Q}_{ds.w}}{\dot{W}_{com}}$$

$$\frac{\Delta(COP_h)}{COP_h} = \frac{\Delta(\dot{Q}_{c.w}) + \Delta(\dot{Q}_{ds.w})}{\dot{Q}_{c.w} + \dot{Q}_{ds.w}} + \frac{\Delta(\dot{W}_{com})}{\dot{W}_{com}} \tag{13}$$

Similarly for cooling COP,

$$\frac{\Delta(COP_{c.t})}{COP_{c.t}} = \frac{\Delta\left(\dot{Q}_{e.w}\right) + \Delta(\dot{Q}_a)}{\dot{Q}_{e.w} + \dot{Q}_a} + \frac{\Delta(\dot{W}_{com})}{\dot{W}_{com}} \tag{14}$$

Putting values and solving we get the formula

$$\frac{\Delta(COP_h)}{COP_h} = \frac{.01}{time_w} + \frac{0.2}{t_{c.w.o} - t_{c.w.i}} + \frac{.01}{time_p}$$

13 Sample calculations

t_w = 22.46 sec

$t_{c.w.o}$ - $t_{c.w.i}$ = 54.6 C – 27.3 C = 27.3 C

tp = 14.69 sec

hence, % error in COP_h

$$= \frac{.01}{22.36} + \frac{0.2}{27.3} + \frac{.01}{14.69}$$

=0.85%

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[1] Trip conditions of compressor

[2] When to run defrost

[3] Type of HX used?

[4] What is refrigerant used boiling and freezing point

[5] Advantages of heat pump over geyser +AC + chiller.

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[8] Difference between EN and ARI standards.

[9] Scrol compresor