78064

EFFICIENT CENTRAL AIR-CONDITIONING DESIGN, OPERATION AND MAINTENANCE

Organised by: BCA ACADEMY
of the built environment

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Efficient Central Air-Conditioning Design & Operation

Day 1		
s/n	Lesson	Duration (Hrs)
1	**Fundamentals of air conditioning**	
a	Basic cooling load calculation	1
b	Basic psychrometry	1
c	Basic vapour refrigeration cycle	1
2	**Central chilled water plants**	
a	Overview of chilled water systems	0.75
b	Introduction to chillers	0.75
c	Introduction to pumps	0.75
d	Introduction to cooling towers	0.75
3	**Chilled water airside systems**	
a	Overview of airside systems	1
Day 2		
s/n	Lesson	Duration
4	**Energy efficient water & air distribution systems**	
a	Energy efficient piping design	1
b	Energy efficient ducting design	1
5	**Chiller plant performance measurement & verification**	0.5
6	**Chiller plant performance optimization**	1.5
7	**Airside optimization**	1
8	**Case studies**	2

Cooling Load Calculation

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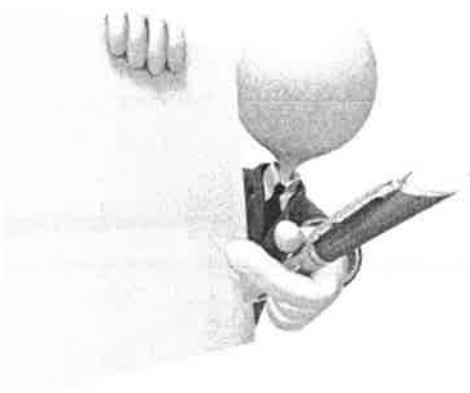
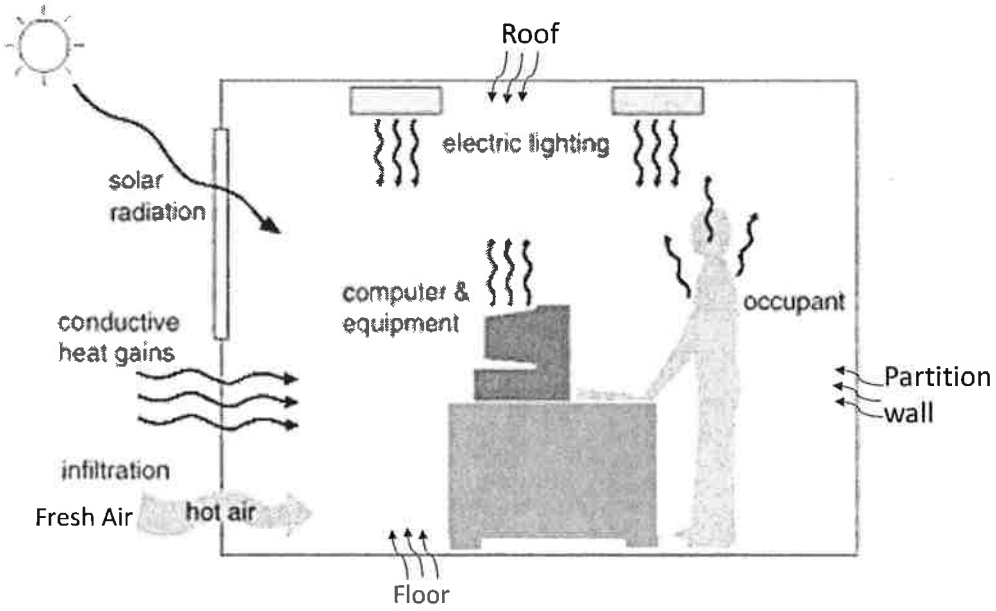
Agenda

1. Components of cooling load
2. Sensible and latent cooling load
3. Basic principles of cooling load calculation
4. Cooling load component breakdown
5. Cooling load profile
6. Cooling load benchmarking data
7. Softwares for cooling load calculation



Components of Cooling Loads

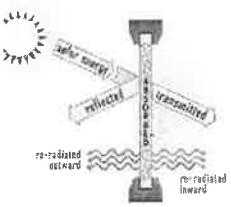


Sensible and Latent Cooling Loads

Sensible cooling loads

1. Dry heat
2. Increase dry bulb temperature only

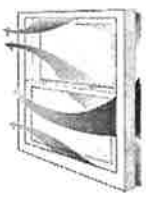


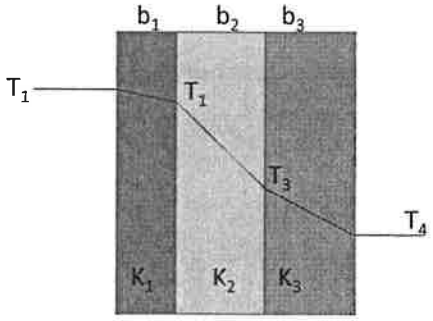
Latent cooling loads

1. Moisture to space which needs to be removed
2. Cooling load is manifested when the moisture condense on the cooling coil





Conduction load (Thro' internal wall / floor and external glass)



Q = U x A x ΔT

Where

U: Thermal transmittance ($W/m^2.K$)

ΔT: Delta Temperature (C)

A: Area

$U = 1 / R_t$

$R_t = R_o + b_1/k_1 + ... Ra + ...b_n/k_n + R_i$

Where

Rt: Total thermal resistance (m^2K/W)

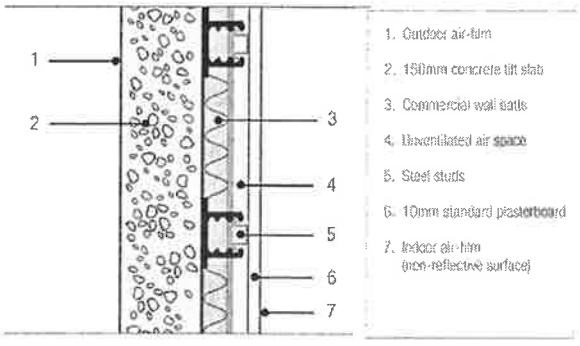
R_o & R_i: Outside and inside air film resistance

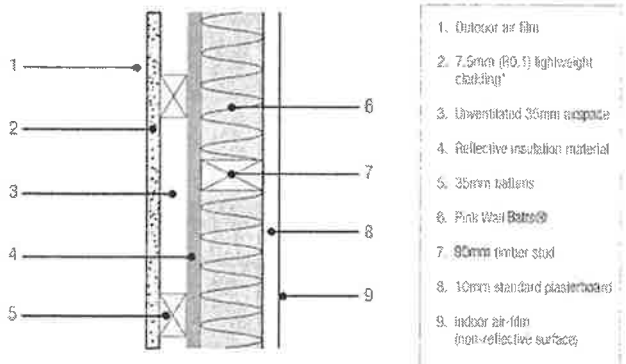
Ra: Thermal resistance of air space

k: Thermal conductivity (W/m.K)

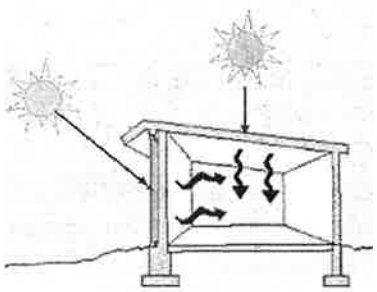
b: Thickness (m)

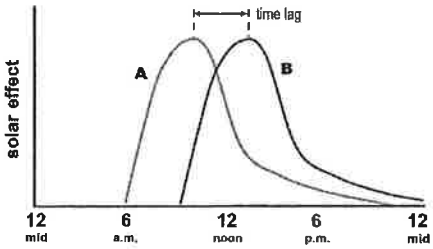
Example of external wall components





Sun lit walls





Curve **A** shows the magnitude of the solar effect on the exterior wall.

Curve **B** shows the resulting heat that is transferred through the wall into the space.

$$Q = U \times A \times CLTD$$

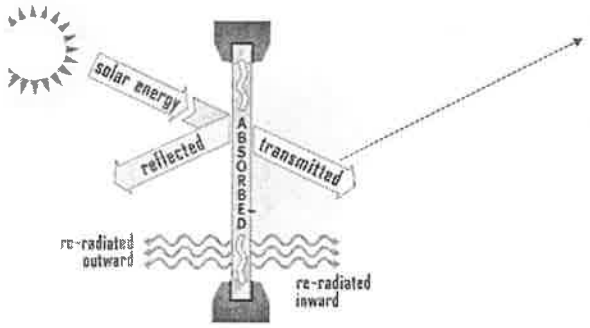
A factor called the cooling load temperature difference (CLTD) is used to account for the added heat transfer due to the sun shining on exterior walls, roofs, and windows, and the capacity of the wall and roof to store heat. The CLTD is substituted for ΔT in the equation to estimate heat transfer by conduction.

Example of CLTD from ASHRAE Fundamental Handbook

Table 32 July Cooling Load Temperature Differences for Calculating Cooling Load from Sunlit Walls 40°North Latitude

Wall Face	Wall Number 1 Hour 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
N	1	0	−1	−1	−2	−1	4	6	6	7	9	12	14	15	16	16	16	16	15	9	6	4	3	2
NE	1	0	−1	−1	−2	1	13	23	26	24	19	16	15	16	16	16	15	13	11	8	6	4	3	2
E	1	0	−1	−1	−1	1	16	28	34	36	33	27	20	17	17	17	16	14	11	8	6	4	3	2
SE	1	0	−1	−1	−2	0	8	18	26	31	32	31	27	22	18	17	16	14	11	8	6	4	3	2
S	1	0	−1	−1	−2	−1	0	2	6	12	18	24	28	29	28	24	19	15	11	8	6	4	3	2
SW	1	0	−1	−1	−1	−1	0	2	4	7	9	14	22	29	36	39	38	34	25	13	7	4	3	2
W	1	1	−1	−1	−1	−1	1	2	4	7	9	12	15	23	33	41	44	44	34	18	9	5	3	2
NW	1	0	−1	−1	−1	−1	0	2	4	7	9	12	14	16	21	28	34	36	31	16	8	5	3	2
Wall Face	**Wall Number 2 Hour 1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**
N	3	2	1	0	−1	−1	−1	2	4	5	6	8	10	12	13	14	15	16	16	15	12	9	7	4
NE	3	2	1	0	−1	−1	1	7	14	20	22	21	18	17	16	16	16	16	14	13	10	8	6	4
E	3	2	1	0	−1	−1	1	8	18	26	31	32	29	24	21	19	18	17	15	13	11	8	6	4
SE	3	2	1	0	−1	−1	0	4	11	18	24	28	29	28	25	22	19	17	16	13	11	8	6	4
S	3	2	1	0	−1	−1	−1	−1	1	4	8	13	18	23	26	27	26	22	18	15	12	8	6	4
SW	4	2	1	1	0	−1	−1	0	1	3	5	7	11	17	23	29	34	36	34	29	22	15	9	6
W	4	3	2	1	0	−1	−1	0	1	3	5	7	9	13	18	26	33	38	41	37	28	19	12	8
NW	4	2	1	1	−1	−1	−1	−1	1	3	5	7	9	12	14	18	23	28	32	30	23	16	11	7

Solar Radiation Thro' Glass



$Q = SF \times CF \times A \times SC_1 \times SC_2$

Where
SF: Solar factor
CF: Solar correction factor
(Depends on direction and wall pitch angle)
A: Window area
SC1: Shading coeff of glass
SC2: Shading coeff of external shading devices



Prof SK Chou

ETTV methodology
SF for Singapore has been experimentally determined to be 211 W/m²

ETTV

3.1 The ETTV formula is given as follows:

$$ETTV = 12(1-WWR)U_w + 3.4(WWR)U_f + 211(WWR)(CF)(SC)$$

where

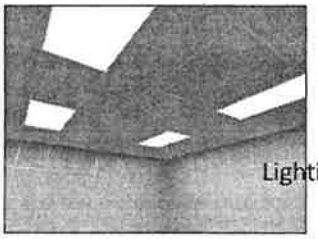
ETTV	:	envelope thermal transfer value (W/m^2)
WWR	:	window-to-wall ratio (fenestration area / gross area of exterior wall)
U_w	:	thermal transmittance of opaque wall ($W/m^2{}^oK$)
U_f	:	thermal transmittance of fenestration ($W/m^2{}^oK$)
CF	:	correction factor for solar heat gain through fenestration
SC	:	shading coefficients of fenestration

3.2 Where more than one type of material and/or fenestration is used, the respective term or terms shall be expanded into sub-elements as shown:

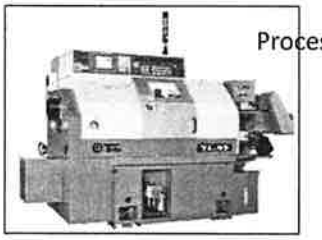
$$ETTV = \frac{12(A_{w1}\times U_{w1} + A_{w2}\times U_{w2} + \ldots + A_{wn}\times U_{wn})}{A_o} + \frac{3.4(A_{f1}\times U_{f1} + A_{f2}\times U_{f2} + \ldots + A_{fn}\times U_{fn})}{A_o} + \frac{211(A_{f1}\times SC_{f1} + A_{f2}\times SC_{f2} + \ldots + A_{fn}\times SC_{fn})(CF)}{A_o}$$

where

Internal Cooling Loads

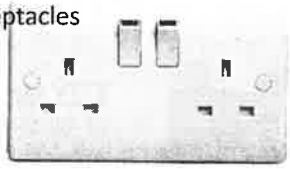


Lighting



Process

Receptacles





People

Internal loads guidelines

2. For receptacle loads, Table A below is for reference.

Table A: Receptacle Loads	Standard	Nominal Values
a. Computer intensive offices	Source:-	22.0 W/m^2
b. General office areas	ASHRAE STD	16.0 W/m^2
c. Large conference areas	90.1:2010	11.0 W/m^2
d. Server/Computer rooms		540.0 W/m^2

Source: http://www.bca.gov.sg/Envsuslegislation/others/GM_Certification_Std2012.pdf

Table 3 Rates of Heat Gain from Occupants of Conditioned Spaces

Degree of Activity		Total Heat, W: Adult Male	Total Heat, W: Adjusted, M/F[b]	Sensible Heat, W	Latent Heat, W
Seated at theater	Theater, matinee	115	95	65	30
Seated at theater, night	Theater, night	115	105	70	35
Seated, very light work	Offices, hotels, apartments	130	115	70	45
Moderately active office work	Offices, hotels, apartments	140	130	75	55
Standing, light work; walking	Department store; retail store	160	130	75	55
Walking, standing	Drug store, bank	160	145	75	70
Sedentary work	Restaurant[c]	145	160	80	80
Light bench work	Factory	235	220	80	140
Moderate dancing	Dance hall	265	250	90	160
Walking 4.8 km/h; light machine work	Factory	295	295	110	185
Bowling[d]	Bowling alley	440	425	170	255
Heavy work	Factory	440	425	170	255
Heavy machine work; lifting	Factory	470	470	185	285
Athletics	Gymnasium	585	525	210	315

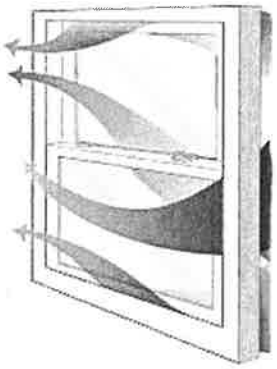
Source: Ashrae 1997 Handbook

Table 7 – Maximum lighting power budget (including ballast loss)

Type of usage	Maximum lighting power budget (Watts / m^2)
Offices	15
Classrooms	15
Lecture theatres	15
Auditoriums / Concert halls	10
Shops / Supermarkets / Departmental stores (including general, accent and display lighting)	25
Restaurants	15
Lobbies / Atriums / Concourses	10
Stairs	6
Corridors	10
Car parks	5
Electronic manufacturing and fine detail / Assembly industries	20
Medium and heavy industries	15
Warehouses / Storage areas	10

Source: SS530-2006

Infiltration Cooling Loads



Estimates of Infiltration Airflow, air changes/hr

Neutral pressure, poor construction	1.0
Neutral pressure, average construction	0.6
Neutral pressure, tight construction	0.3
Pressurized, poor construction	0.5
Pressurized, average construction	0.3
Pressurized, tight construction	0.0

Sensible KW = CMH_{IA} x DT / 3000
DT: Delta T (C) between IA & RA

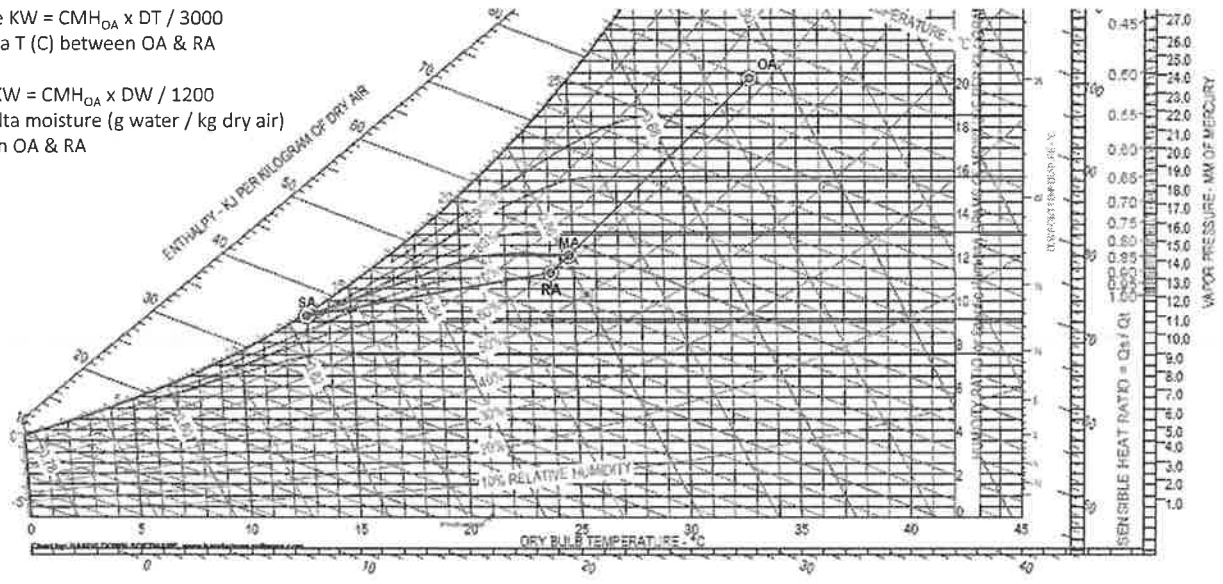
Latent KW = CMH_{IA} x DW / 1200
DW: Delta moisture (g water / kg dry air) between IA & RA

Source: Trane clinic

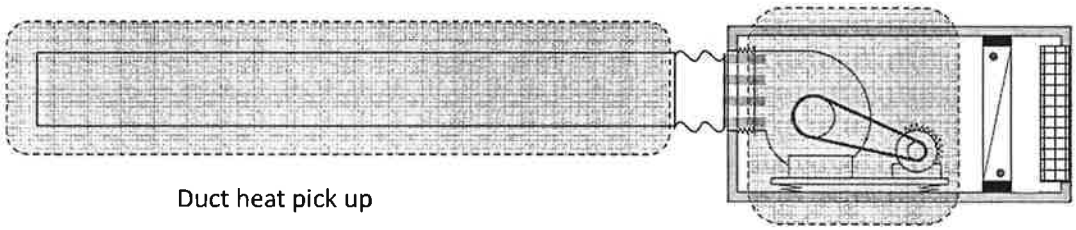
Ventilation Cooling Loads

Sensible KW = CMH_{OA} x DT / 3000
DT: Delta T (C) between OA & RA

Latent KW = CMH_{OA} x DW / 1200
DW: Delta moisture (g water / kg dry air) between OA & RA



System heat gains



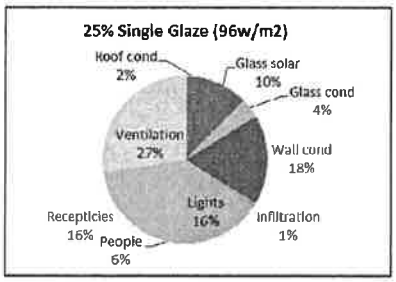
Duct heat pick up

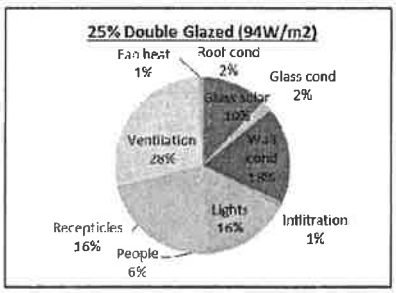
Motor & fan pick up

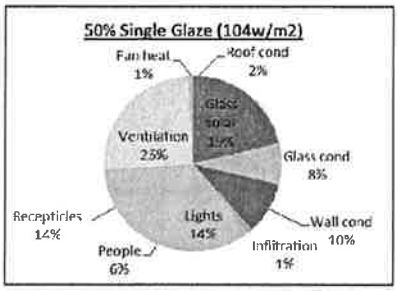
Sensible and Latent Cooling Loads

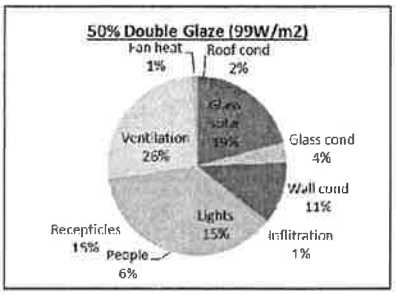
cooling load components	sensible load	latent load	space load	coil load
conduction through roof, walls, windows, and skylights	✓		✓	✓
solar radiation through windows, skylights	✓		✓	✓
conduction through ceiling, interior partition walls, and floor	✓		✓	✓
people	✓	✓	✓	✓
lights	✓		✓	✓
equipment/appliances	✓	✓	✓	✓
infiltration	✓	✓	✓	✓
ventilation	✓	✓		✓
system heat gains	✓			✓

Cooling Load Component Breakdown



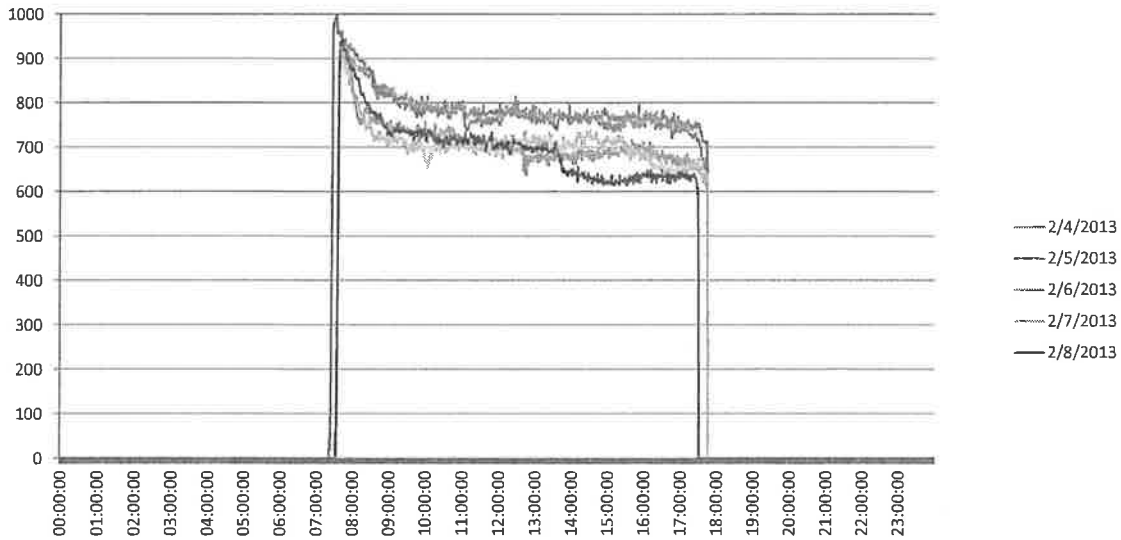




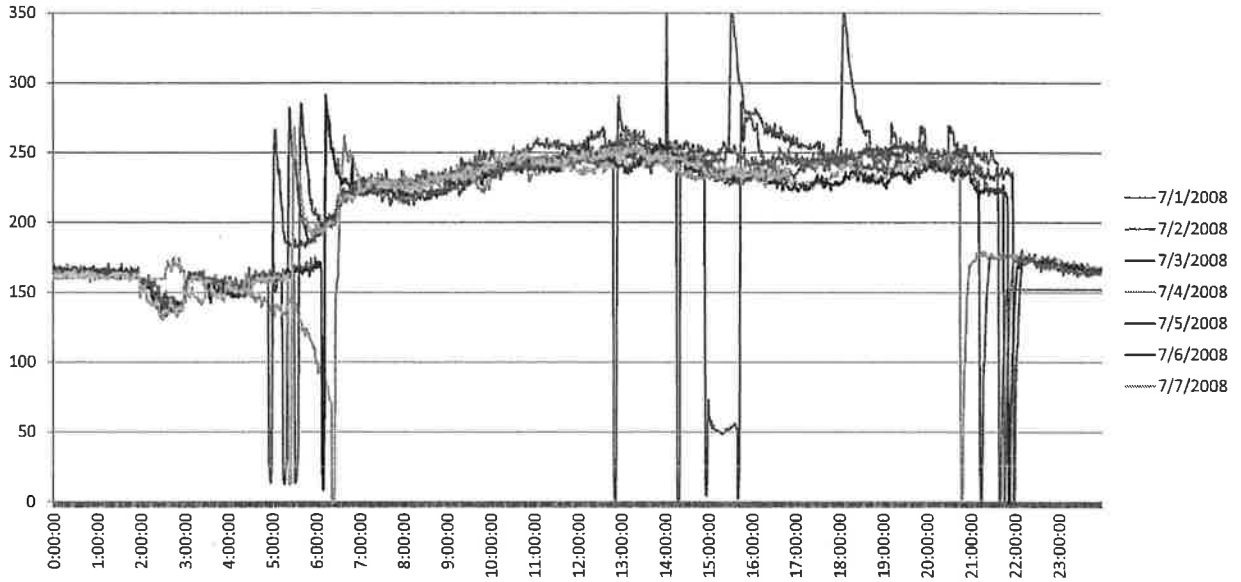


Solar Load
Ventilation Dir

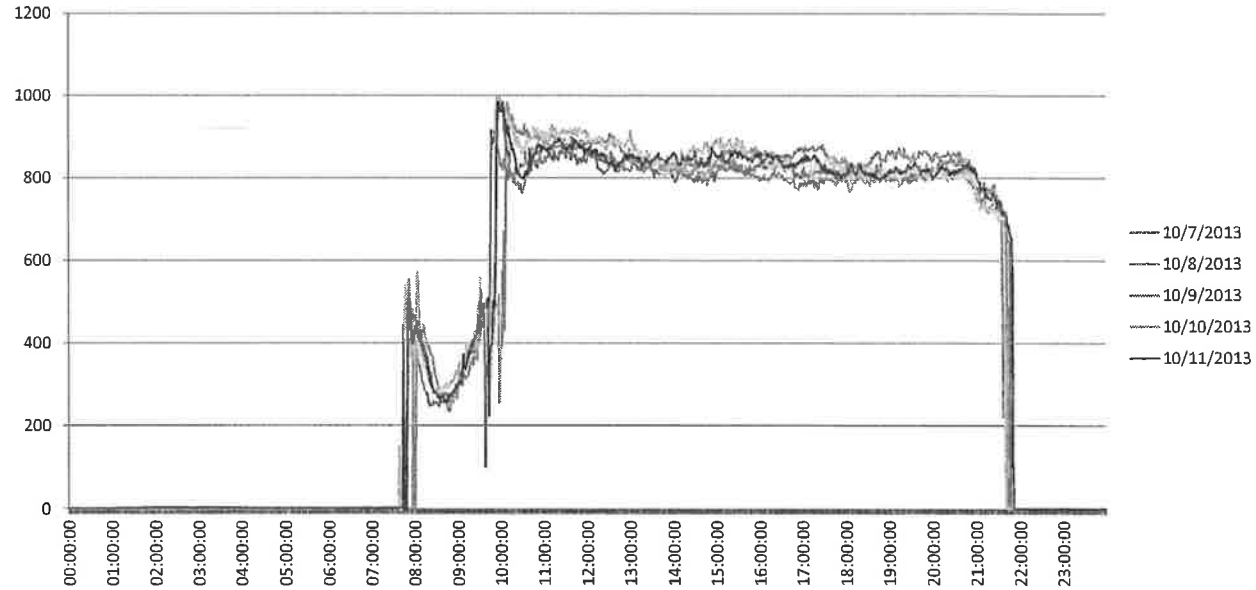
Cooling Load Profile (Office)



Cooling Load Profile (Hotel)

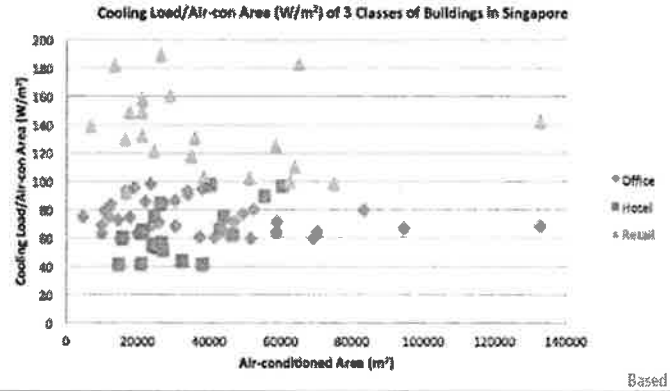


Cooling Load Profile (Malls)



Cooling Load Benchmarking Data

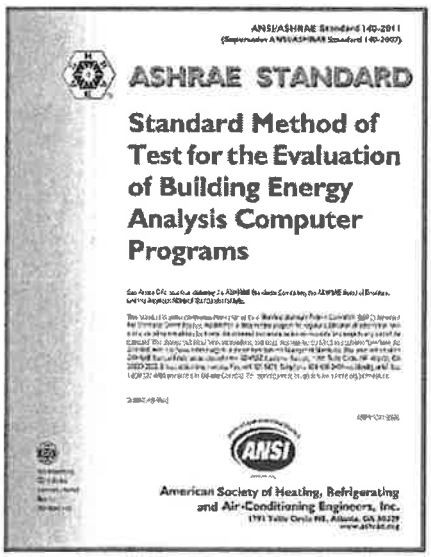
	Measured Cooling Load/Air-con Area (W/m²)	
	Range	Average
Office (35 projects)	60 – 98	76
Hotel (24 projects)	42 – 98	64
Retail (21 projects)	93 – 190	135



Based on data from BCA and NEA

Software For Calculating Cooling Load

- Ashrae standard 140



- DOE2
- eQUEST
- EnergyPlus
- EnergyPro
- IES (Integrated environmental solutions)
- HAP (Carrier HAP)
- TRACE 700 (Trane TRACE)
- Visual DOE

Measurement & Verification Pte Ltd

Thank you for your attention

Discussion and Q&A



Measurement & Verification Pte Ltd

BASIC PSYCHOMETRICS

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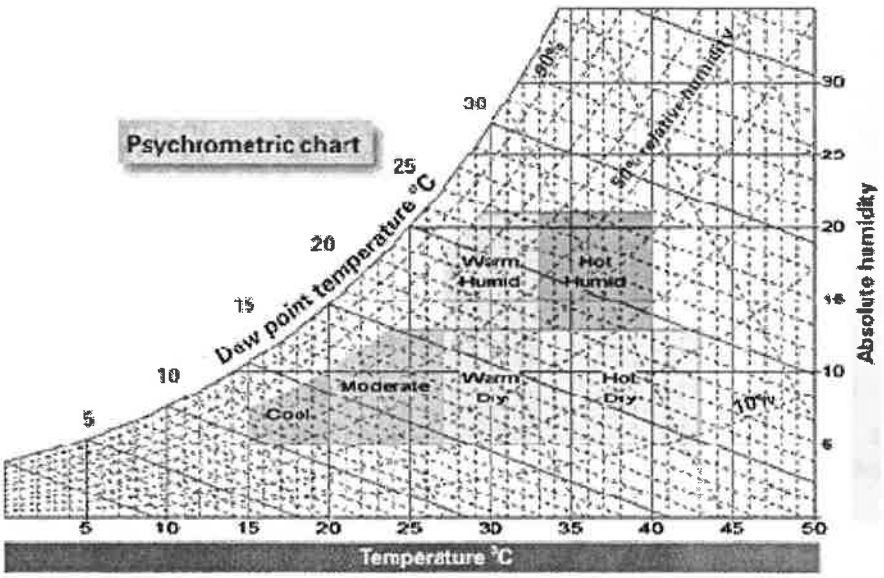
Agenda

1. Psychometrics- Seven Properties Of Air
2. Psychometric Chart
3. Formulae
4. Exercise

What is Psychometrics?



PSYCHOMETRICS

SEVEN PROPERTIES OF AIR

- DRY BULB TEMPERATURE
- HUMIDITY RATIO
- VOLUME
- WET BULB TEMPERATURE
- RELATIVE HUMIDITY
- ENTHALPY
- Dew Point TEMPERATURE

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PSYCHOMETRIC CHART

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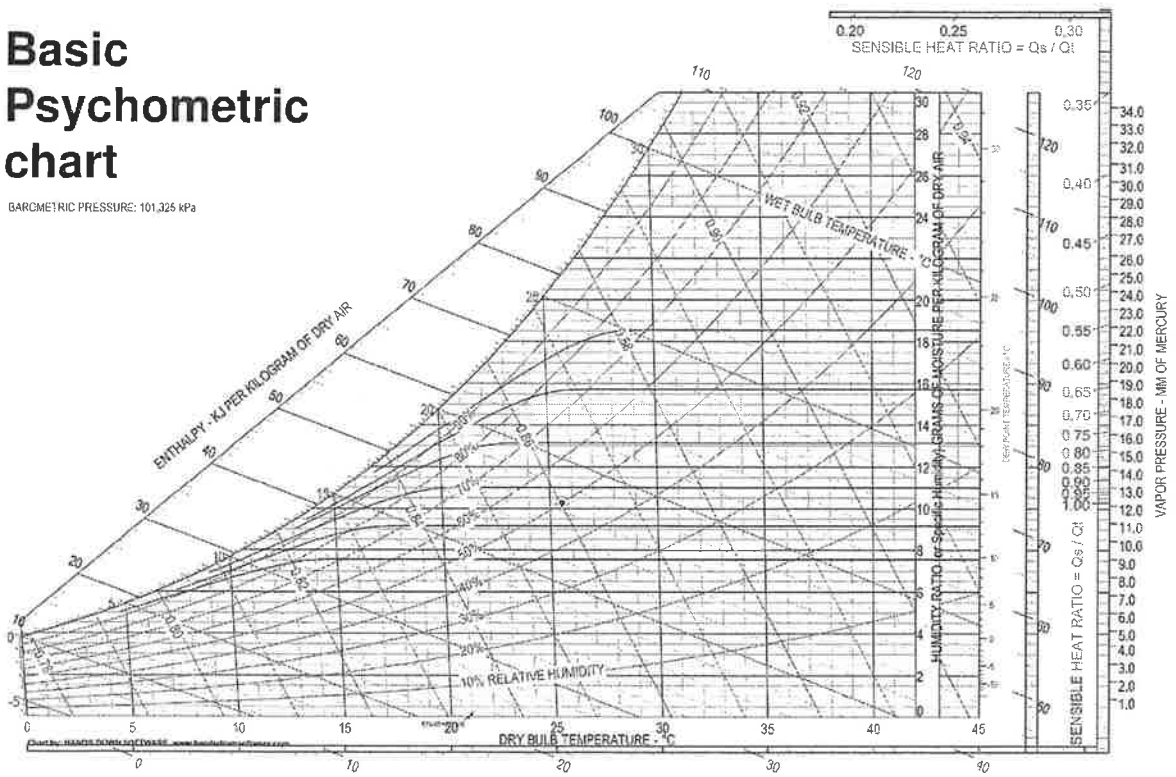
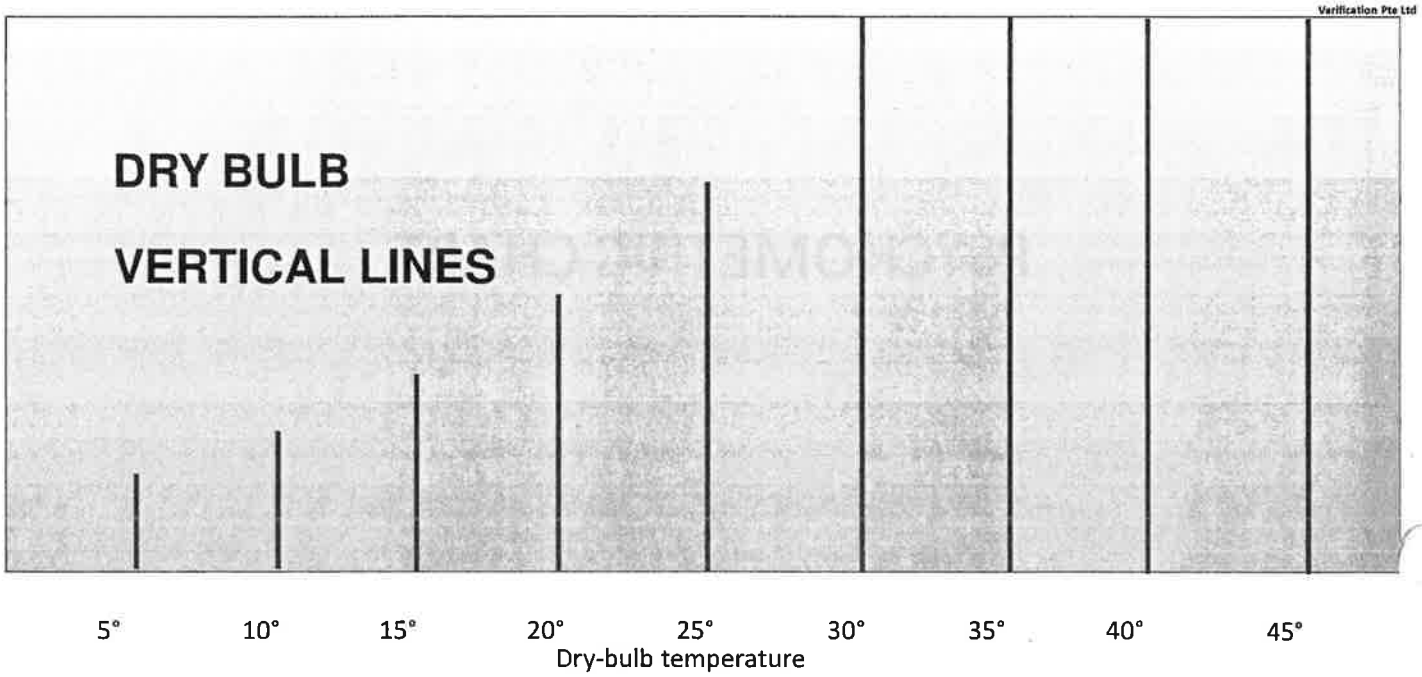
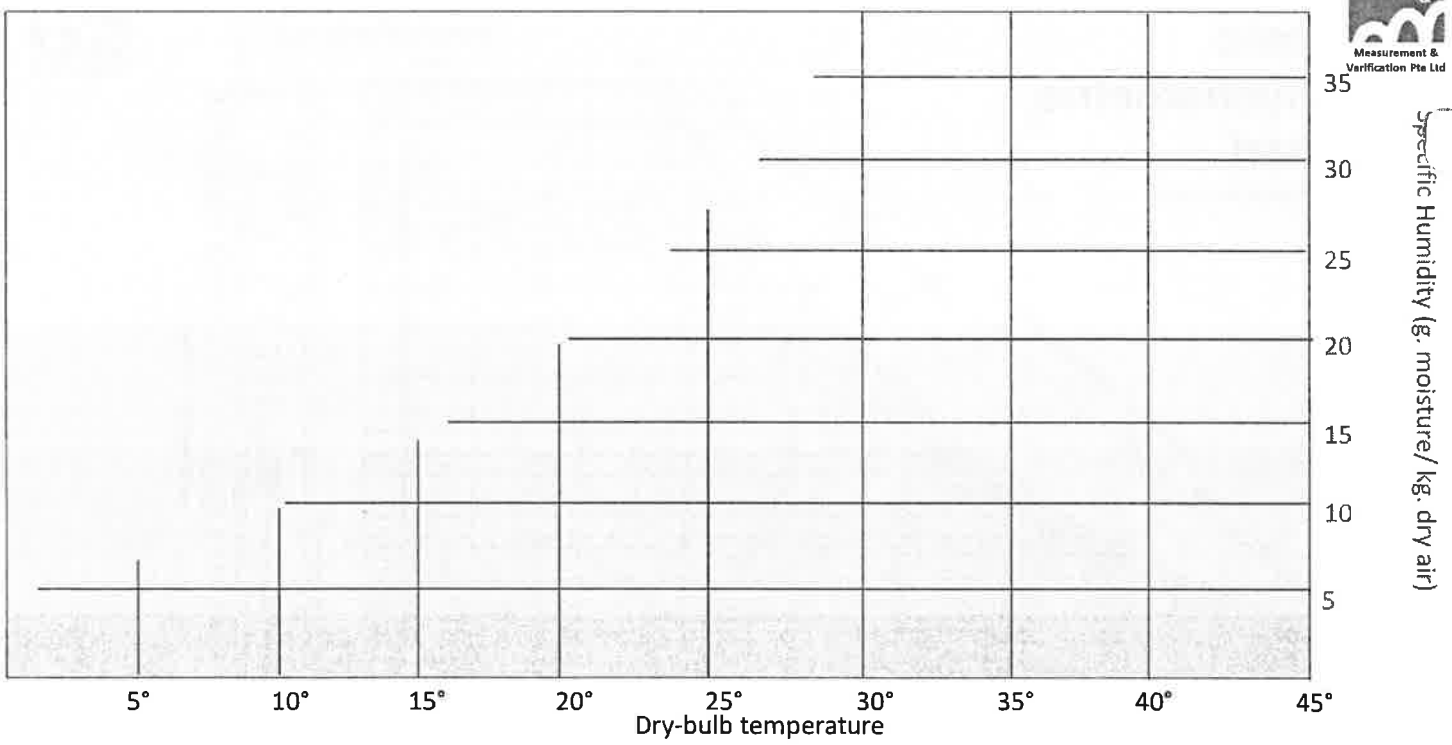
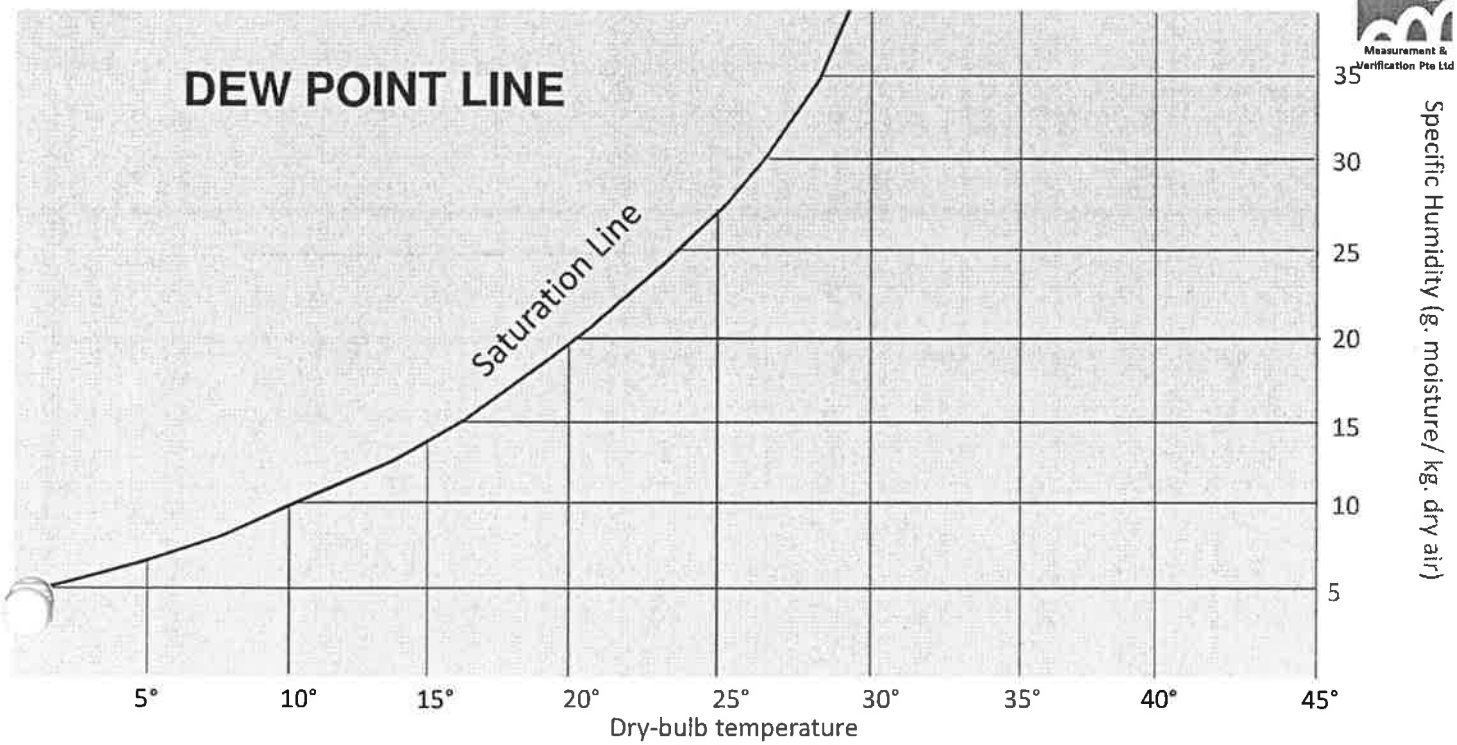
Basic Psychometric chart

BAROMETRIC PRESSURE: 101,325 kPa

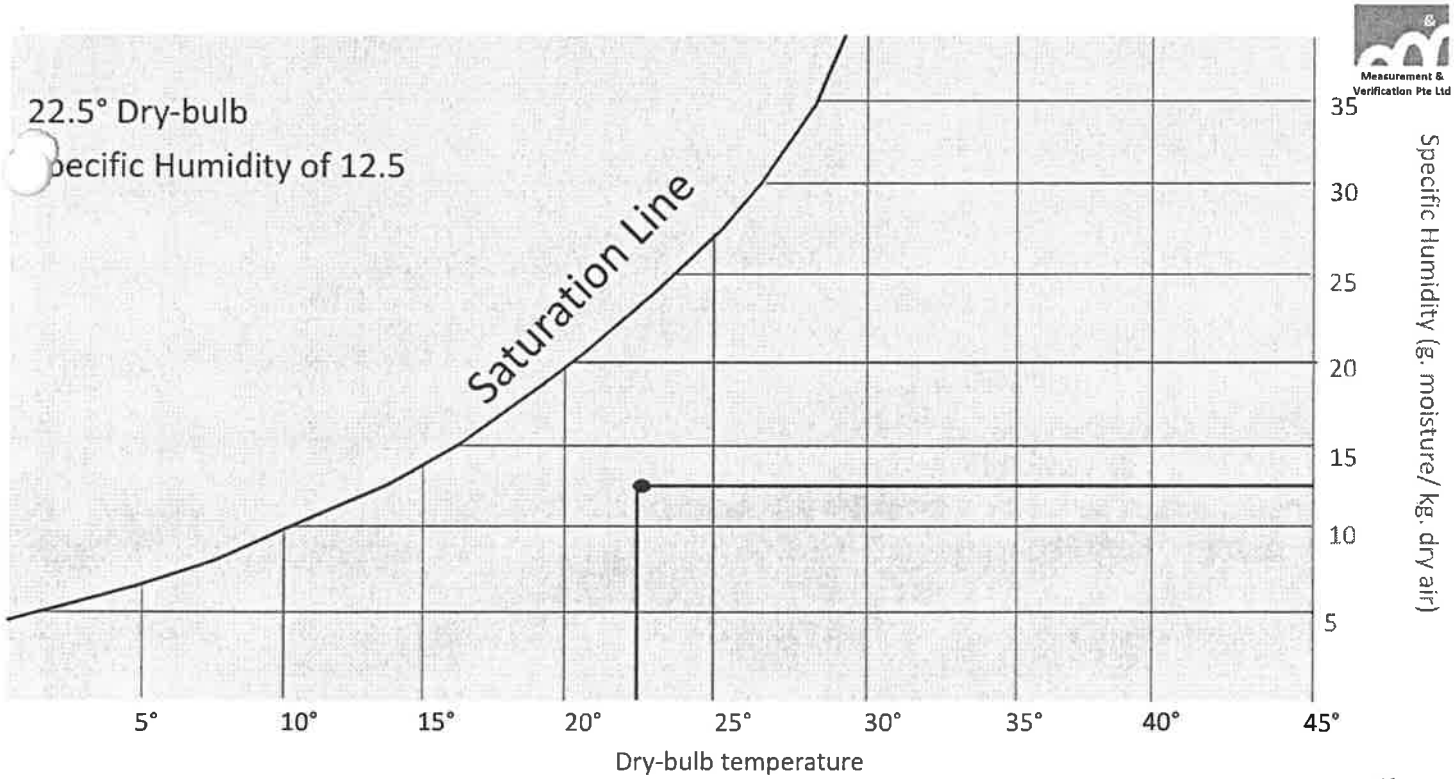



DRY BULB
VERTICAL LINES
5°
10°
15°
20°
25°
30°
35°
40°
45°
Dry-bulb temperature


35
30
25
20
15
10
5
Specific Humidity (g. moisture/kg. dry air)
5°
10°
15°
20°
25°
30°
35°
40°
45°
Dry-bulb temperature



Proprietary Information of Measurement & Verification Pte Ltd



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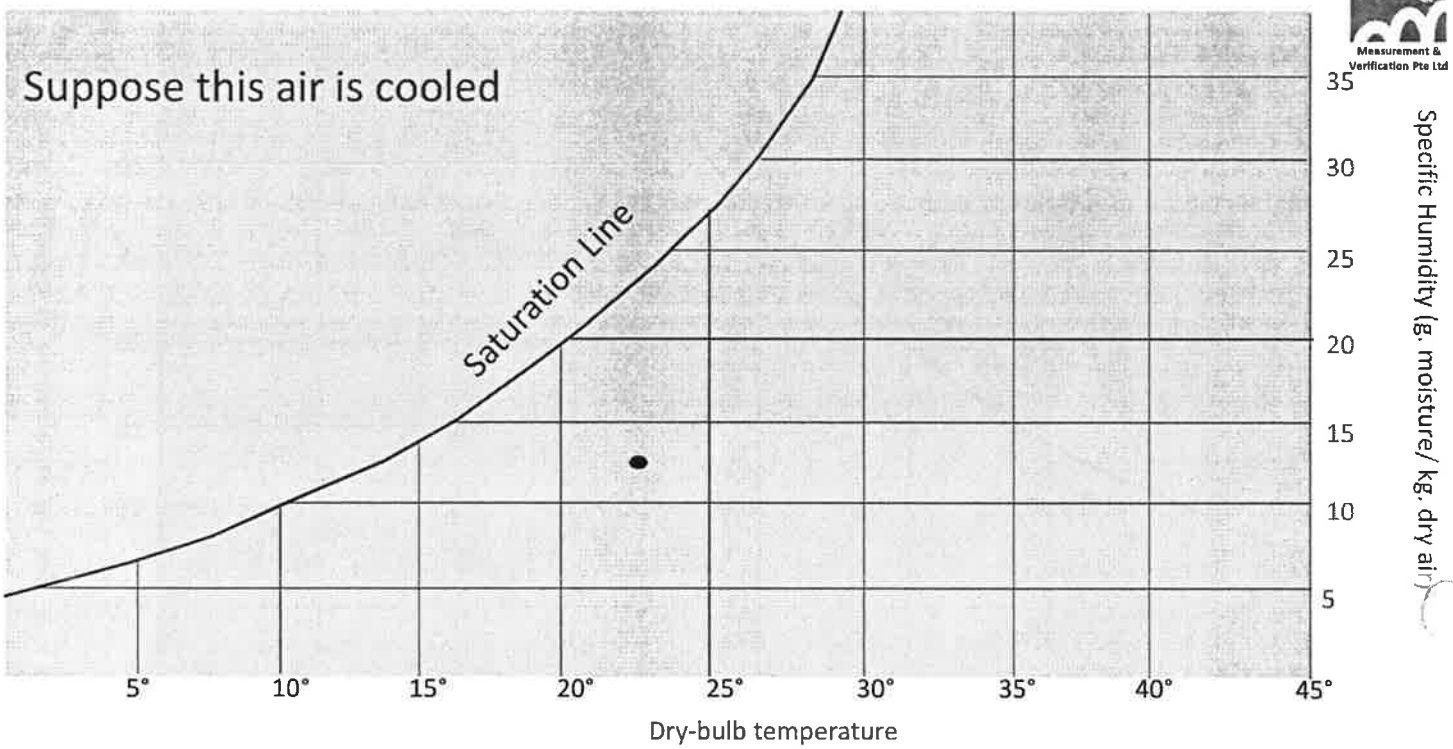

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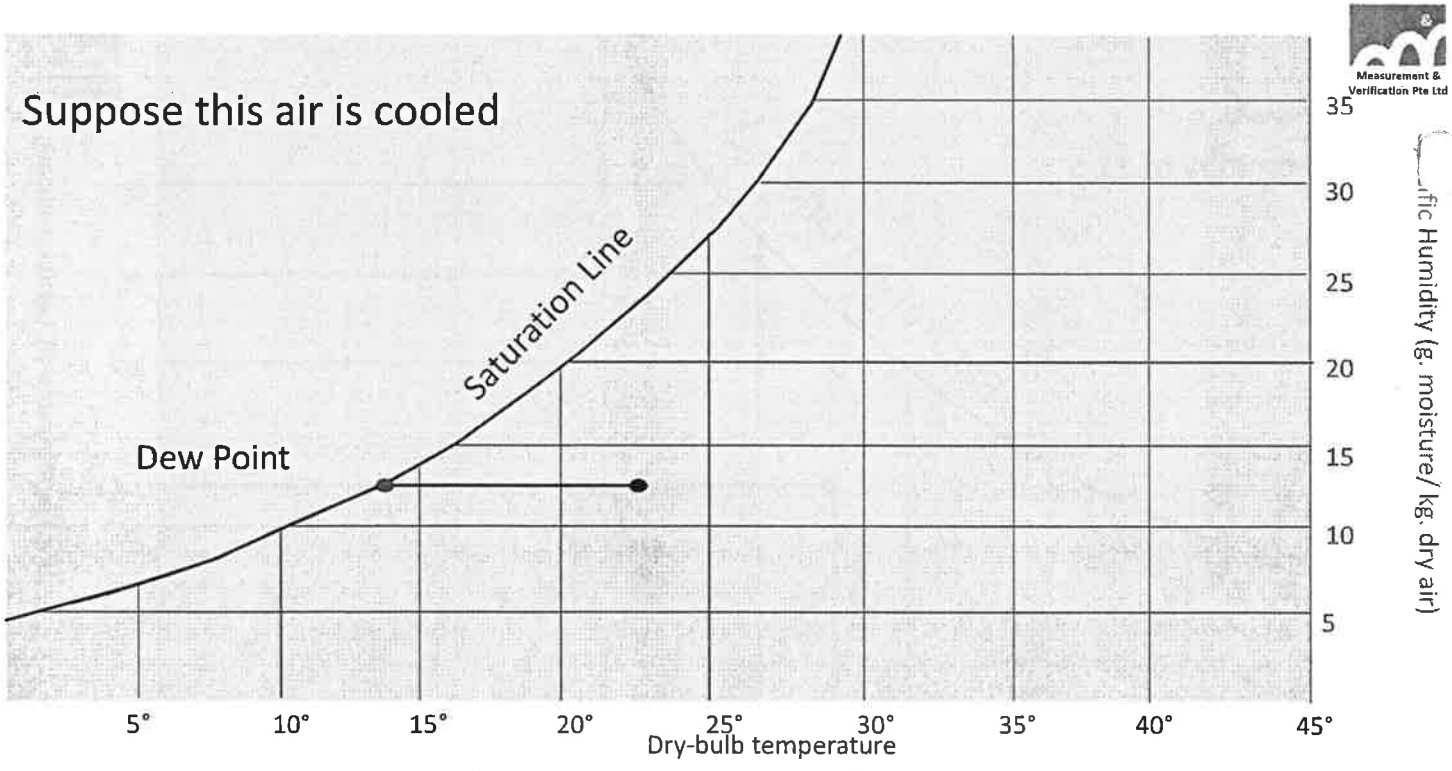

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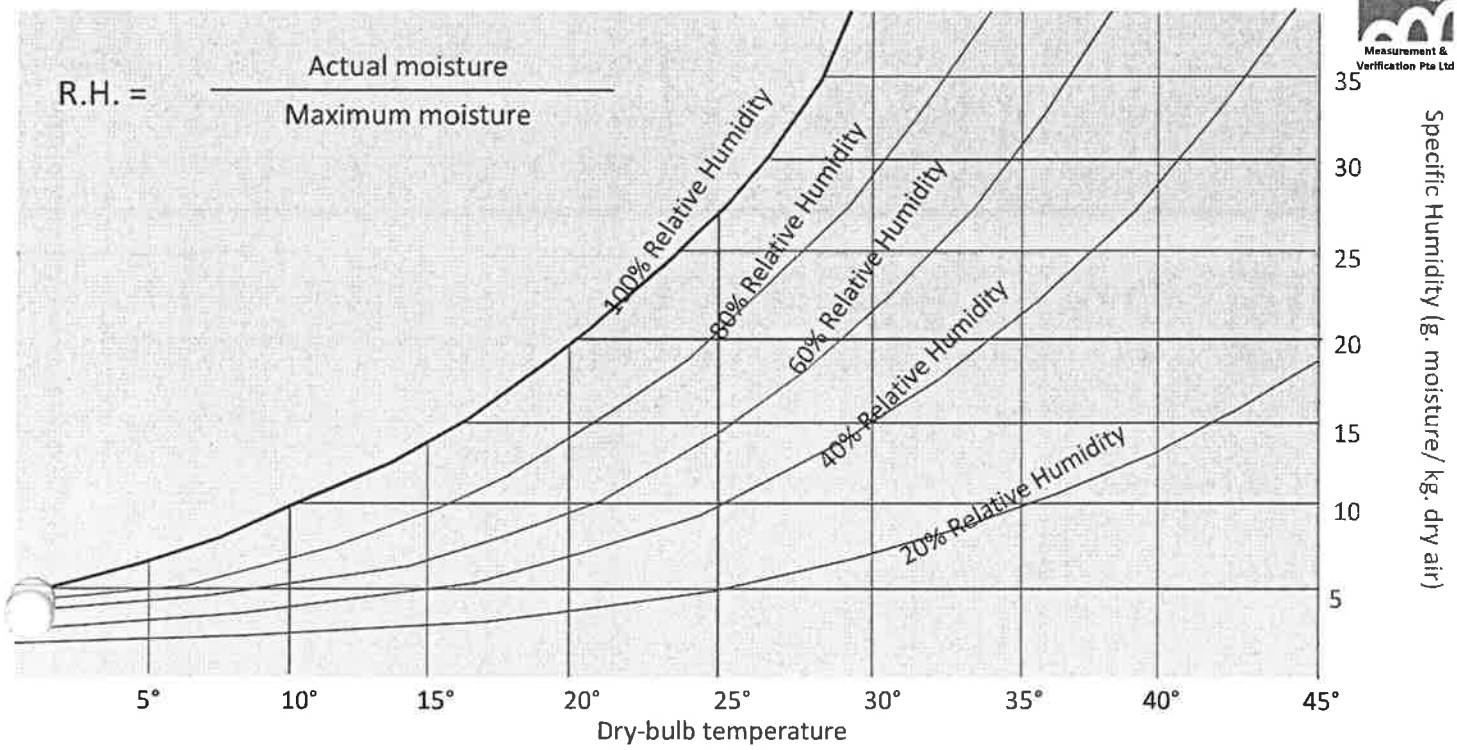
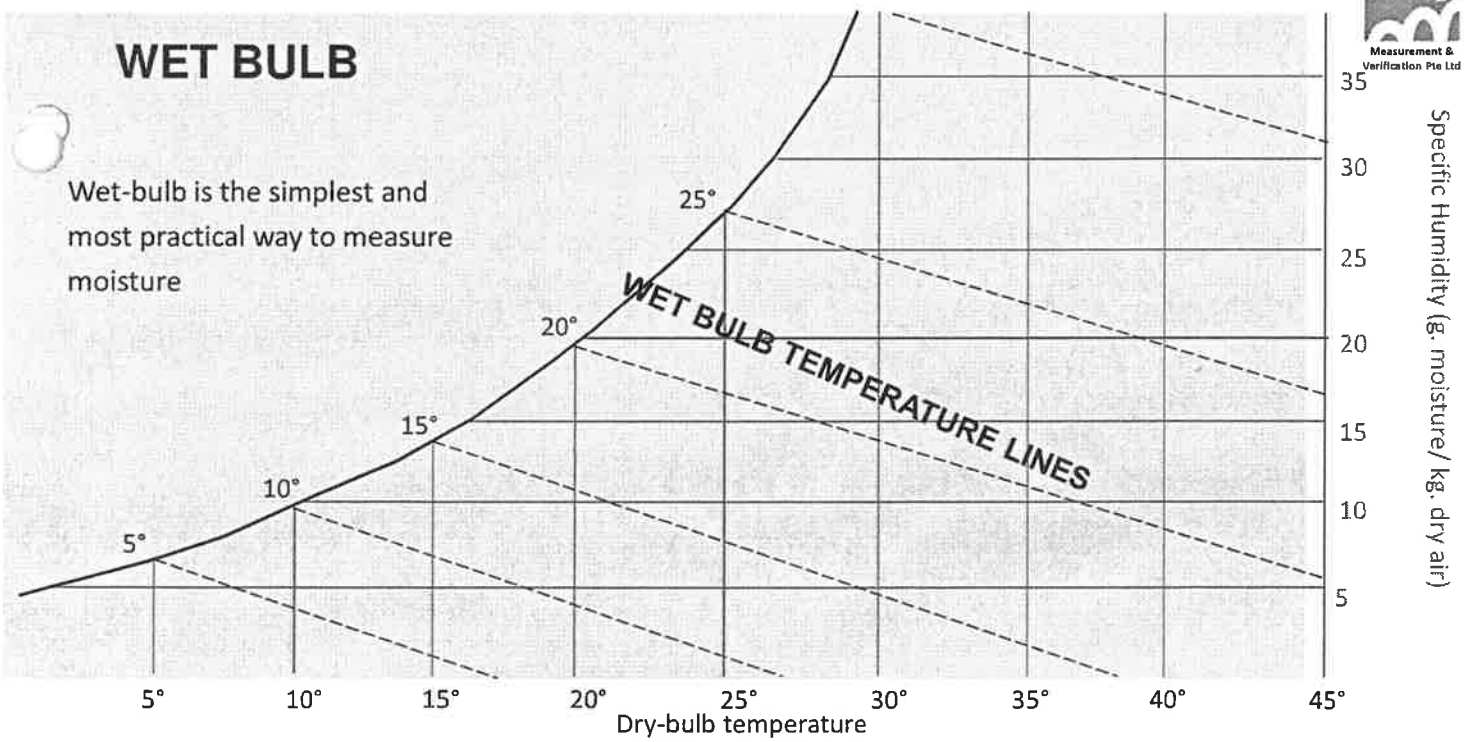
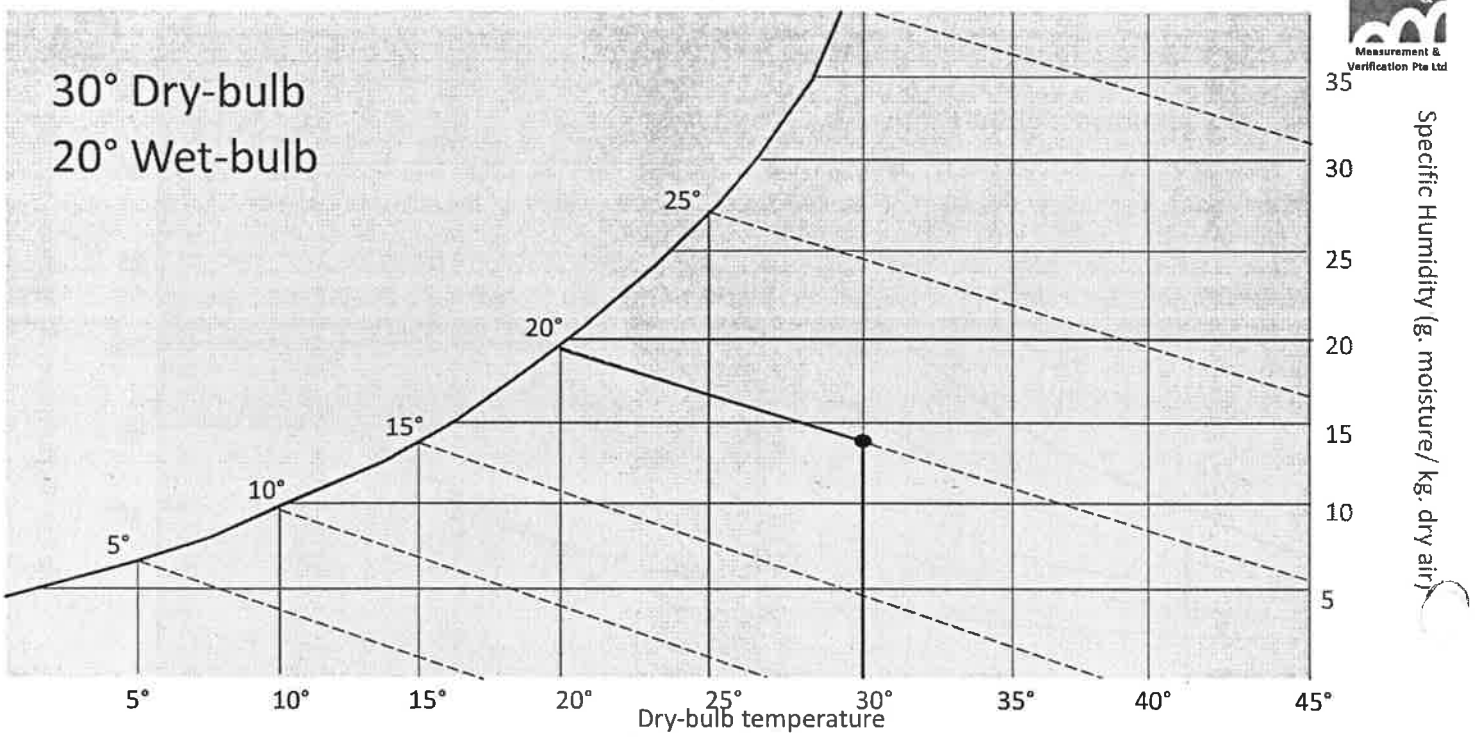

R.H. = Actual moisture / Maximum moisture
100% Relative Humidity
80% Relative Humidity
60% Relative Humidity
40% Relative Humidity
20% Relative Humidity
5°
10°
15°
20°
25°
30°
35°
40°
45°
Dry-bulb temperature
5
10
15
20
25
30
35
Specific Humidity (g. moisture/ kg. dry air)


WET BULB
Wet-bulb is the simplest and most practical way to measure moisture
WET BULB TEMPERATURE LINES
5°
10°
15°
20°
25°
5°
10°
15°
20°
25°
30°
35°
40°
45°
Dry-bulb temperature
5
10
15
20
25
30
35
Specific Humidity (g. moisture/ kg. dry air)



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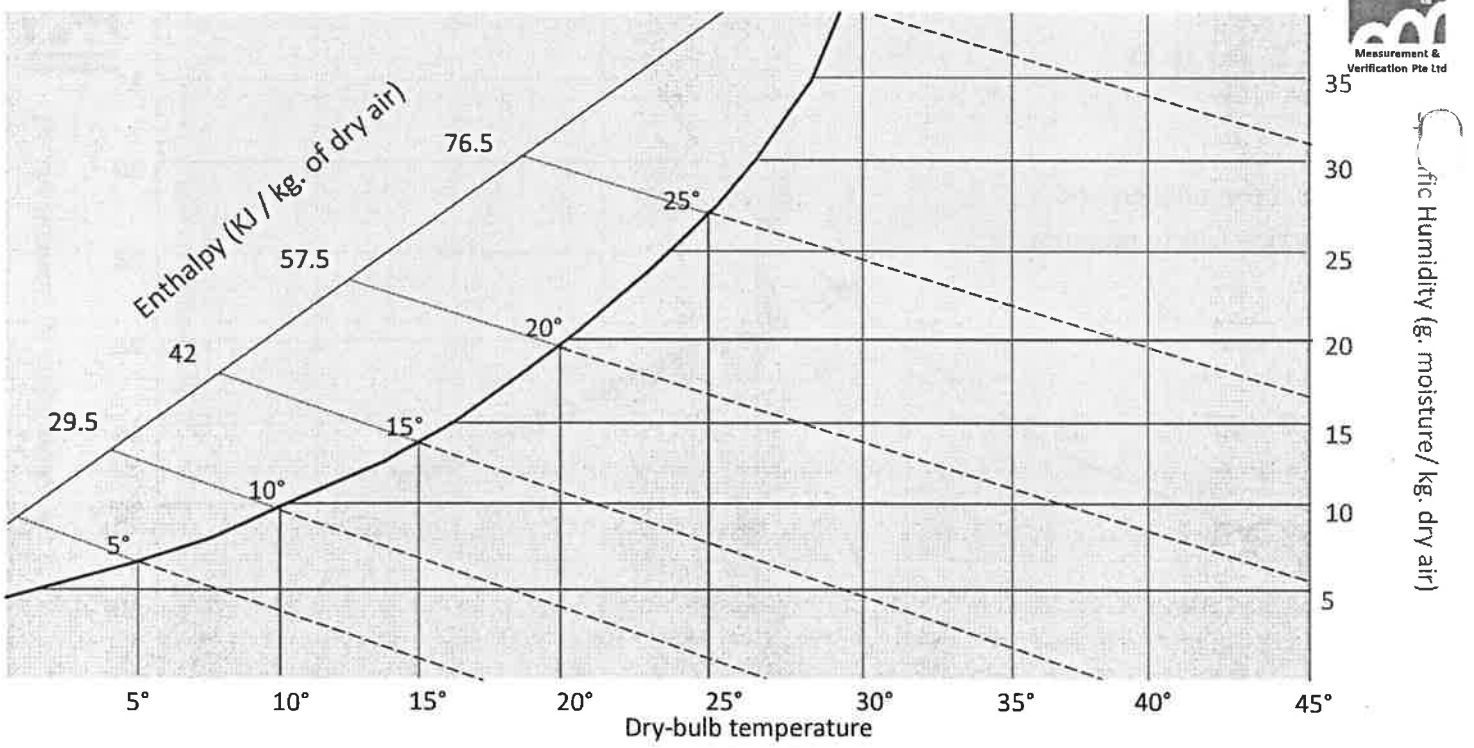

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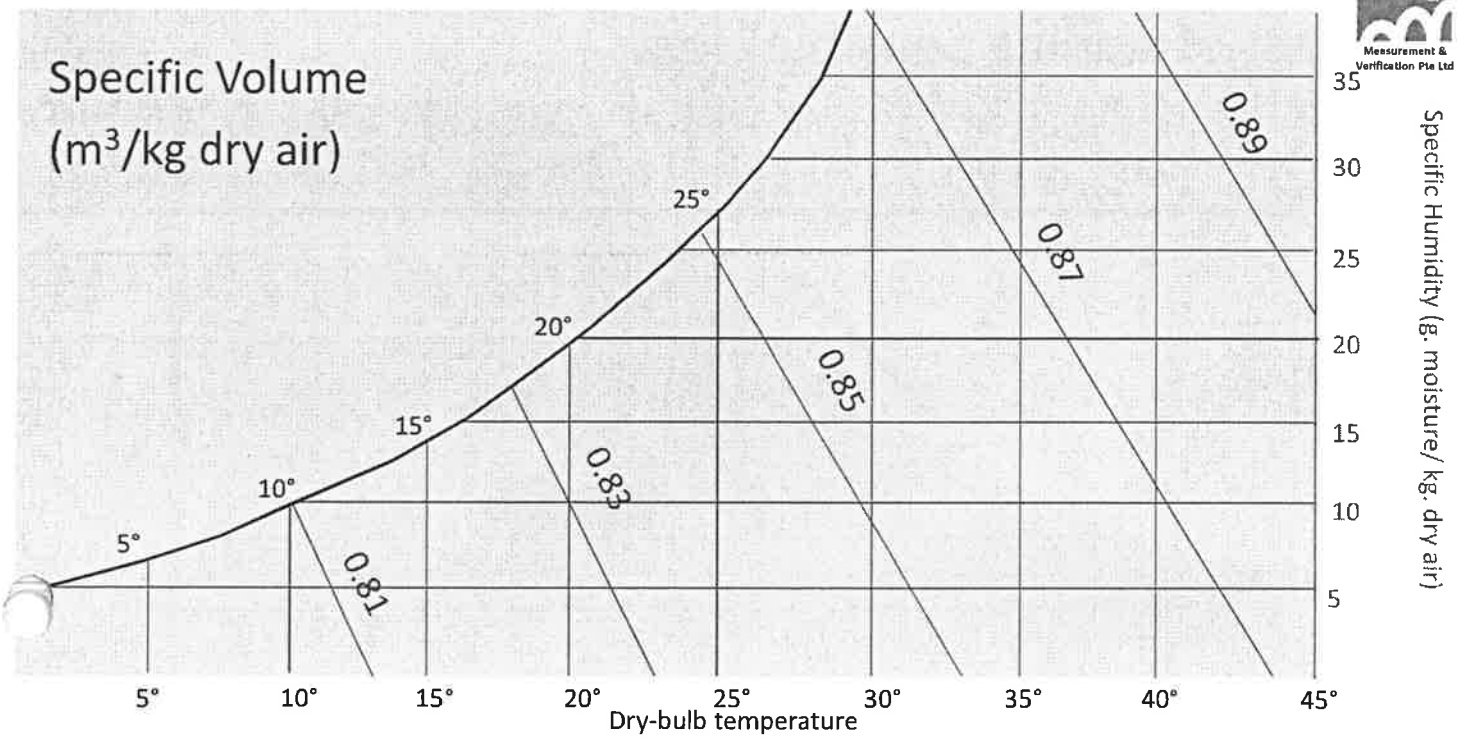

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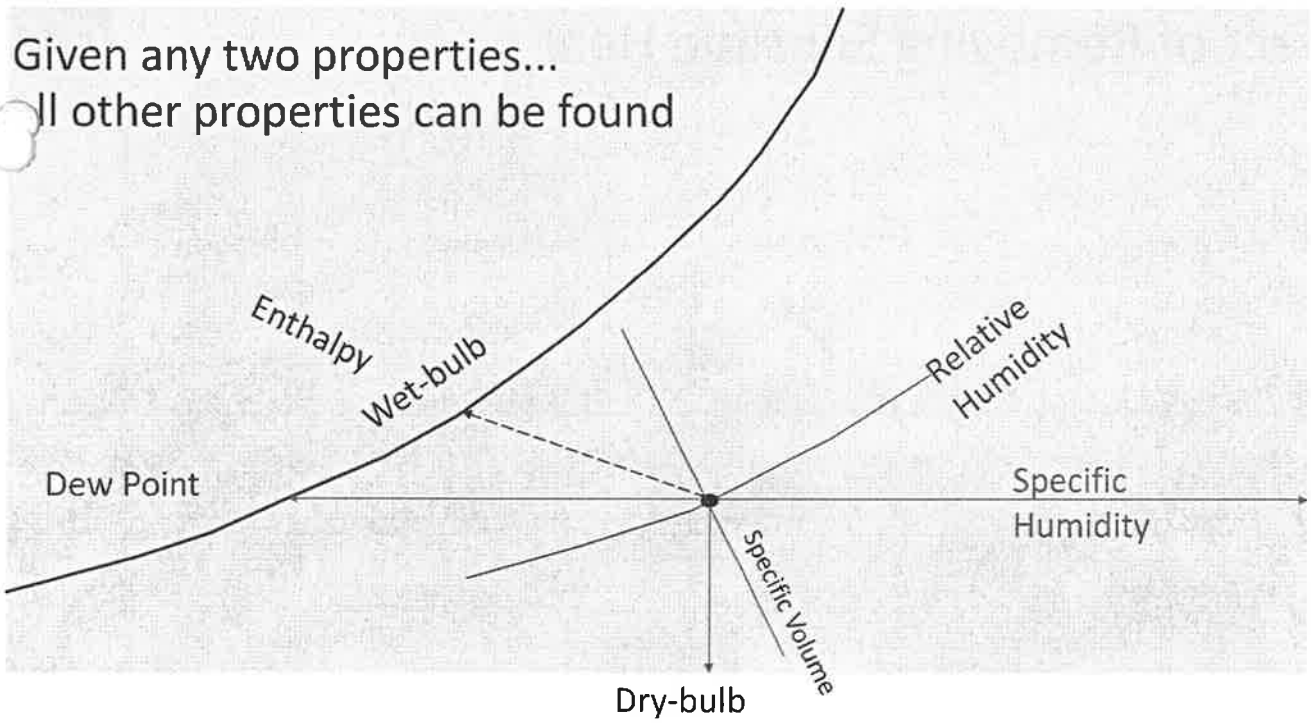

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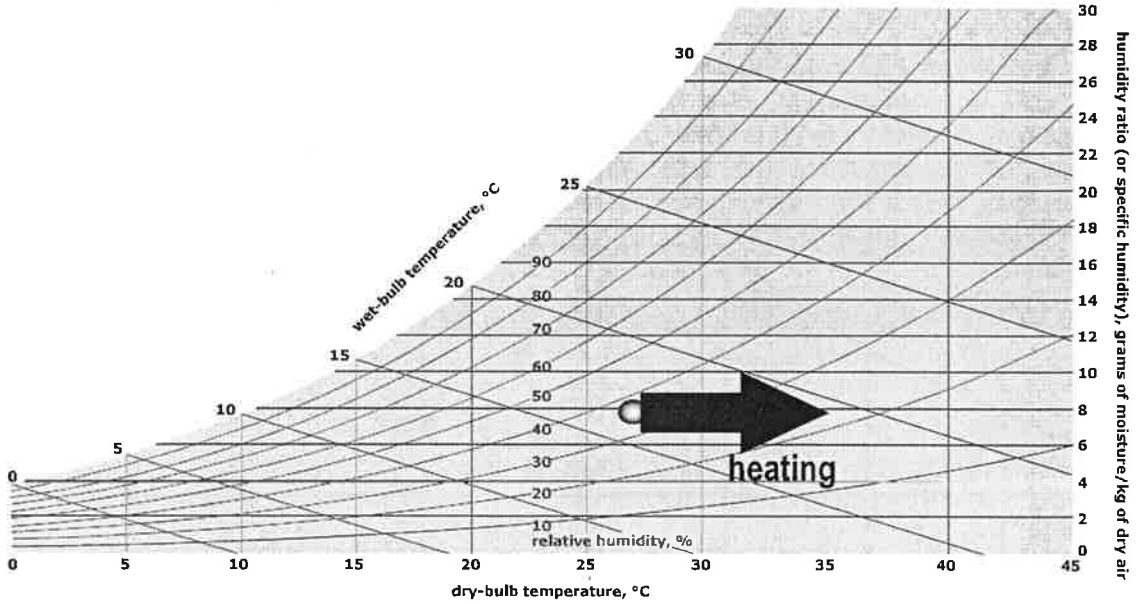
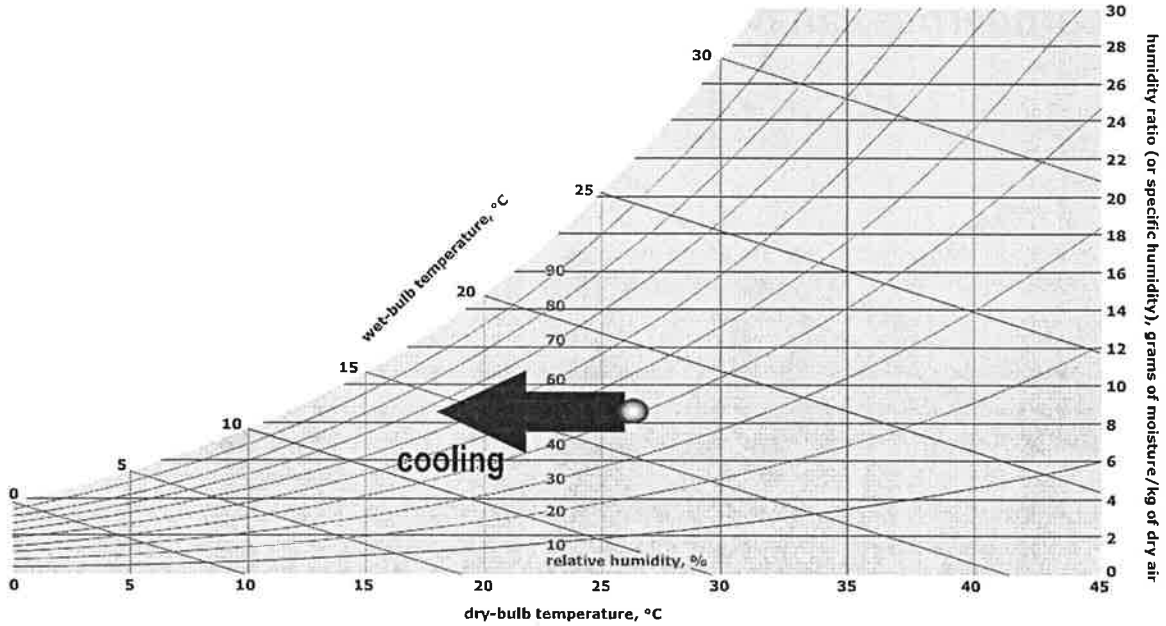
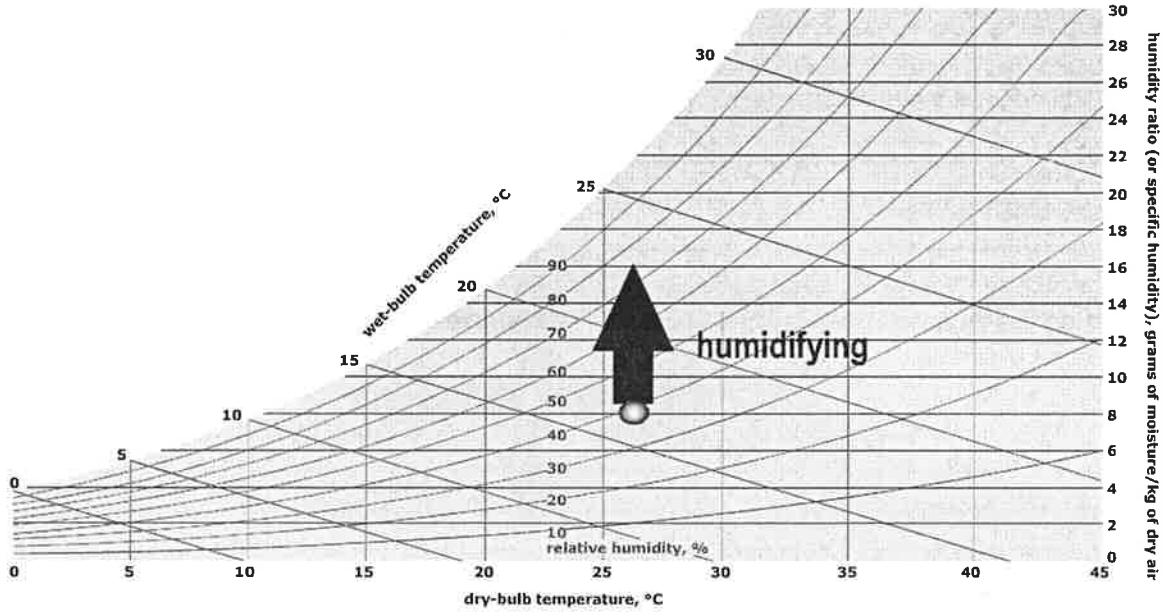
Effect of Adding Sensible Heat



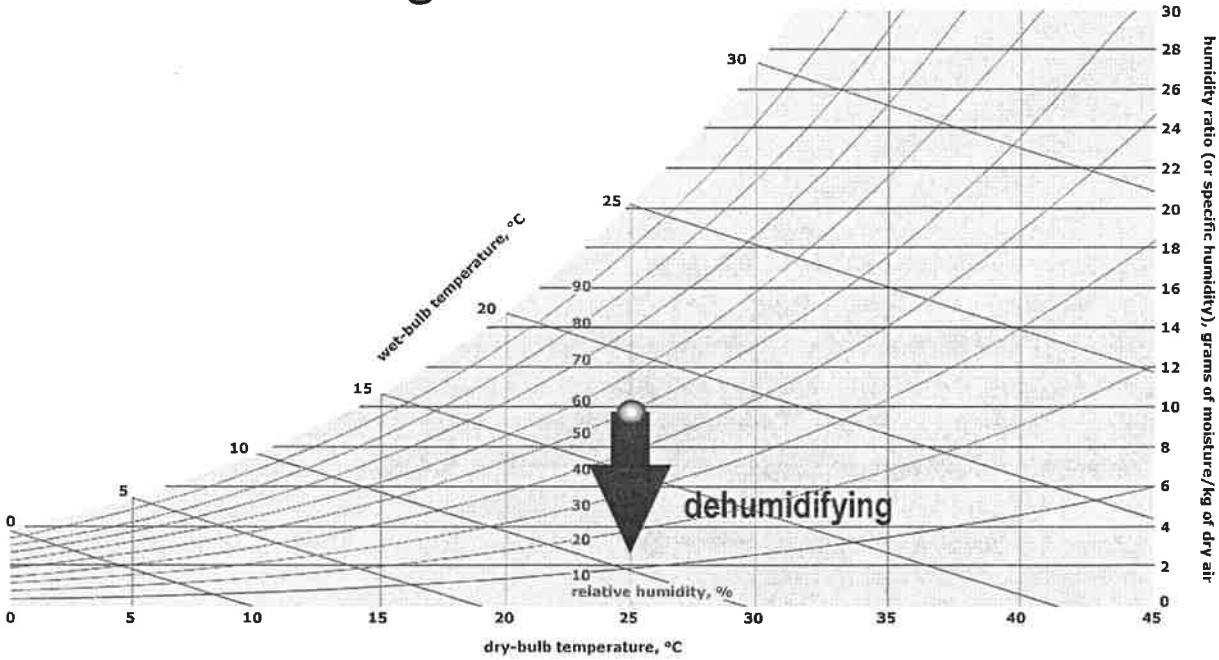
Effect of Removing Sensible Heat



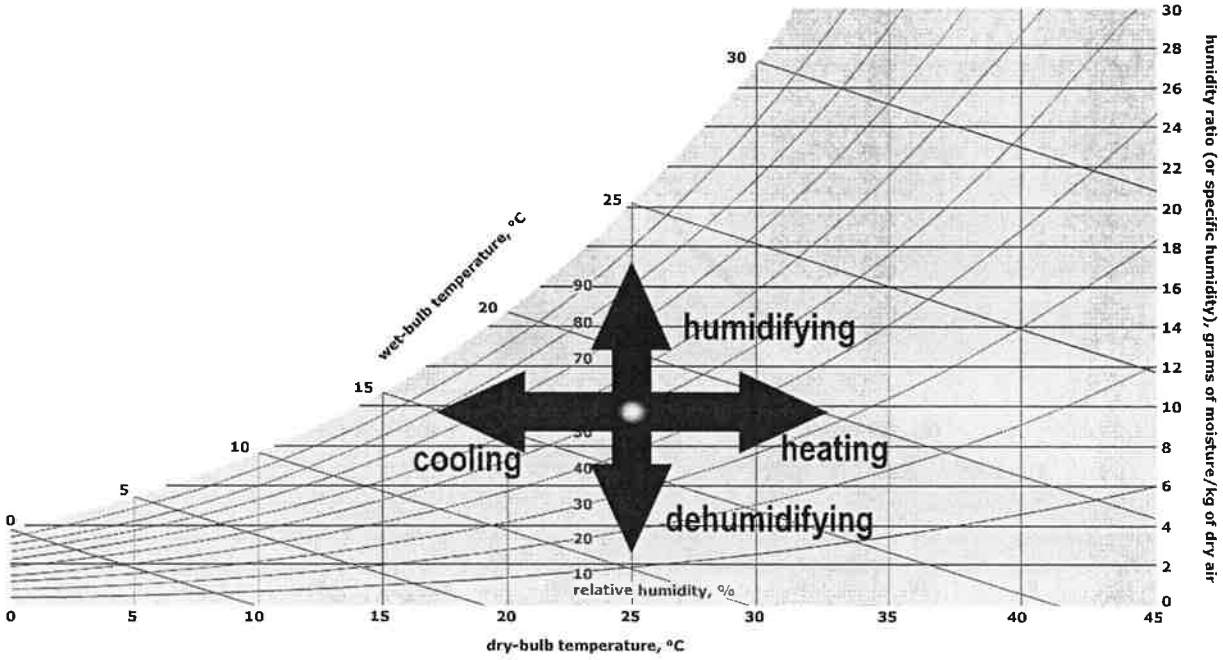
Effect of Adding Moisture



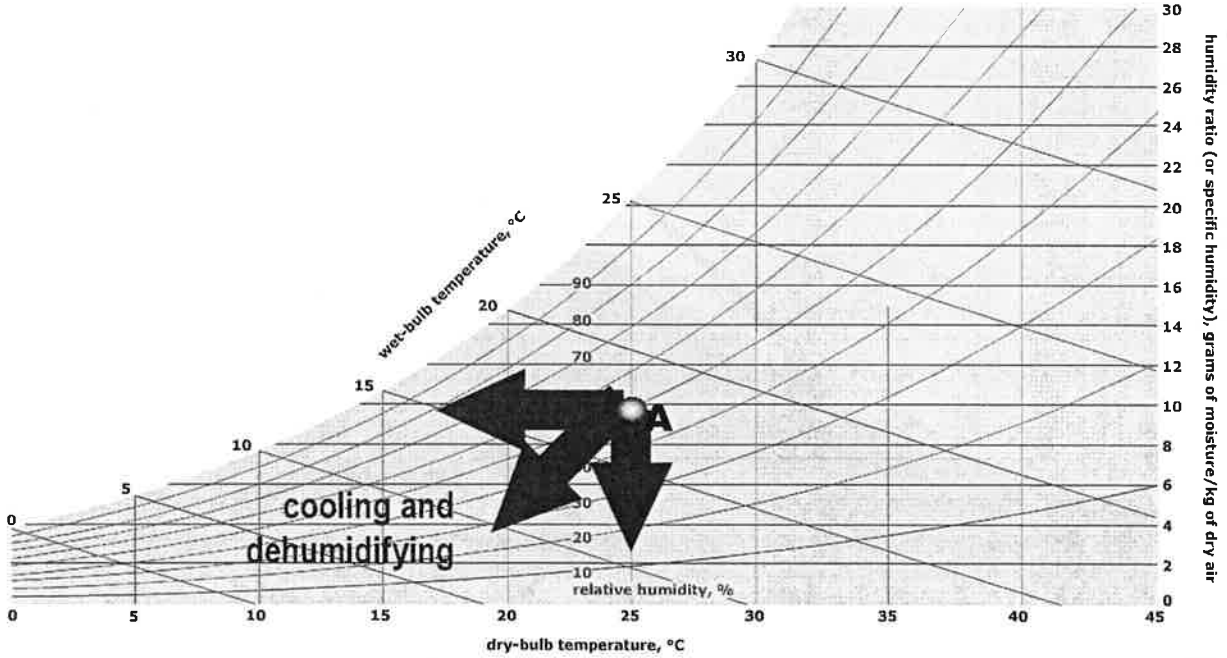
Effect of Removing Moisture



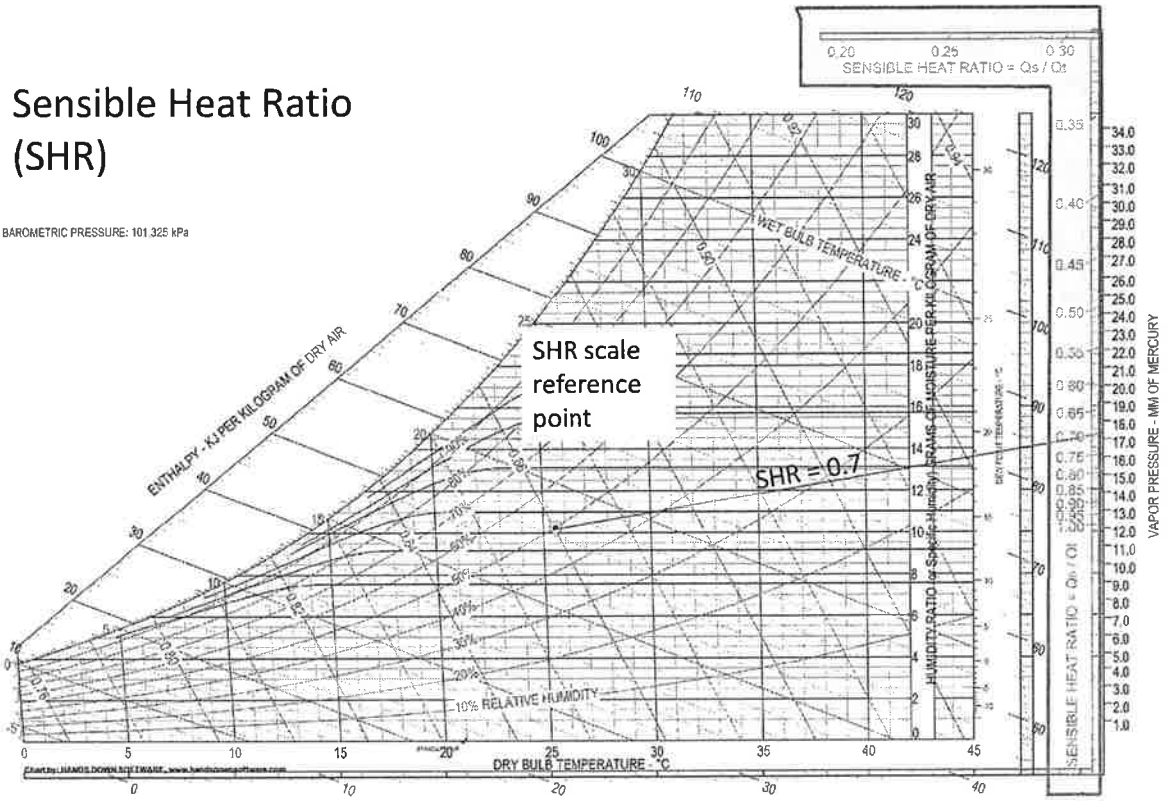
Removing Sensible Heat and Moisture



Removing Sensible Heat and Moisture



Sensible Heat Ratio (SHR)



FORMULAE

Sensible load Q(W) = $\dot{M}$ (kg/s) x Cp (J/kg.k) x DT (C)

Total load Q (W) = $\dot{M}$ (kg/s) x Cp (J/kg.k) x DH (J/kg.k)

At standard air conditions
Density = 1.2 kg/m^3 and
Specific heat = 1004 J/kg.K

SI Formulars

Sensible KW = CMH x DT (C) / 3000

Latent KW = CMH x DW (g/kg dry air) / 1200

Total KW = CMH x DH (kj/kg) / 3000

IP formulars

Sensible tons = 1.085 x CFM x DT (F) / 12000

Latent tons = CFM x DW (grain/lb dry air) / 17391

Total tons = 4.5 x CFM x DH (btu/lb) /12000

EXERCISE

Exercise

Plot a typical air-conditioning process on the psyc chart.

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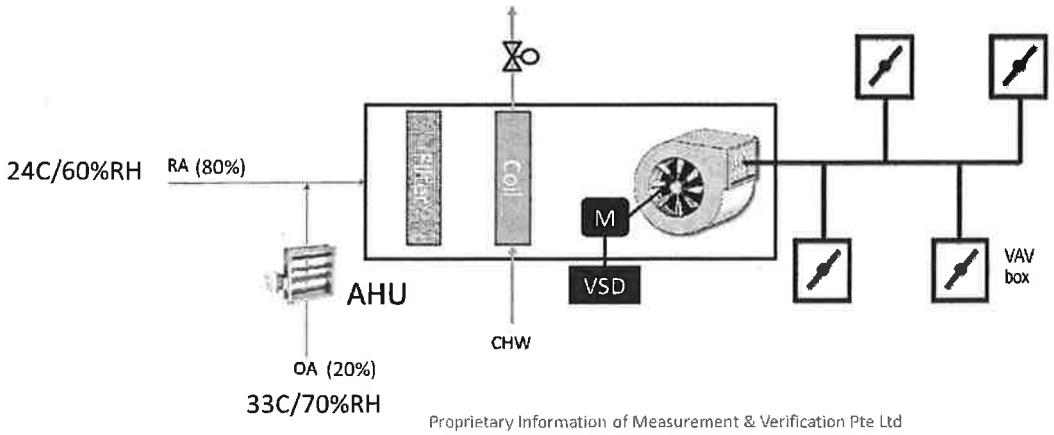
Question 1:

I'm starting with an outdoor condition of 33C and 70% Relative Humidity and I'm mixing that air with return air that is at 24C and 60% relative humidity. If the mixed air is 80% return air and 20% outside air, how do I find the correct mixed air temperature and % Rh and graph this on the psychometric chart properly?



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Question 1 Answer

To find the solutions for this question I would approach the problem like this:
1. Plot OA point on chart at 33C, 70% Rh
2. Plot RA point on chart at 24C, 60% Rh
3. Connect the OSA point and the RA points to form the mixed air line.
4. Measure length of line OA to RA (eg "a" mm)
5. Measure along the MA line starting from the OA point, (OA CMH / Total CMH) x "a" mm. This is your MA temperature and humidity.
6. Or use the following simple formula to find the mixed air Db temperature.
(RA temp x RA%) + (OA temp x OA%) = Mixed Air temp, plot this on the MA line to find the corresponding %Rh

$(24 \times 0.8) + (33 \times 0.2) = 25.8$

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Question 2:

How do I figure out the required supply air temperature for the room? I know that I need to find the structural heat gains, electrical heat gain, occupant load and infiltration load. I can add the sensible heat and latent heat gains for the space and figure out what the total heat gain for the room is. But how do I figure out the necessary supply air temperature and % Rh to maintain that room at 24C and 60% Rh?

Assume room total heat = 35KW and room sensible heat = 26.3KW.

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Question 2 Answer

To find this answer requires the use of the sensible heat ratio. The sensible heat ratio is a ratio of sensible heat to total heat based on your space load. The protractor on the upper left side of the psychometric chart lets you plot the ratio or slope on the chart. Look closely at the protractor, the numbers on the inside radius are the ratios of sensible heat. Suppose your sensible heat ratio for your problem was 0.75 based on your rooms' sensible heat loads and latent heat loads: (sensible heat + latent heat = total heat) so that sensible heat divided by total heat = The Sensible Heat Ratio (SHR)

1. Draw a line from the center point on the base line of the protractor through the 0.75 on the radius of the protractor. This establishes your slope.
2. Now plot your desired Db and %RH for the space on the chart, 24C Db 60%RH.
3. Transfer the sensible heat line or slope (remember the protractor?) so that it starts at your desired room temperature (the RA point) and travels to the left at the .75 SHR slope and intersects the 95% curve on the chart, at 95%Rh.
4. The point of intersection will establish the required supply air temperature (the SA point) to maintain the desired space temperature.
5. You could select any desired supply air temperature located on this line in theory and adjust your CMH, but most A/C equipment delivers air at 95% Rh or higher so you are some what limited, so much for theories.

15°C

~~35~~ 26.3 = CMH × DT / 3000

CMH = ((24.15)/3000) / 35 =

Question 3:

Now that I know what the required supply air temperature is, how do figure out what my CMH requirements are? Suppose the room total cooling load is 35KW and the room sensible cooling load is 26.3KW.

Question 3 Answer

1. Once you have established your supply air temperature, use the supply air temperature and the desired room air temperature to find the CMH, using the following formula:
   Sensible KW = CMH x DT (c) / 3000

2. Another method would involve the total heat formula:

Total KW = CMH x DH (kj/kg) / 3000. This requires you to find the enthalpy of the return air and supply air from the psychrometric chart.

Question 4.

In my original problem I was bringing in 20% OA at 33C and 70% Rh.
Size the coil to handle the total load.

Question 4 answer

Does the OA affect the load in the room? Simply stated NO!

The mixed air temperature is higher than the desired space air temperature, it does put an additional load on the cooling coil but the room never sees this added load.

1. Locate the MA point on the mixed air line and find the enthalpy corresponding to this point.
2. Locate the SA point on the chart and find the corresponding enthalpy.
3. Using the difference of these two values and the CMH that we found in question 3, solve for Sensible KW and total KW
Sensible KW = CMH x DT (c) / 3000
Total KW = CMH x DH (kj/kg) / 3000
4. Notice in this problem the total heat the coil sees is going to be more than the load of the room, the CMH will remain the same.

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Question 5.

Is there a way to find the amount of moisture that the coil is removing from the air?

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Question 5 Answer

Yes there is, the psychrometric chart that we've been using lists these values as, g of water/ kg of dry air. This is referred to as the humidity ratio W. Some charts list this as grains of moisture; note there are 7000 grains/lbs. Here's how to find the answer using our problem as an example.

1. Locate the MA point on the chart and extend a horizontal line to the right side of the chart and find the MA humidity ratio W
2. Locate the SA point on the chart and extend a horizontal line to the right side of the chart and find the SA humidity ratio W
3. Take the difference and use this modified formula:

g/Hr = CMH x Density (kg/m^3) x ΔW to find the grams of water per hour removed.

The density of air at "standard" conditions (21°C DB dry air at sea level) = 1.2 kg/m^3

THANK YOU



BASIC VAPOUR REFRIGERATION CYCLE

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Agenda

1. Vapour Compression Cycle
2. Pressure – Enthalpy Chart
3. Refrigeration Cycle And Efficiency
4. Improving Coeficient Of Performance
5. Exercise

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VAPOUR COMPRESSION CYCLE

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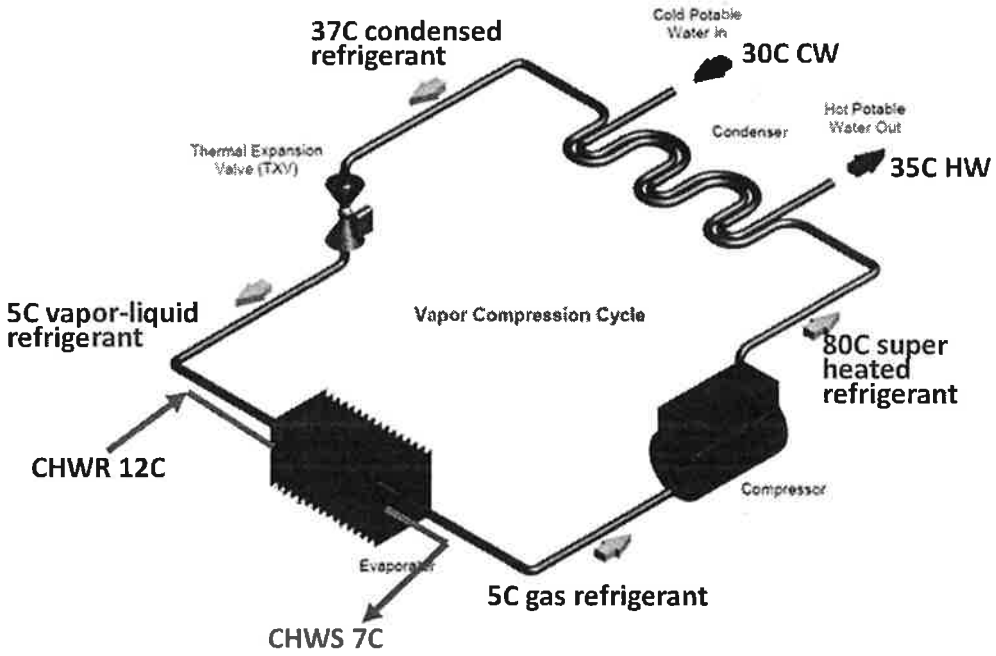
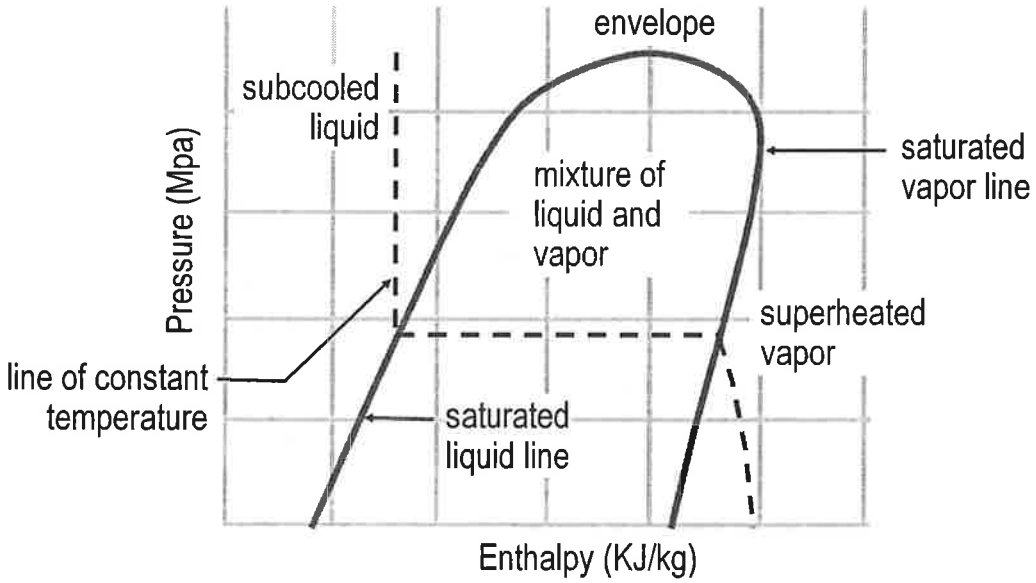
Vapour Compression Cycle

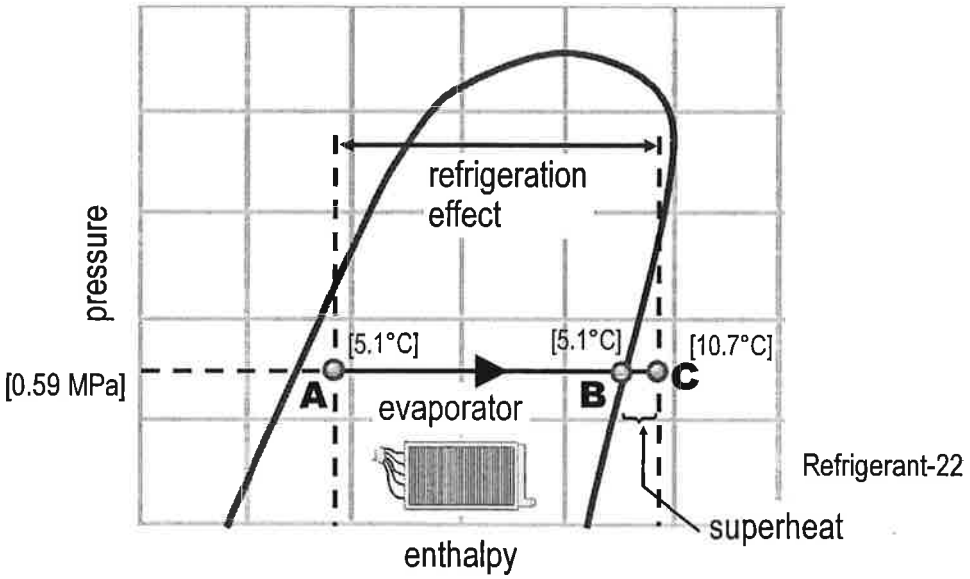


PRESSURE – ENTHALPY CHART

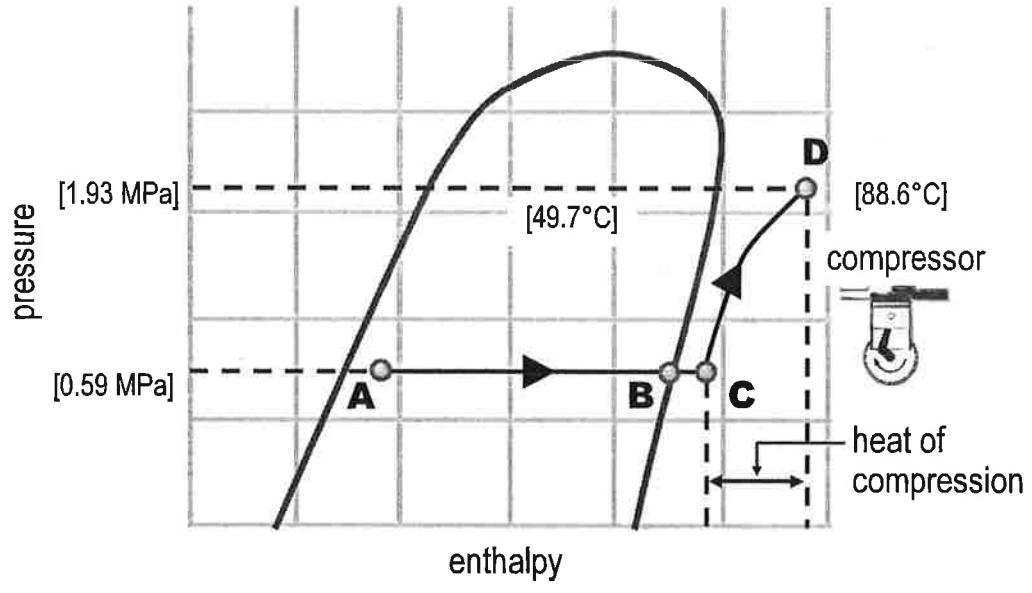
Pressure - Enthalpy (PH) Chart



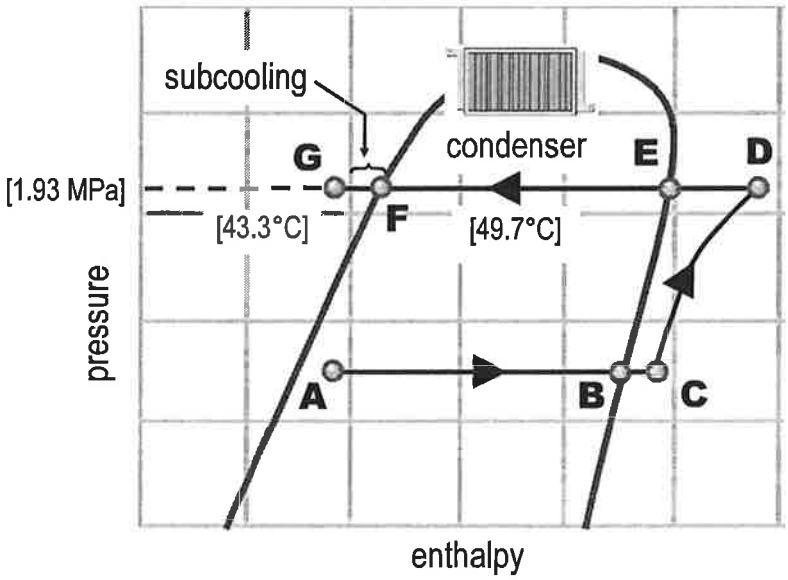
Evaporator



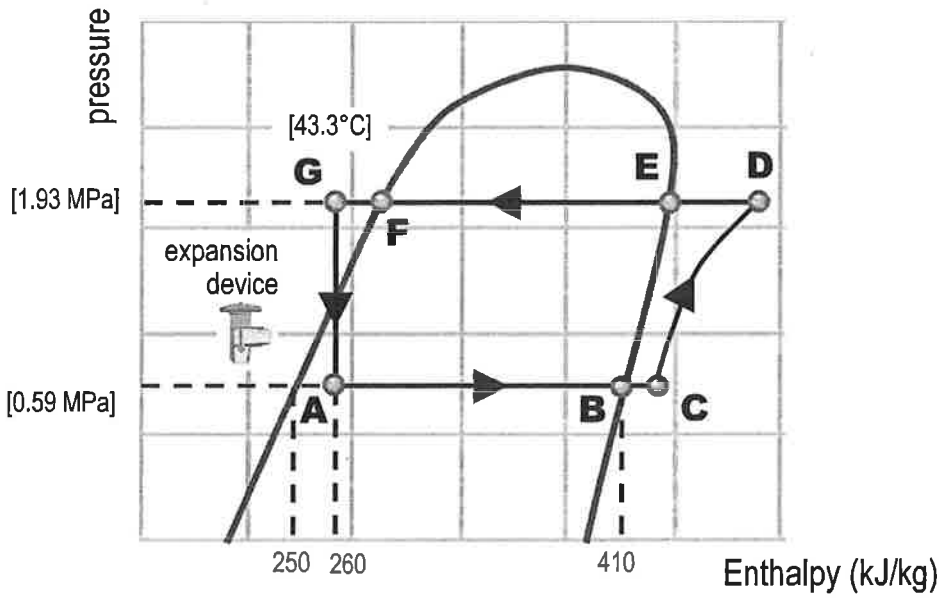
Compressor



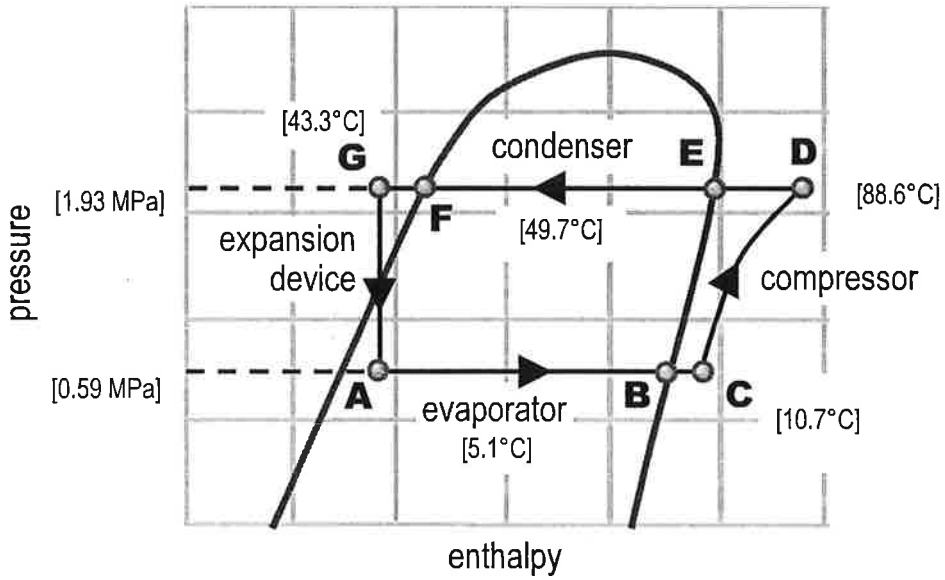
Condenser



Expansion Device



Refrigeration Cycle



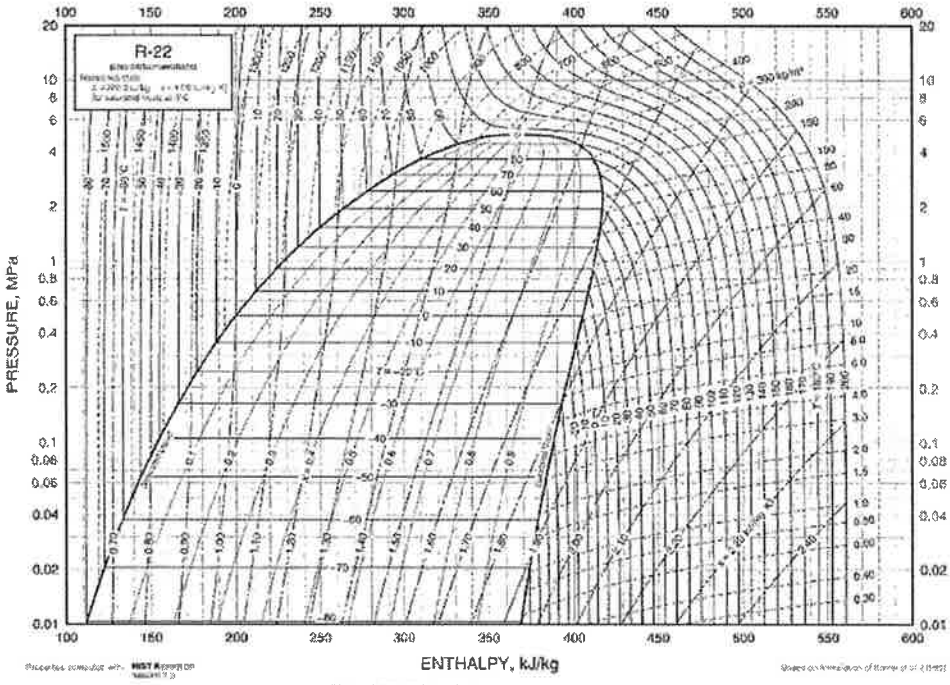


Fig. 2 Pressure-Enthalpy Diagram for Refrigerant 22

30.4

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REFRIGERATION CYCLE AND EFFICIENCY

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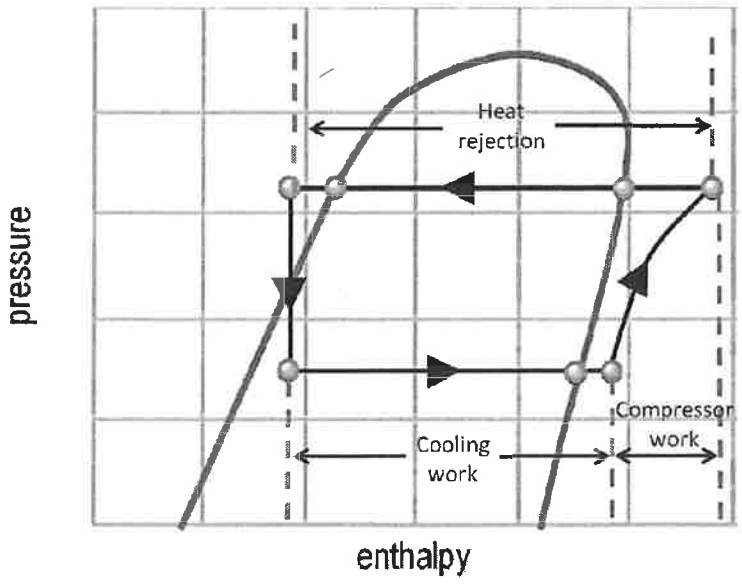
Refrigeration Cycle & Efficiency



Refrigeration Cycle & Efficiency

$$\text{Cooling COP} = \frac{\text{Cooling work (KW)}}{\text{Power input (KW)}}$$

$$\text{Heating COP} = \frac{\text{Heat rejection (KW)}}{\text{Power input (KW)}}$$

$$= \frac{\text{Cooling work(KW) + Power input (KW)}}{\text{Power input (KW)}}$$

$$= \text{Cooling COP} + 1$$

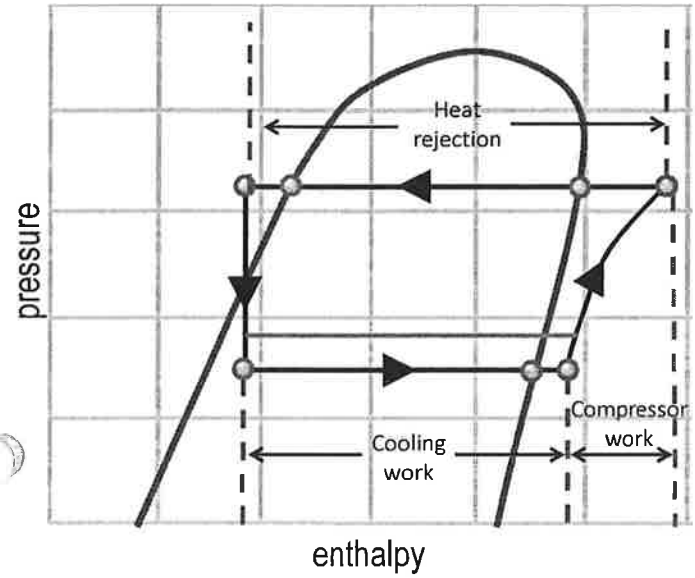
$$\text{Integrated COP} = \text{Cooling COP + Heating COP}$$

$$= 2 \times \text{Cooling COP} + 1$$

Assumes that all heat rejection is recovered for HW heating

IMPROVING COEFICIENT OF PERFORMANCE

Improving Cooling COP



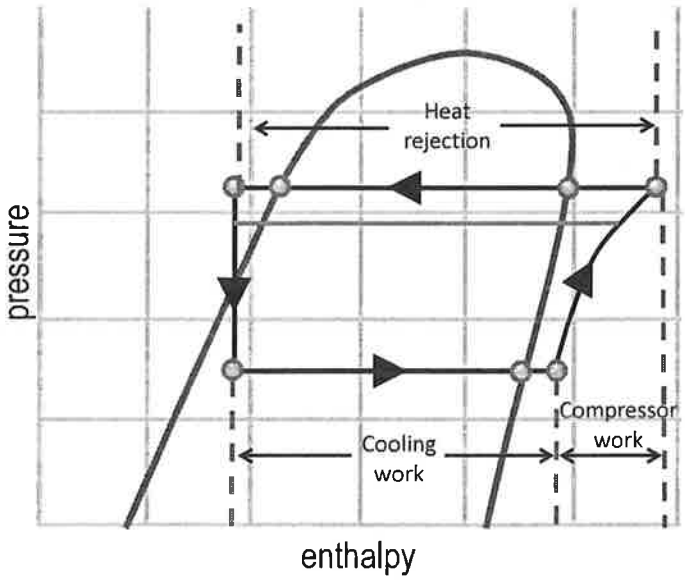
higher chilled water supply temperature

Raise evaporating pressure

Use more efficient evaporator HX which has lower approach temperature

Raise evaporator air or water temp set-pt. But trade off is higher air or water flow rate across evaporator.

Improving Cooling COP



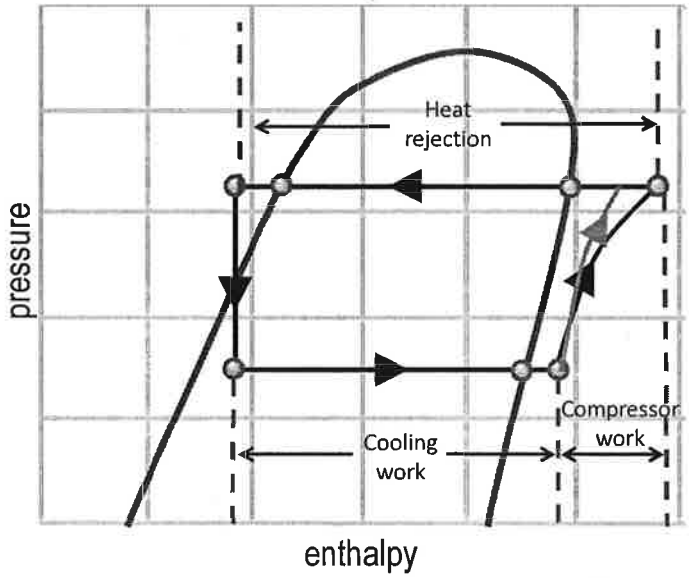
colder condenser temp.

Lower condensing pressure

Use more efficient condenser HX which has lower approach temperature

Lower condenser water supply temperature or increasing condenser water flow rate.

Improving Cooling COP



Reduce compressor power

Use more energy efficient compressor

EXERCISE

Exercise

1) Plot the R22 refrigeration cycle on the PH diagram.

2) Assuming a R22 flow rate of 1kg/s, calculate the

- Cooling capacity
- Heat rejection
- Compressor power
- Cooling COP
- Heating COP
- Integrated COP



Exercise 1

KW = Mass flow (kg/s) x Delta H (KJ/kg)

THANK YOU



Overview of Chilled Water Systems

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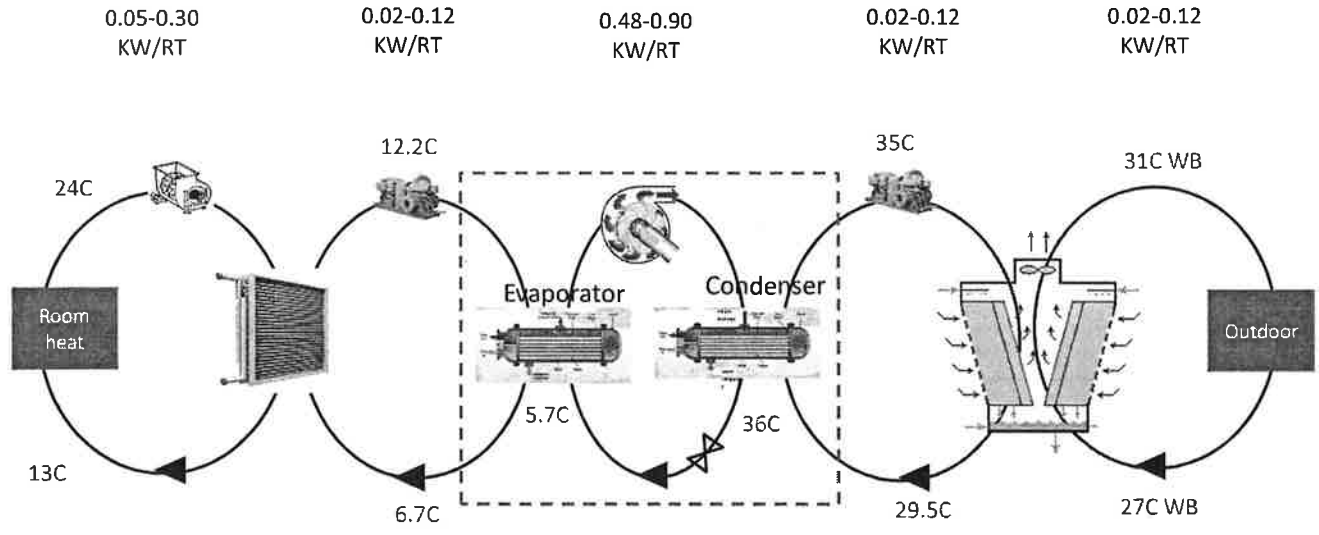
Agenda

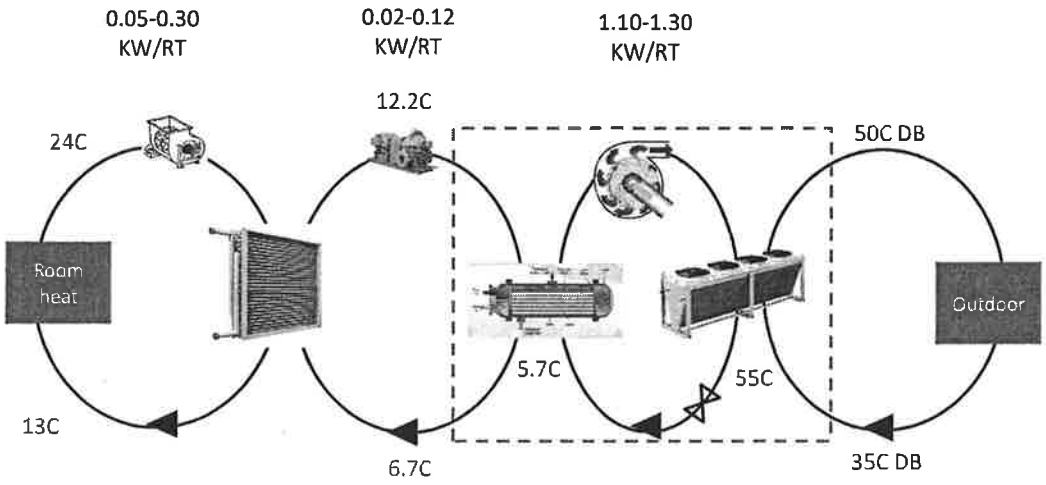
1. Air cooled and water cooled CHW systems
2. Constant primary CHW system
3. Constant primary, variable secondary CHW system
4. Variable primary CHW system
5. Series counter flow CHW system

Air cooled and water cooled CHW systems

Water cooled CHW system



Air cooled CHW system



Water cooled vs Air cooled CHW system

	Water cooled CHW system	Air cooled CHW system
Energy efficiency	High eff (0.60 to 0.75 KW/RT)	Low eff (1.10 to 1.30 KW/RT)
Space requirement	More	Less
First cost	More	Less
Maintenance cost	More	Less
Life span	Approx 15 years	Approx 10 years

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Constant Primary Flow System

Primary constant flow system
3-way valves

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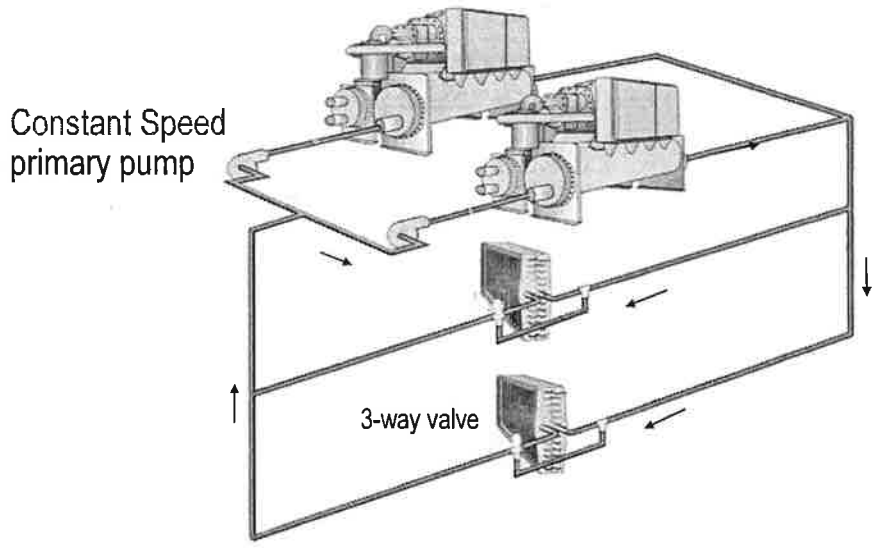
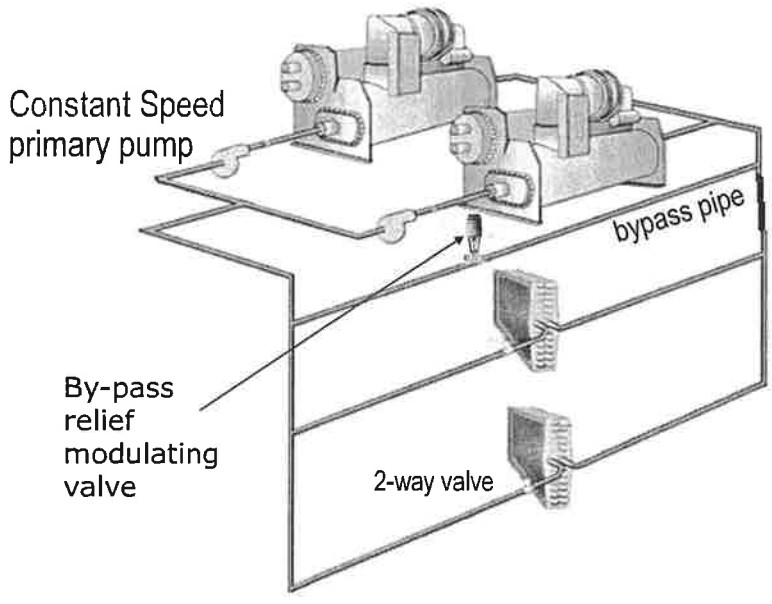


Primary constant flow system
3-way valves

- This system is the typical chw design 20yrs ago. While chw plants are rarely designed this way, many such system are still in operation today.
- This system satisfy a important requirement that chiller manufacturers demand in the past. That is the chw flow through the chiller must be constant whenever it is on operation.
- The pumps are fixed speed and all of them are run whenever any chiller is in operation.
- If a room becomes too cold, the t-stat in the room will command the 3-way valve to by-pass the chw from the coil, thereby reducing its cooling capacity.
- The drawback of this system is that the pump energy is never reduced even though the system may be operating at very low load conditions.

Primary constant flow system
2-way valves



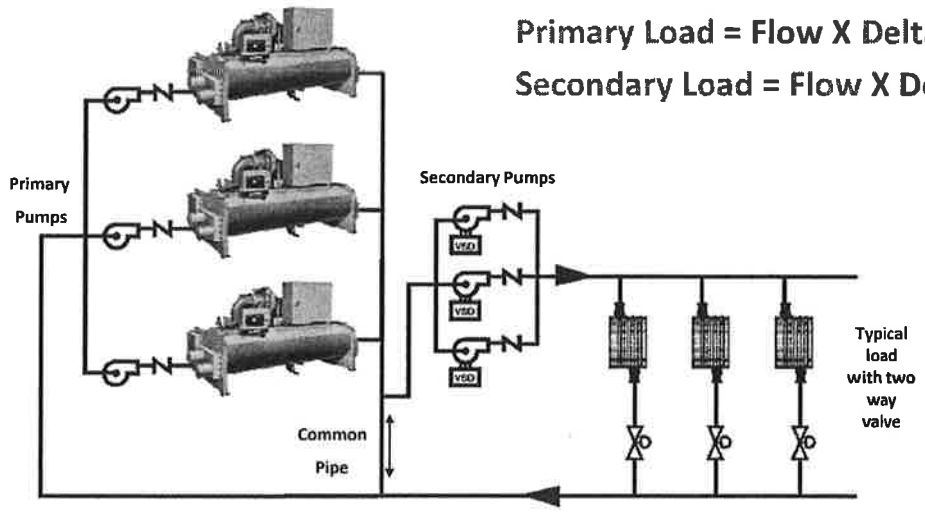
Primary constant flow system
2-way valves

- Later on, a more sophisticated chw system evolved. Note that this system still uses constant speed pumps.
- There are 2 key difference between this and the previous system.
- 1st, 2-way valves are used at the Cooling coils. If a room is too cold, the room T-sat will command the 2-wat valve to close a little to reduce the chw flow into the coil thereby reducing its cooling capacity.
- 2nd difference is the presence of a by-pass pipe and a modulating valve on it. The purpose of the by-pass pipe is to by-pass unwanted chw from the coil back to the chiller. This helps to maintain constant flow through the chiller to satisfy the chiller's constant flow requirement.

- This is system saves some valve and piping cost but does little to save any energy.

Constant Primary, Variable Secondary Flow System

Constant primary, variable secondary flow system



Constant primary, variable secondary flow system

- An even more sophisticated system evolved to save pumping energy. Designers split the chw loop into two parts. The production or primary loop where chw flow needs to be constant and the distribution loop or secondary loop where the flow can be varied to save energy. A by-pass pipe w/o any fittings on it is used to separate or de-couple the primary loop from the secondary loop.
- The key difference is that the secondary pumps have VSDs. They only pump enough chw to meet the cc demand. Any excess chw is by-passed via the by-pass pipe. This system satisfy the need for constant flow through the chillers and yet can conserve significant pumping energy at part load.
- However, it incurs additional capital costs as we need more pumps and space to locate them.

Constant primary, variable secondary flow system

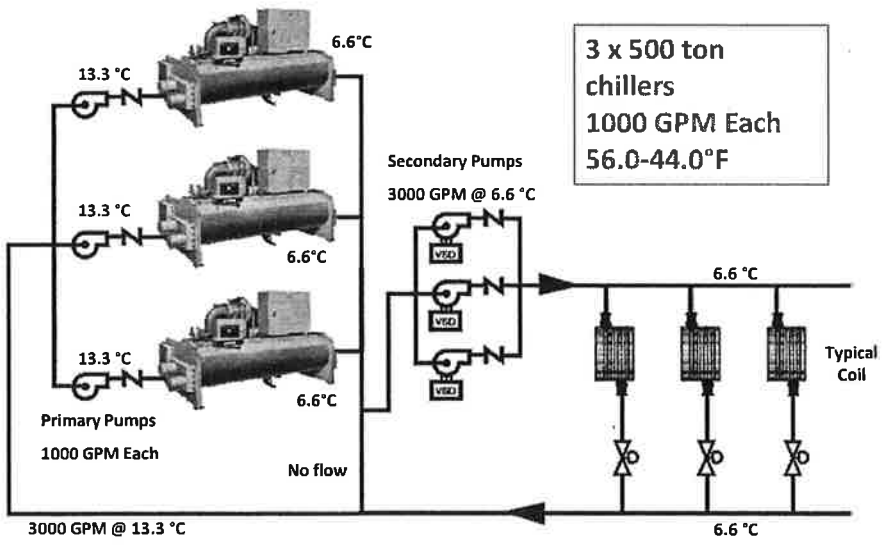
Advantages

- Easier to control (vs Variable Primary Flow (VPF) system)
- Can base load chillers (eg Heat recovery chiller or Absorption chiller)
- Can have secondary pumps of different head.

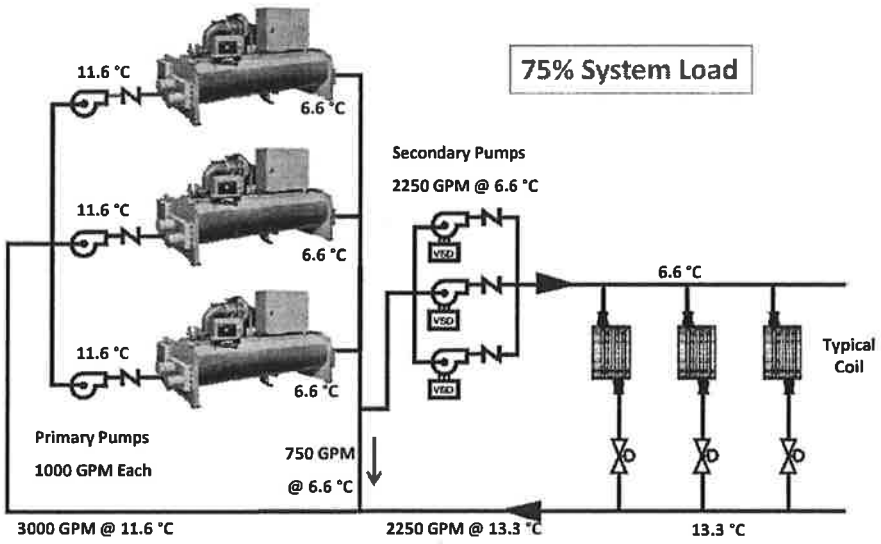
Disadvantages

- Need more space for secondary pumps
- Highest Installed Cost (Sec Pumps, Piping, electricals, etc.)
- Higher pressure losses due to more fittings (vs VPF)
- Less energy efficient because of constant primary flow (vs VPF)
- Less energy efficient because of lower Delta T (vs VPF)

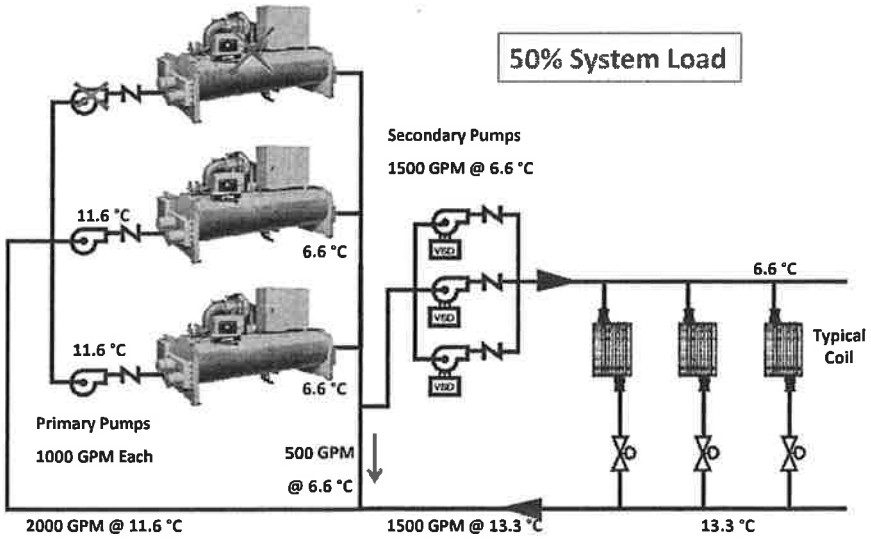
Constant primary, variable secondary flow system



Constant primary, variable secondary flow system

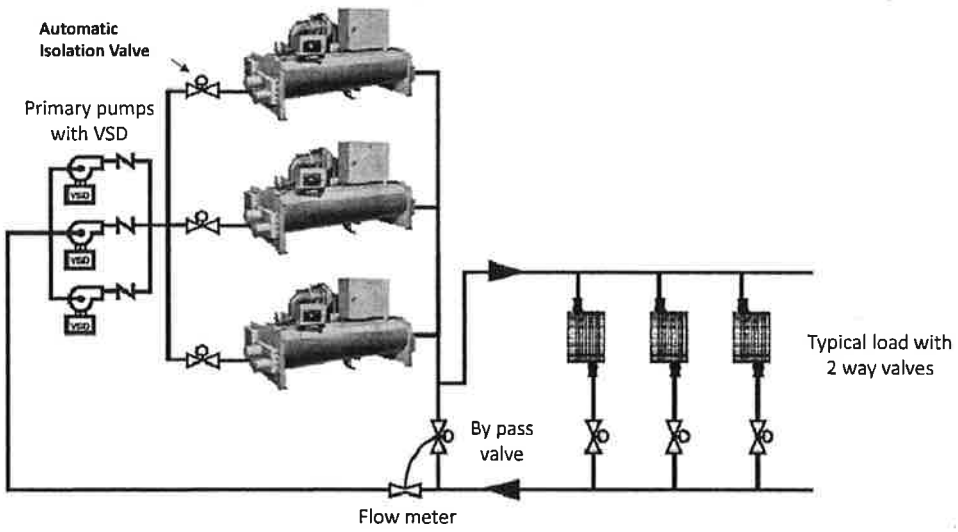


Constant primary, variable secondary flow system



Variable Primary Flow (VPF) System

Variable Primary Flow (VPF) System



Variable Primary Flow (VPF) System

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Advantages

- Lower Installed Cost (Usually by 5%)
  - No secondary Pumps or piping, valves, electrical, installation, etc.
- Less Plant Space Needed
- Better CHW Pump Energy Efficiency
  - Higher CHW DT
  - Lower Pump Design Head

Disadvantages

- Need more sophisticated controls and operators

Variable Primary Flow (VPF) System

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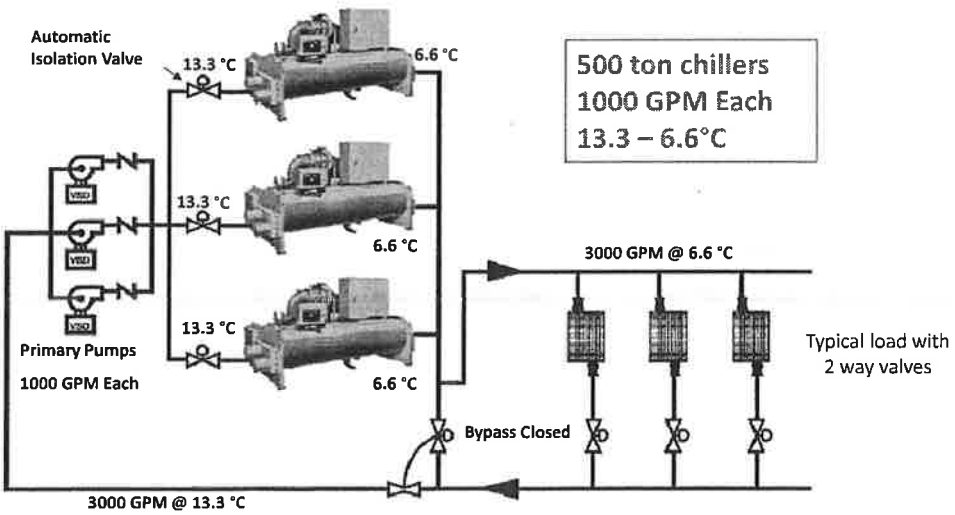
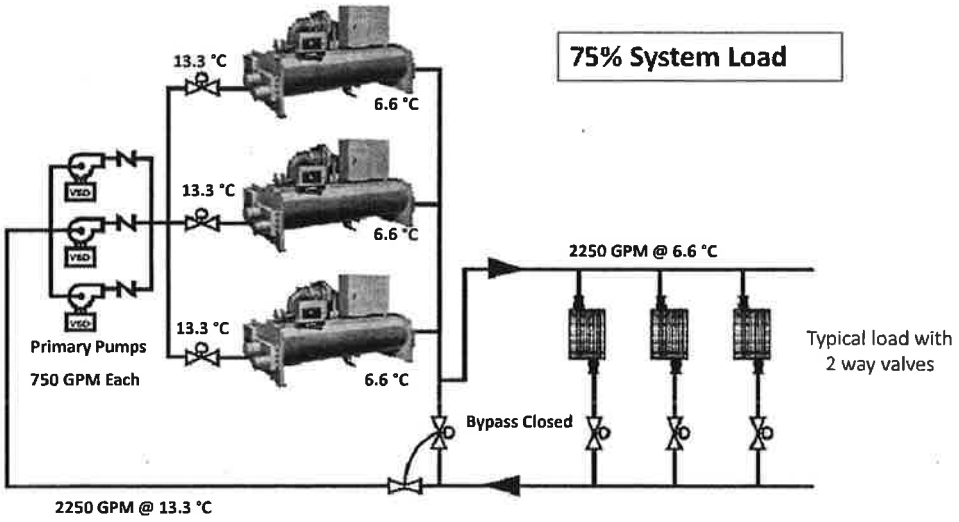
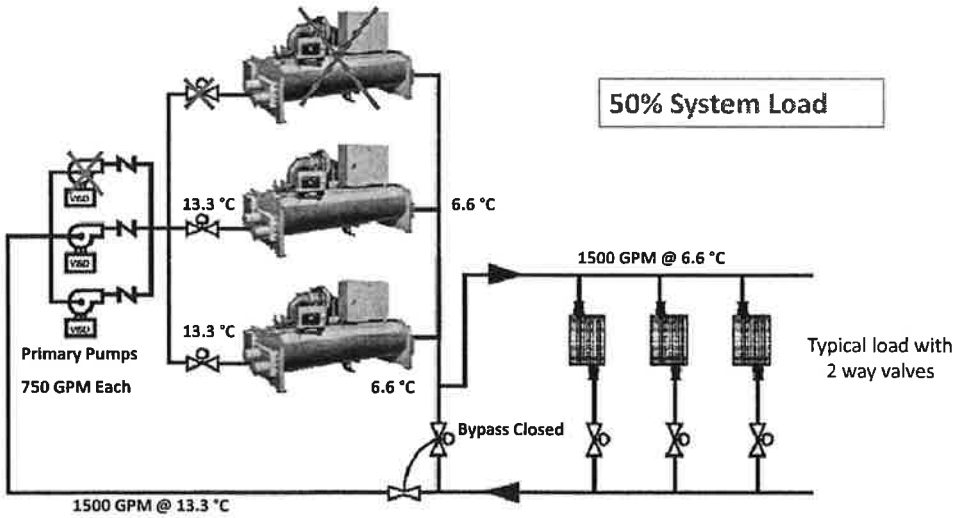


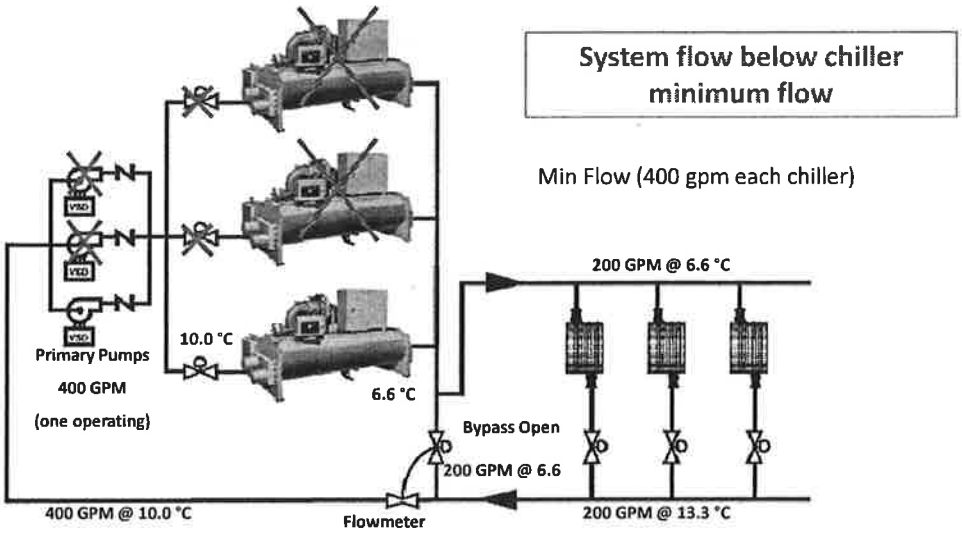
Variable Primary Flow (VPF) System



Variable Primary Flow (VPF) System

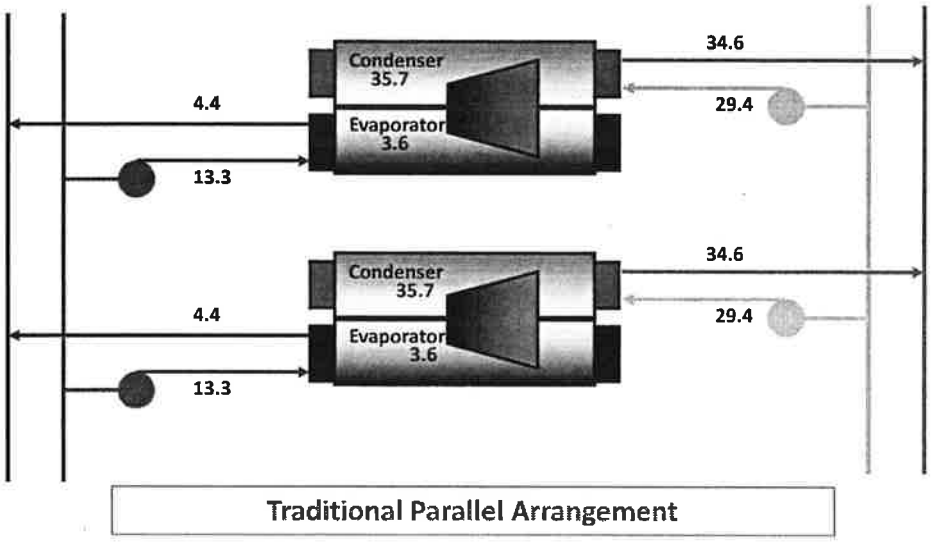


Variable Primary Flow (VPF) System



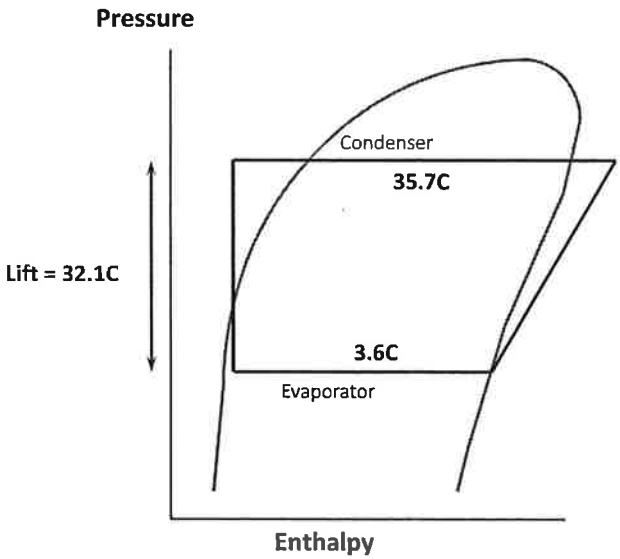
Series Counter Flow System

Series Counter Flow System



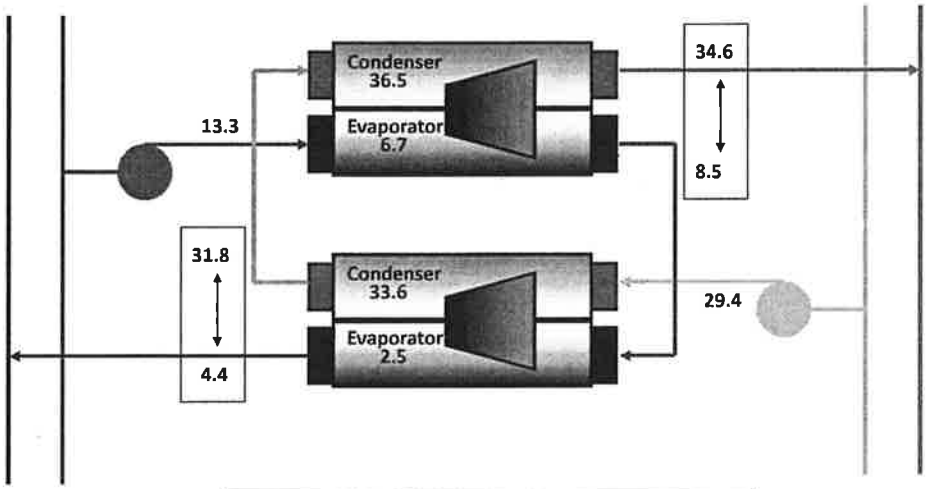
Traditional Parallel Arrangement

Series Counter Flow System



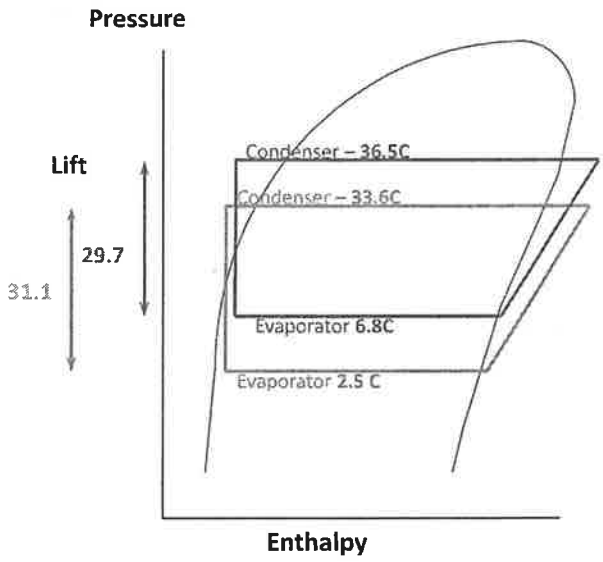
Installing chillers
in parallel
(single lift operation)
requires each unit
to produce the
total system lift

Series Counter Flow System



Series-Counter flow Arrangement

Series Counter Flow System



Installing Chillers in series-counterflow reduces the lift requirements for both units.

Series counter flow system

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Advantages

- Higher chiller efficiency due to lower compressor lift

Disadvantages

- Higher pressure drop across the series evaporators and condensers.
  - Compensate by designing higher CHW and CW DTs and variable flow
- Each capacity step is larger
  - Each capacity step is now 2 series chillers instead of 1 for parallel chillers.
  - More suited for very large loads or very constant loads.
- More complicated piping and controls

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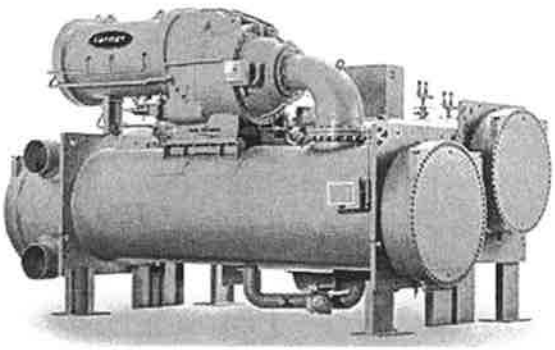
THANK YOU

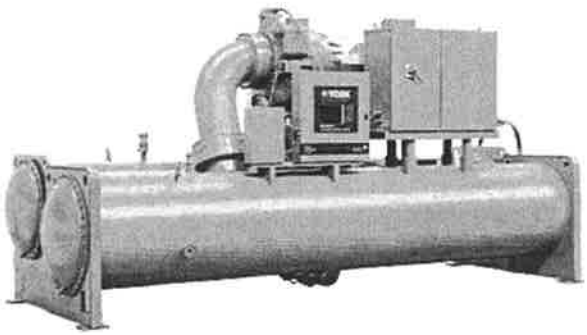


CHILLERS

Steven Kang
Steven.kang@mnv.com.sg

Chiller Technologies





Agenda

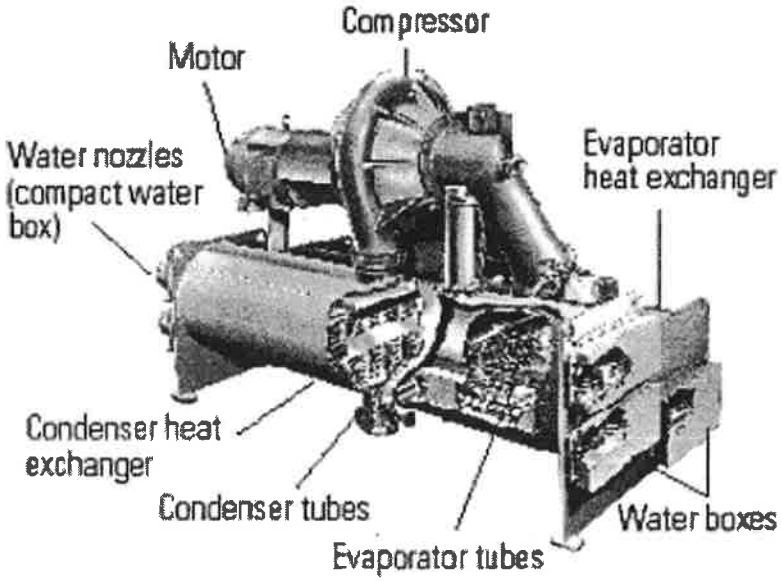
Measurement & Verification Pte Ltd

1. Overview of chillers
2. Compressors
3. Chiller Performance Range
4. Chiller Part Load Performance
5. Refrigerants

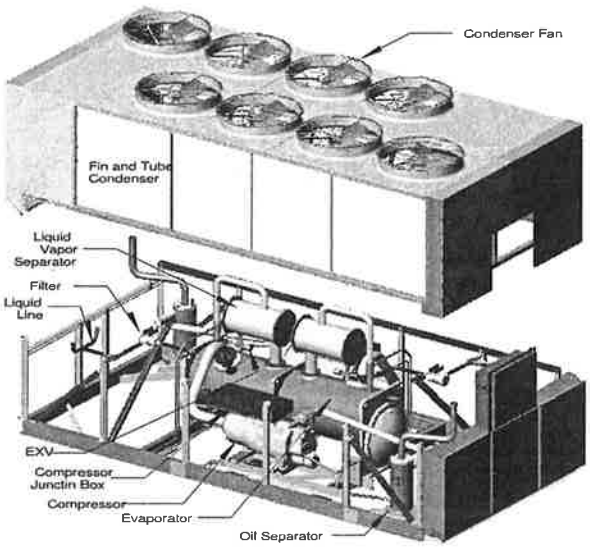
Measurement & Verification Pte Ltd

Overview of chillers

Water cooled chiller

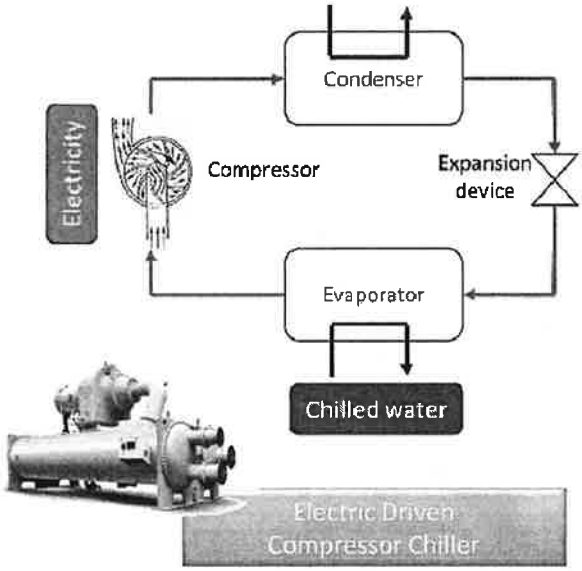


Air cooled chiller

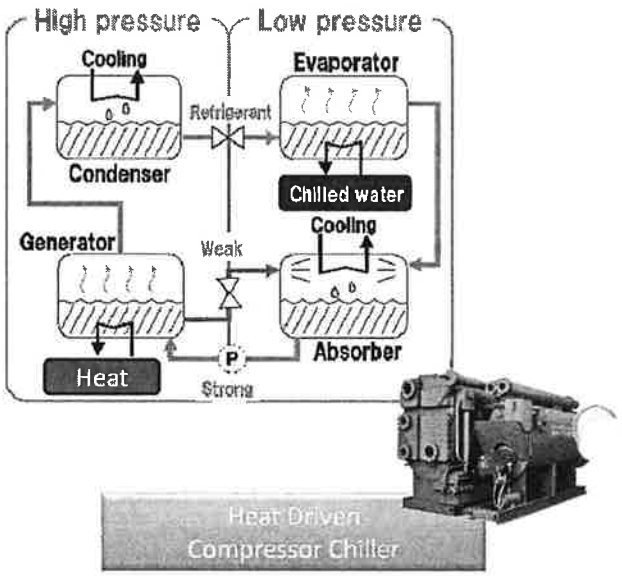


Proprietary Information of Measurement & Verification Pte Ltd

Absorption chiller



Electric Driven Compressor Chiller



Heat Driven Compressor Chiller

Absorption chiller



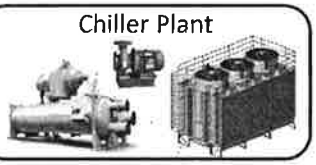
Natural Gas

COP 0.55



Power Plant

COP 6.00





CT make-up water

8.8 L/H per RT

Overall COP 3.30



7C chilled water

Absorption chiller makes economic sense when there is waste heat (eg co-gen)



Natural Gas

COP 0.94





CT make-up water

11.7 L/H per RT

Overall COP 0.94



7C chilled water

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COMPRESSORS

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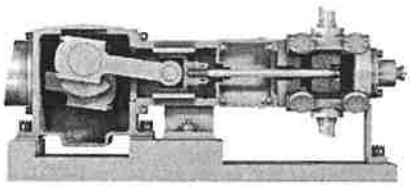
Compressor Types

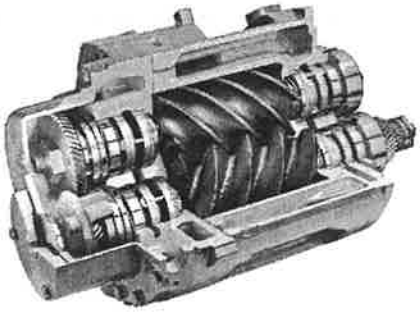
Reciprocating

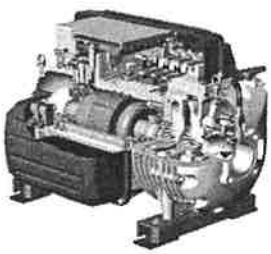




Scroll

Helical-rotary





Centrifugal

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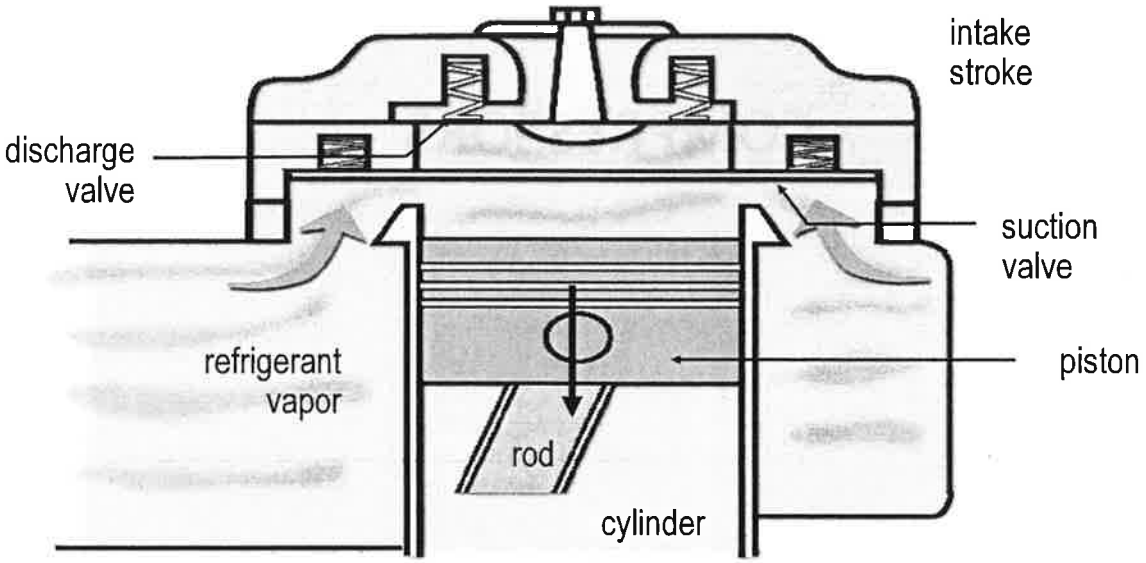
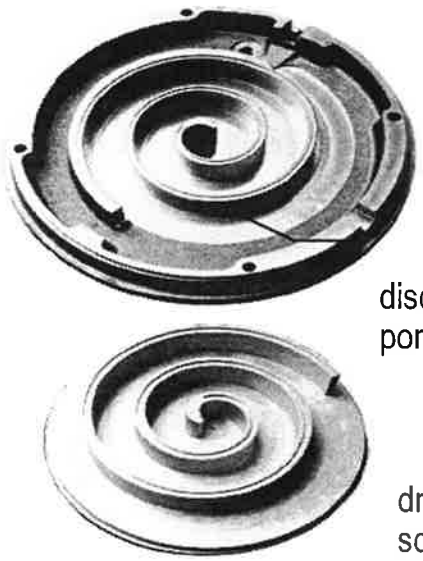
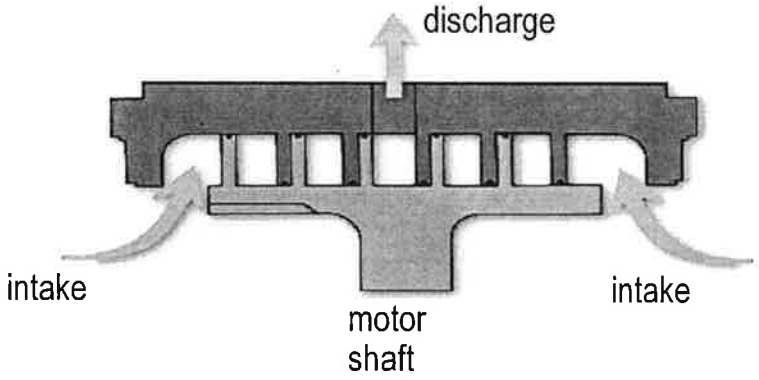
Reciprocating Compressor



Scroll Compressor





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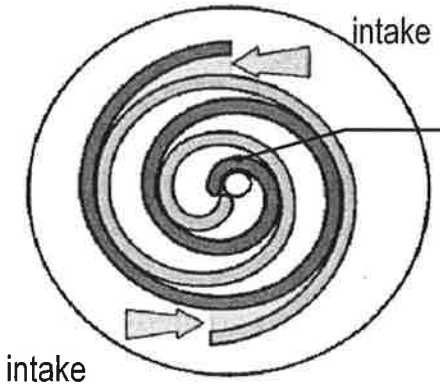
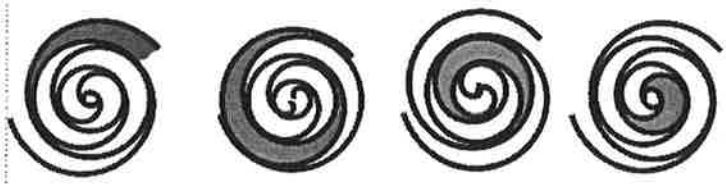
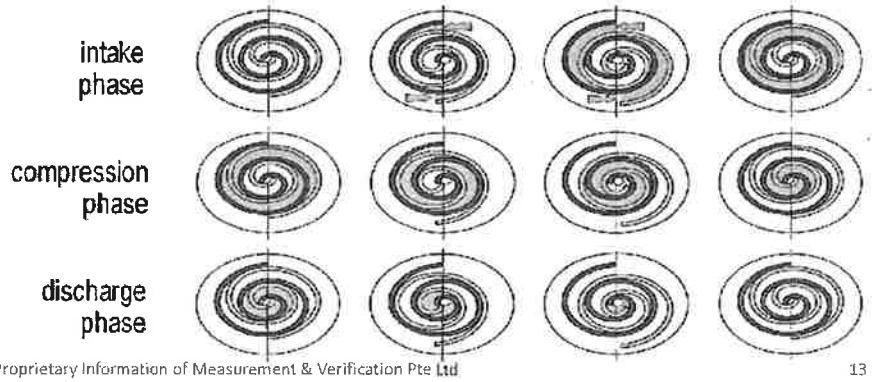
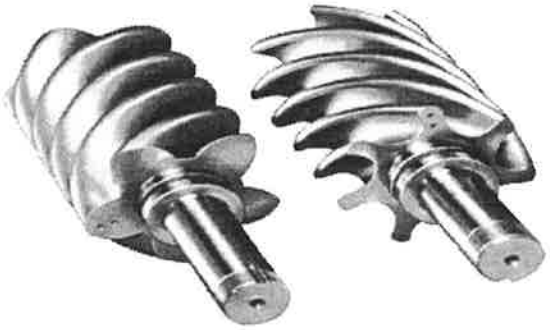
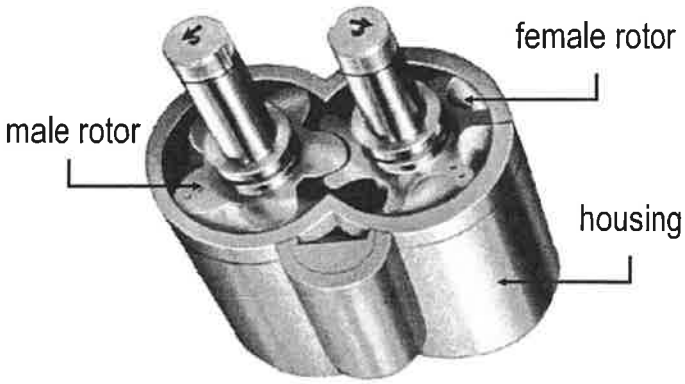




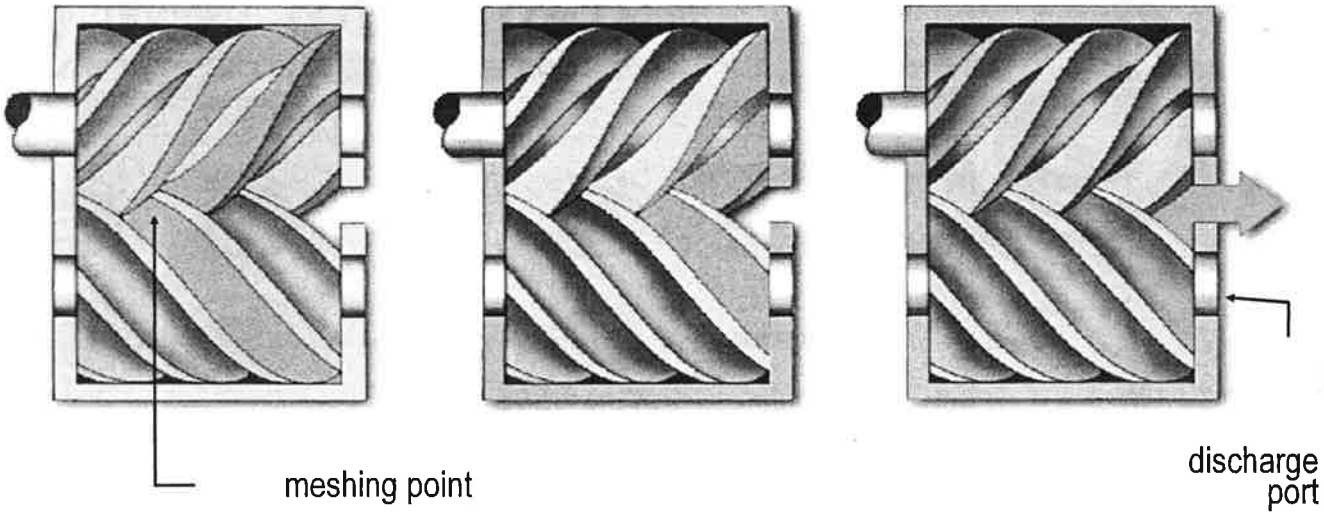


Helical-Rotary Compressor

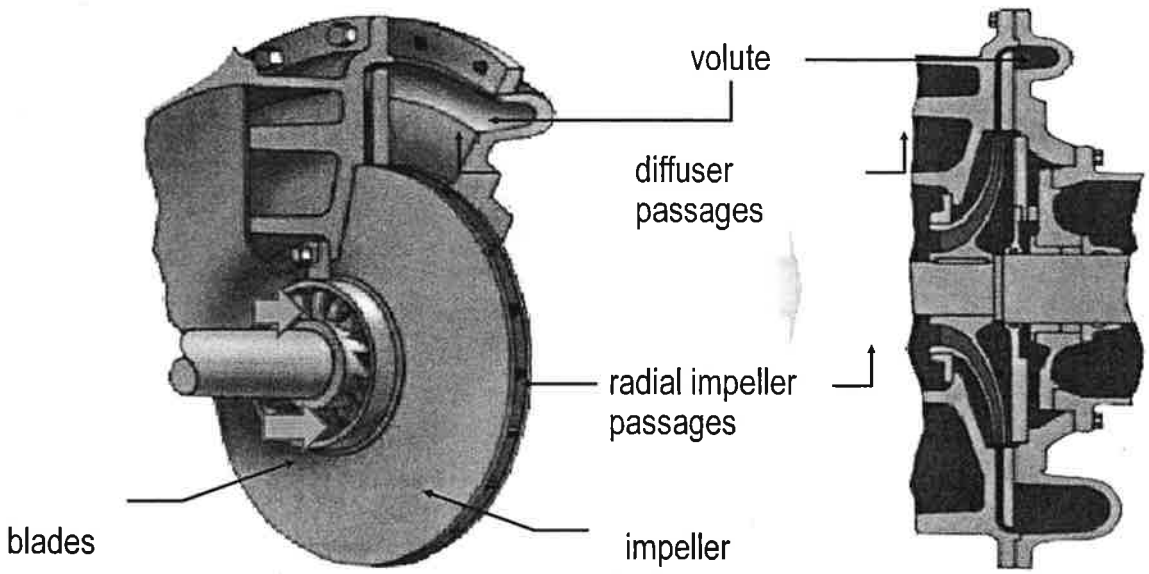




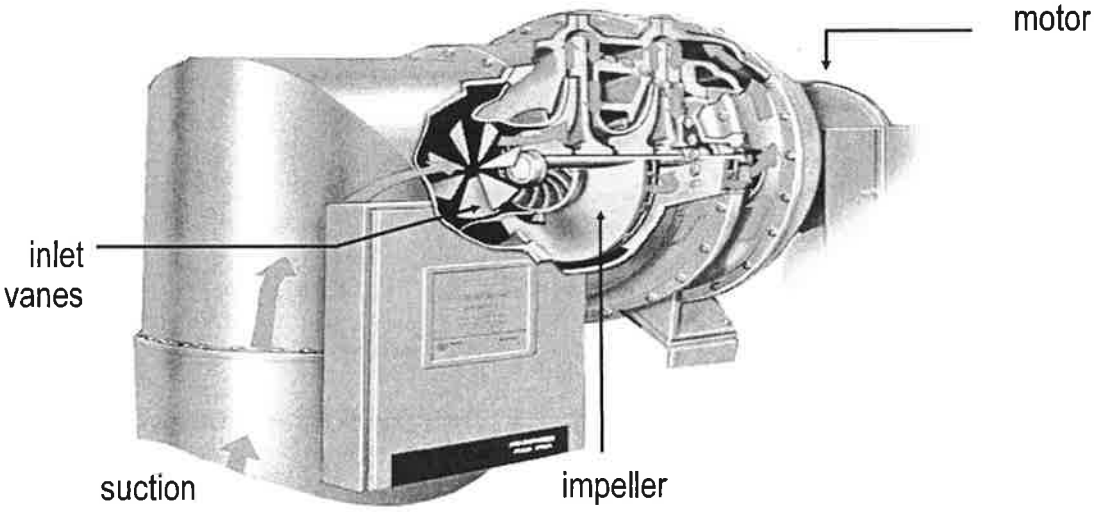
Helical-Rotary Compressor



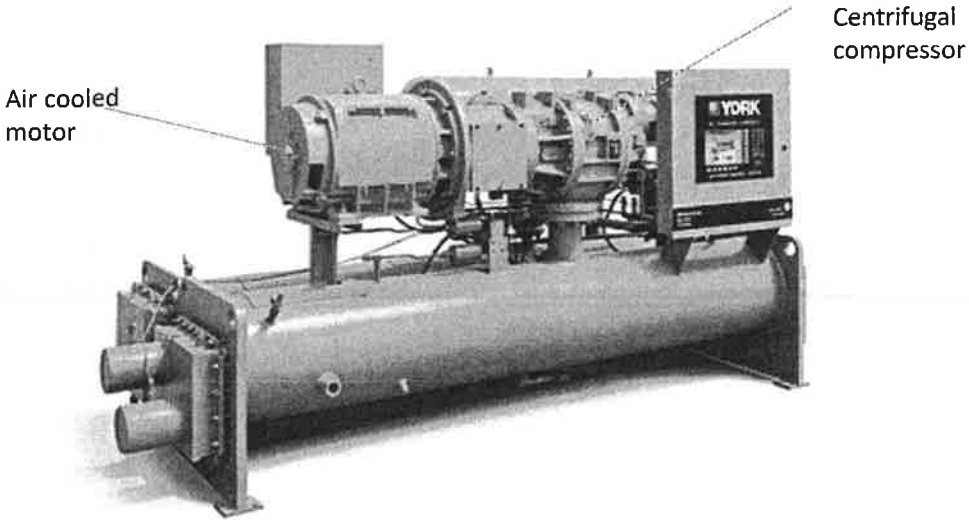
Centrifugal Compressor



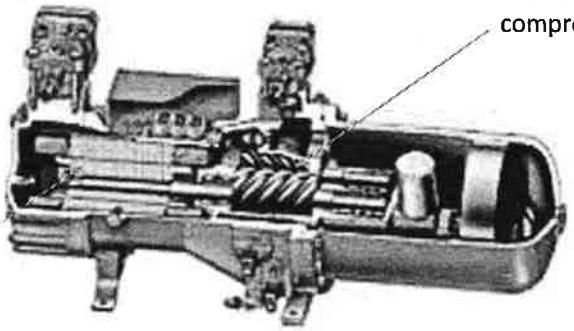
Centrifugal Compressor



Open Compressor

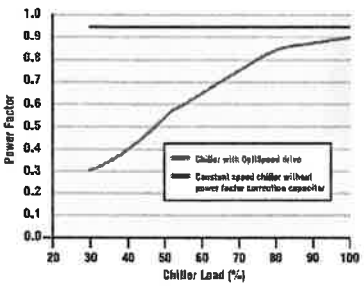


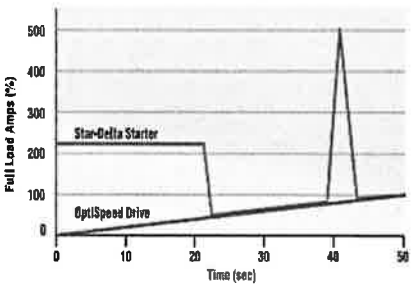
Hermetic Compressor



VSD compressors

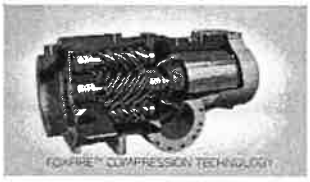
- Higher part load efficiency (vs constant speed)
- Lower inrush current
- High power factor







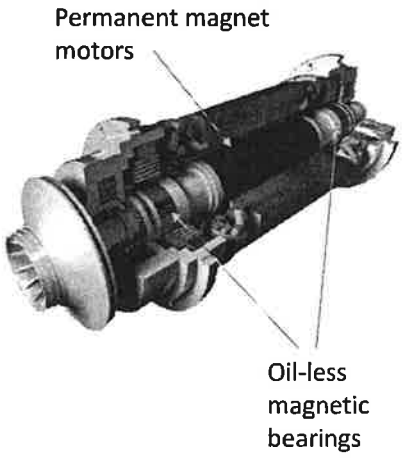
VSD Centrifugal compressor



VSD Screw compressor

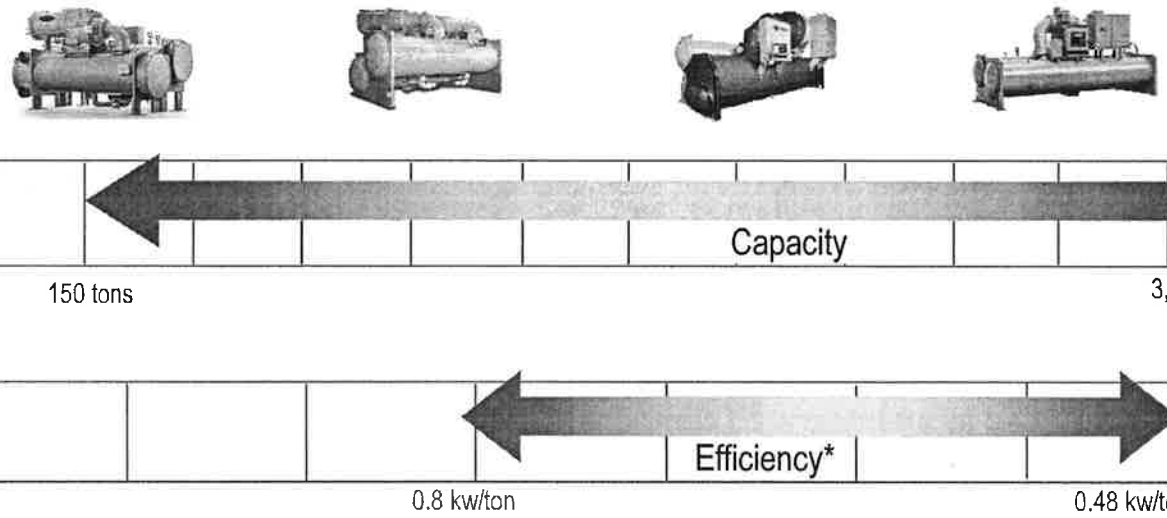
Oil-less magnetic bearing chillers

- Magnetic bearing design eliminates mechanical seals, wear surfaces, gears and slide valves for longer machine life and increased reliability.
- Permanent magnet motors offers higher efficiency
- Oil-free design eliminates oil management systems for improved compressor and system reliability. The oil-free design also eliminates the possibility of efficiency-robbing oil contamination of heat-transfer surfaces.
- VFD designed as an integrated component with the compressor. Reduced in-rush.



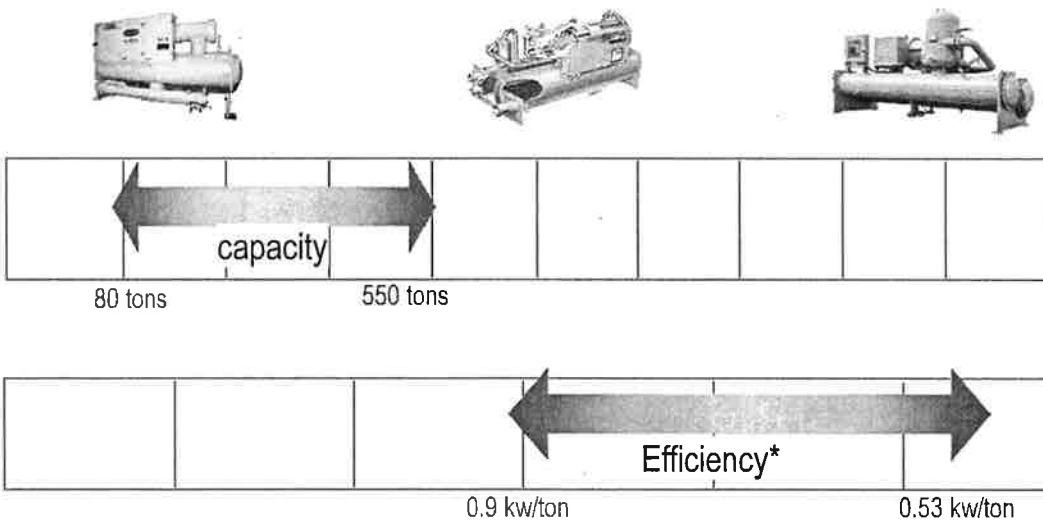
CHILLER PERFORMANCE RANGE

Water Cooled Centrifugal Chiller Performance



* Full Load efficiency rated at standard ARI conditions

Water Cooled Screw Chiller Performance

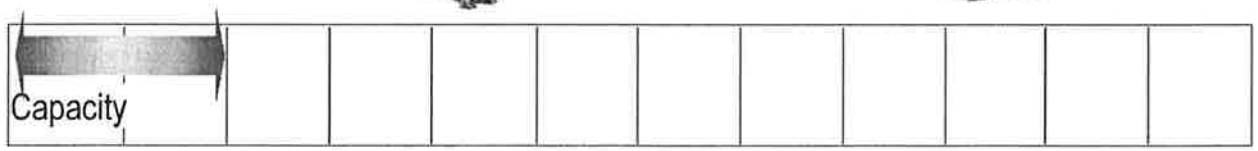


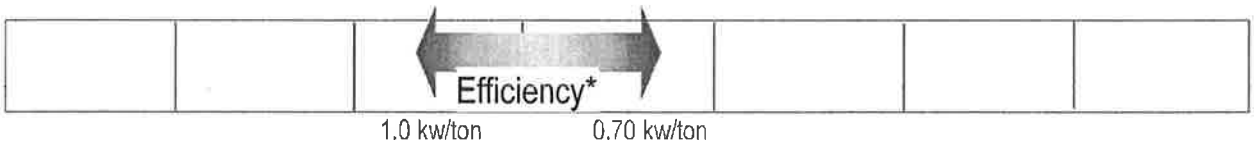
* Full Load efficiency rated at standard ARI conditions

Water Cooled Scroll Chiller Performance









* Full Load efficiency rated at standard ARI conditions

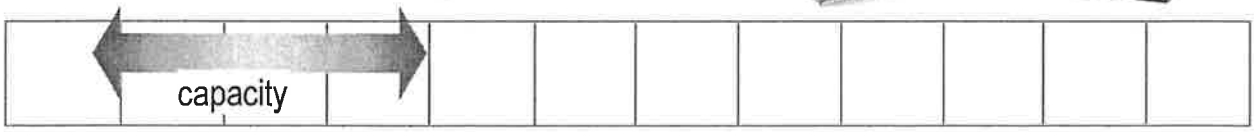
Air Cooled Screw Chiller Performance











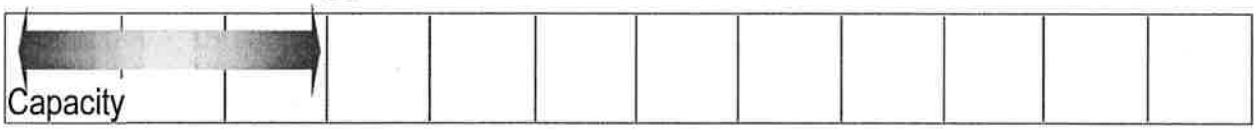


* Full Load efficiency rated at standard ARI conditions

Air Cooled Scroll Chiller Performance





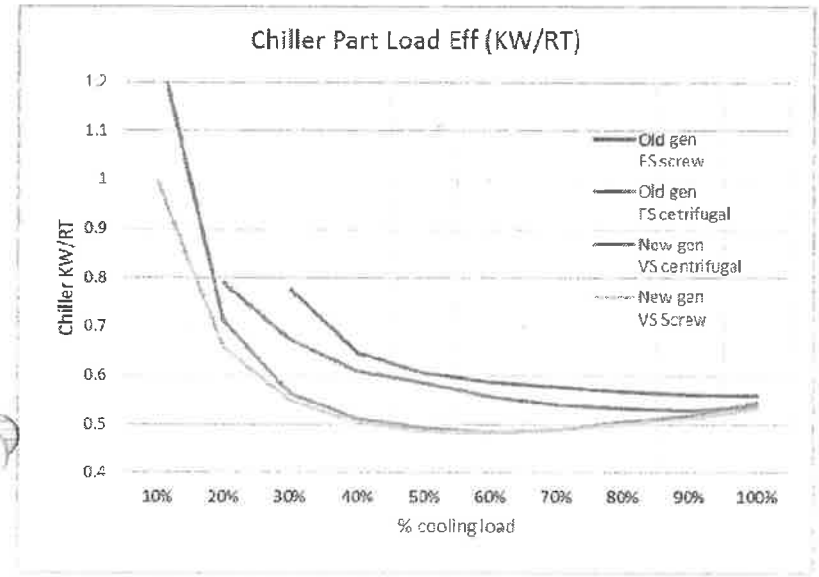




* Full Load efficiency rated at standard ARI conditions

CHILLER PART LOAD PERFORMANCE CURVE

W/C Chiller part load performance



<u>Old gen fixed speed chillers (R134a)</u>
Peak eff: ~ 0.530 to 0.54 KW/RT
Peak eff point: 80-90% load

<u>New gen variable speed chillers (R134a)</u>
Peak eff: ~ 0.49 to 0.50 KW/RT
Peak eff point: 50-60% load

Performance rated at
CHW: 6.70C at fixed 2.4gpm/RT
CW: 29.4C at fixed 3.0 gpm/RT

Chiller performance are indicative and do not represent any particular brand.

Microsoft Excel Worksheet

REFRIGERANTS

GREEN MARK REQUIREMENT



Part 3- Environmental Protection

Refrigerants

Reduce the potential damage to the ozone layer and the increase in global warming through the release of ozone depleting substances and greenhouse gases. (1 point)

(a) Refrigerants with ozone depletion potential (ODP) of zero or with global warming potential (GWP) of less than 100.
(b) Use of refrigerant leak detection system at critical areas of plant rooms containing chillers and other equipments with refrigerants.

ODP and GWP Level of Refrigerants

Refrigerant Number	Chemical Name	Chemical Formula	Molecular Mass	Safety Group	Atmospheric Lifetime (Yrs)	ODP	GWP
11	trichlorofluoromethane	CCl_3F	137.4	A1	50	1	3800
12	dichlorodifluoromethane	CCl_2F_2	120.9	A1	102	1	8100
22	chlorodifluoromethane	$CHClF_2$	86.5	A1	12.1	.055	1500
32	difluoromethane	CH_2F_2	52	A2	5.6	0	650
123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$	153	B1	1.4	.02	90
125	Pentafluoroethane	CHF_2CF_3	120	A1	32.6	0	2800
134a	1,1,1,2-tetrafluoroethane	CF_3CH_2F	102	A1	14.6	0	1300
245fa	1,1,2,2,3-Pentafluoropropane	$CHF_2\ CH_2\ CF_3$	134.05	B1	8.8	0	820
290	Propane	$CH_3CH_2CH_3$	44	A3	<1	0	~0
404a	R-125/143a/134a (44/52/4)			A1			3260
407C	R-32/125/134a (23/25/52)			A1		0	1530
410A	R-32/125 (50/50)			A1		0	1730
500	R-12/152a (73.8/26.2)			A1		.74	6010
507a	R-125/143a (50/50)			A1			
600	Butane	$CH_3CH_2CH_2CH_3$	58.1	A3	<1	0	~0
717	Ammonia	NH_3	17	B2	N/A	0	0
718	Water	H_2O	18	A1	N/A	0	<1
744	Carbon dioxide	CO_2	44	A1	N/A	0	1

Theoretical Efficiency of R-123 Vs. R-134a

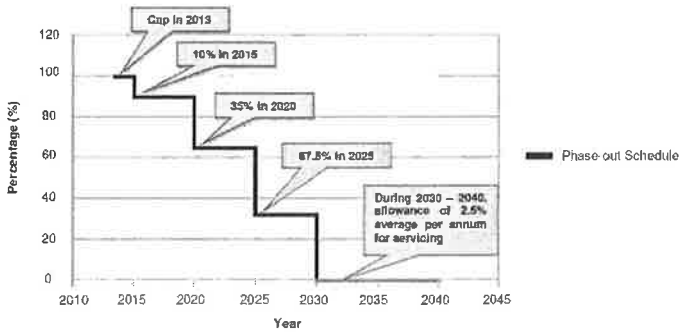
Model	Carnot	Ideal Cycle	
Property		R-123	R-134a
Chilled water Supply Temp (°F)	44	44	44
Cond. Water Supply Temp (°F)	85	85	85
Compressor Inlet Temp (°F)	44	44	44
Compressor Outlet Temp (°F)	95	95	95
Suction Line Press. Drop (°F)	N/a	0	0
Liq. Line Press. Drop (°F)	N/a	0	0
Evaporator Superheat (°F)	N/a	0	0
Condenser Subcooling (°F)	N/a	0	0
Isentropic Compressor Eff.	N/a	100%	100%
Compressor Motor Eff.	N/a	100%	100%
COP	10.88	8.949	8.472
KW/ton	0.323	0.393	0.415

Singapore HCFC Phase out plan

Why is there a cap on production and import of HCFCs in Singapore?

At the 19th Meeting of the Parties to the Montreal Protocol held in Sep 2007, Parties agreed to adjust their commitments to accelerate the phase out of HCFCs. Article-5 Parties, which include Singapore are obliged to freeze the production and consumption of HCFCs in 2013, followed by subsequent reduction from 2015 to 2030.

Singapore does not produce HCFCs. All HCFCs used in Singapore are imported from overseas. The use of HCFCs for local industries and end-users is still allowed till 2030, subject to the following stepwise phase-out schedule:



Source: http://www.nea.gov.sg/docs/default-source/anti-pollution-radiation-protection/chemical-pollution/mea-ods/circular-on-hydrochlorofluorocarbons-(hcfcs)-phase-out-management-plan-nbsp-in-singapore.pdf?sfvrsn=0

Montreal Protocol HCFC phase out plan for Non-Article 5 countries

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Comparison of the Montreal Protocol and United States Phaseout Schedules

Montreal Protocol		United States	
Year to be Implemented	% Reduction in Consumption and Production, Using the Cap as a Baseline	Year to be Implemented	Implementation of HCFC Phaseout through Clean Air Act Regulations
2004	35.0%	2003	No production and no importing of HCFC-141b
2010	75.0%	2010	In addition to the HCFC-141b restrictions, no production and no importing of HCFC-142b and HCFC-22, except for use in equipment manufactured before 1/1/2010 (so no production or importing for NEW equipment that uses these compounds)
2015	90.0%	2015	In addition to the HCFC-141b, HCFC-142b and HCFC-22 restrictions, no production and no importing of any other HCFCs, except for use as refrigerants in equipment manufactured before 1/1/2020
2020	99.5%	2020	No production and no importing of HCFC-142b and HCFC-22
2030	100.0%	2030	No production and no importing of any HCFCs

Next generation refrigerants

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Next Generation Refrigerants:

High COP

Low GWP

Near zero ODP

Low toxicity and low flammability

Comparable Cost (vs HFCs and HCFCs)

http://www.achrnews.com/articles/116977-is-the-next-generation-refrigerant-here

THANK YOU

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PUMPS

Steven Kang
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Agenda

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1. Overview of pumps
2. Pumps types
3. Pump Curves, Operating Point & System Curve
4. Efficiency

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Overview of pumps

Pumps for chilled water plant

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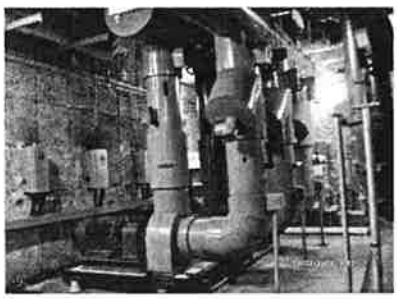

Chilled Water Pump



Condenser Pump



Cooling tower make up water pump



Water treatment dosing pump

Pumps are everywhere in a building

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- Chilled water pumps
- Condenser water pumps
- Cooling tower make up water pumps
- Cooling tower water treatment pumps
- Condensate water pumps (collection of condensate)
- Domestic cold water pumps
- Domestic hot water pumps
- Booster pumps
- Basement sump pumps

World pumping energy

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- American Hydraulic Institute's research indicates that electric motors in pump systems consume 13% of the world's total electricity output, and that many pump systems run at 60% efficiency or less.

http://www.engineeringnews.co.za/article/heavy-industry-urged-to-improve-pump-energy-efficiency-2008-10-03

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PUMP TYPES

Pumps in HVAC

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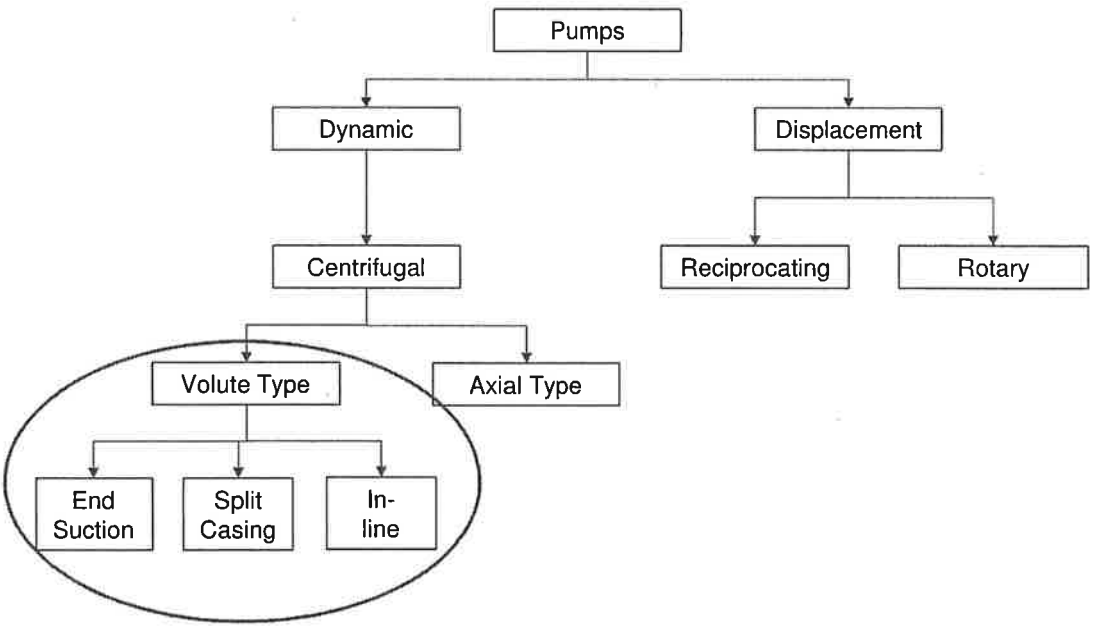

End suction centrifugal pump

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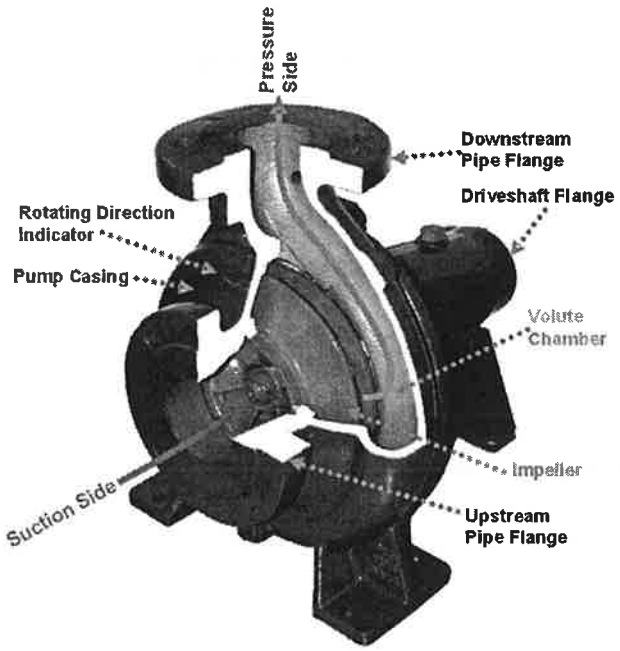
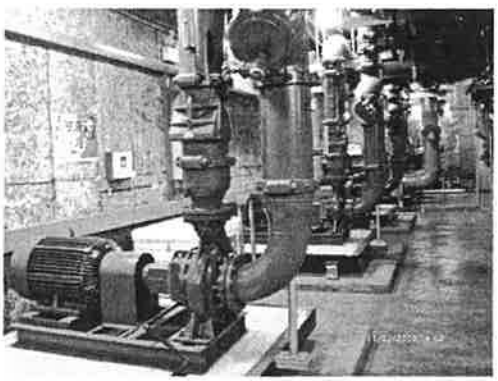
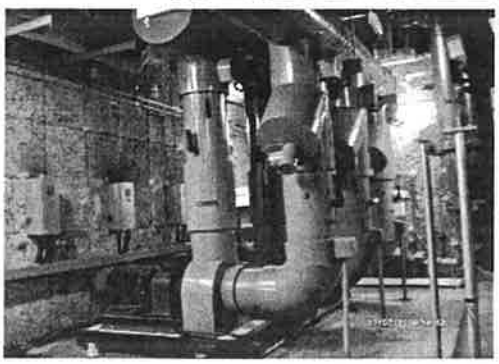
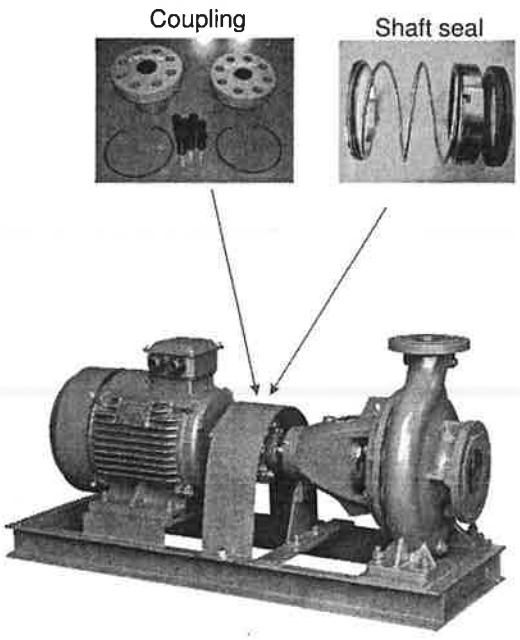


End suction centrifugal pump

Measurement & Verification Pte Ltd







End suction centrifugal pump

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Performance Range

- Capacity: 2 to 1400m³/h 388 l/s
- Head: 2 to 115m
- Efficiency : up to 90%
- Impeller Dia: 125-381mm
- Motor Power: 0.18-200kW
- Motor speed: 2, 4, 6pole
- Frequency: 50Hz, 60Hz
- Max. working pressure: 1.2MPa

Component Material

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Component	Material Combination				
	Standard	Option	Option	Option	Option
Volute	Cast Iron	Cast Iron	Ductile Iron	Ductile Iron	Bronze
Impeller	304SS	Bronze	304SS	Bronze	Bronze
Seal Housing	Cast Iron	Cast Iron	Ductile Iron	Ductile Iron	Bronze
Backplate	Cast Iron	Cast Iron	Ductile Iron	Ductile Iron	Bronze
Shaft	420SS	Carbon Steel	420SS	Carbon Steel	316SS
Case Wear Ring	Bronze	Bronze	Bronze	Bronze	Bronze
Bearing Frame	Cast Iron	Cast Iron	Cast Iron	Cast Iron	Cast Iron
Slinger	Rubber	Rubber	Rubber	Rubber	Rubber

Split casing centrifugal pump

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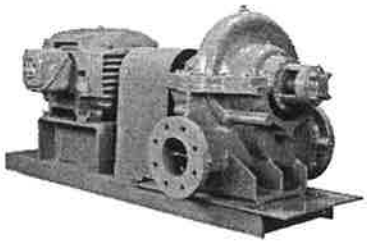
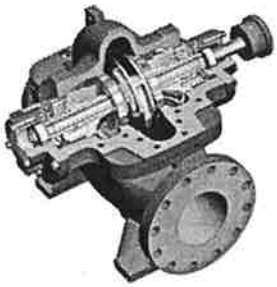
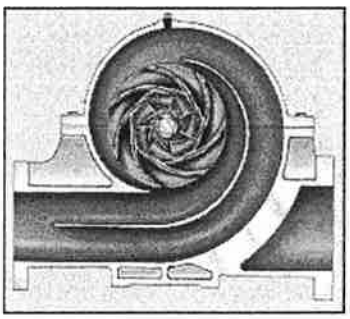


Horizontal coupled motor





Centrifugal pumps – Split casing type

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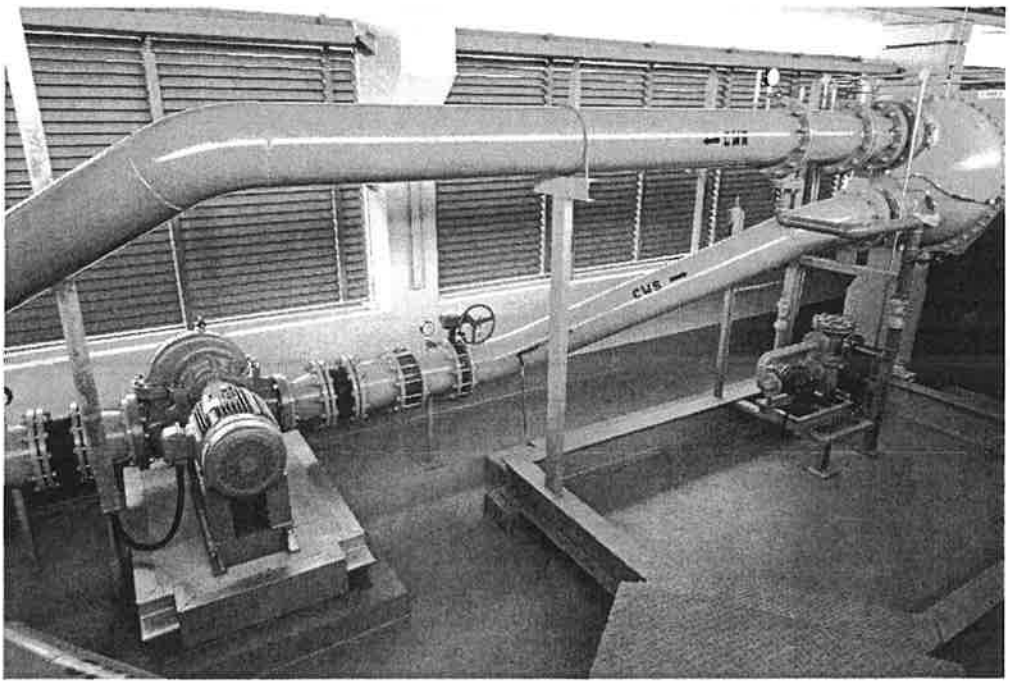

Split casing centrifugal pump

Measurement & Verification Pte Ltd

Performance Range

- Capacity: 8 to 14,000 m^3/h
- Head: 5 to 210m
- Max. efficiency : up to 92%
- Motor power: 0.37 to 3000KW
- Motor speed: 2, 4, 6, 8, 10 pole
- Frequency: 50Hz, 60Hz

Component Material

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Item	Cast Iron S.S. Fitted		Ductile Iron S.S. Fitted		All Stainless Steel		All Bronze	
	Material	Spec.NO.	Material	Spec.NO.	Material	Spec.NO.	Material	Spec.NO.
Casing	Cast Iron	A48	Ductile Iron	A536	St.Steel	AISI-300	Bronze	B145
Case Wear Ring	Cast Iron	A48	Ductile Iron	A536	St.Steel	AISI-300	Bronze	SAE660
Bearing Housing	Cast Iron	A48	Ductile Iron	A536	St.Steel	AISI-300	Bronze	B145
Casing Bolts	Steel	Grade 8 Com.std.	Steel	Grade 8 Com.std.	St.Steel	AISI-300	St.Steel	AISI-300
Brg.Housing "O"Ring	Synthetic	Buna-N	Synthetic	Buna-N	Synthetic	Buna-N	Synthetic	Buna-N
Case Gasket	Vegetable Fiber	HYD-401	Vegetable Fiber	HYD-401	Vegetable Fiber	HYD-401	Vegetable Fiber	HYD-401
Brg.Housing Bolts	Steel	Grade 8 Com.std.	Steel	grade 8 Com.std.	Steel	grade 8 Com.std.	St.Steel	AISI-300
Bearing Cap	Cast Iron	A48	Cast Iron	A48	Cast Iron	A48	Cast Iron	A48
Bearing Cap Bolts	Steel	Grade 8 Com.std.	Steel	grade 8 Com.std.	Steel	grade 8 Com.std.	St.Steel	AISI-300
Bearing Cap Gasket	Vegetable Fiber	HYD-401	Vegetable Fiber	HYD-401	Vegetable Fiber	HYD-401	Vegetable Fiber	HYD-401
Impeller	St.Steel	AISI-300	St.Steel	AISI-300	St.Steel	AISI-300	Bronze	B145
Impeller Wear Rings	Bronze	SAE660	Bronze	SAE660	St.Steel	AISI-300	Bronze	SAE660
Impeller Key	Steel	C1018	Steel	C1018	St.Steel	AISI-300	St.Steel	AISI-300
Shaft	St.Steel	AISI-420	St.Steel	AISI-420	St.Steel	AISI-300	St.Steel	AISI-300
Shaft Sleeve	St.Steel	AISI-300	St.Steel	AISI-300	St.Steel	AISI-300	Bronze	SAE660
Packing Gland	St.Steel	AISI-300	St.Steel	AISI-300	St.Steel	AISI-300	Bronze	B145
Gland Studs	St.Steel	AISI-300	St.Steel	AISI-300	St.Steel	AISI-300	St.Steel	AISI-300
Recirculation Lines	TEFE Tubing	Com.std.	TEFE Tubing	Com.std.	TEFE Tubing	Com.std.	TEFE Tubing	Com.std.
Recirculation Fittings	Brass NPT	Com.std.	Brass NPT	Com.std.	St.Steel	AISI-300	Brass NPT	Com.std.
Lantern(Optional)	St.Steel	AISI-300	St.Steel	AISI-300	St.Steel	AISI-300	St.Steel	AISI-300
Mechanical Seal	Seal face:Silicon carbide Vs Carbon, Metal:316SS, O-ring:Viton							

Inline Centrifugal Pump

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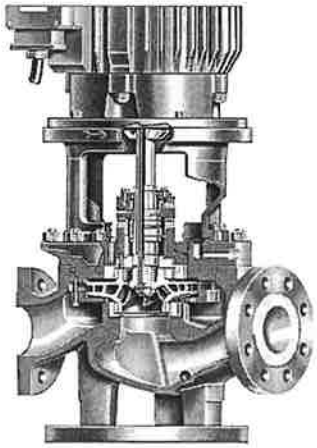

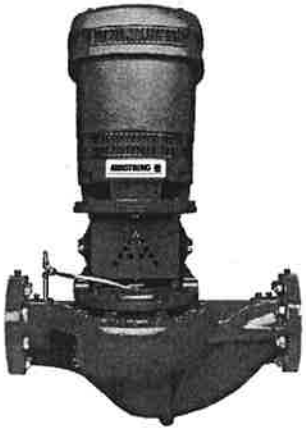

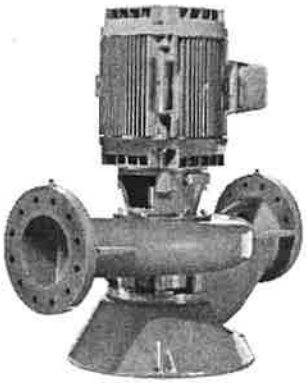

Inline centrifugal Pump

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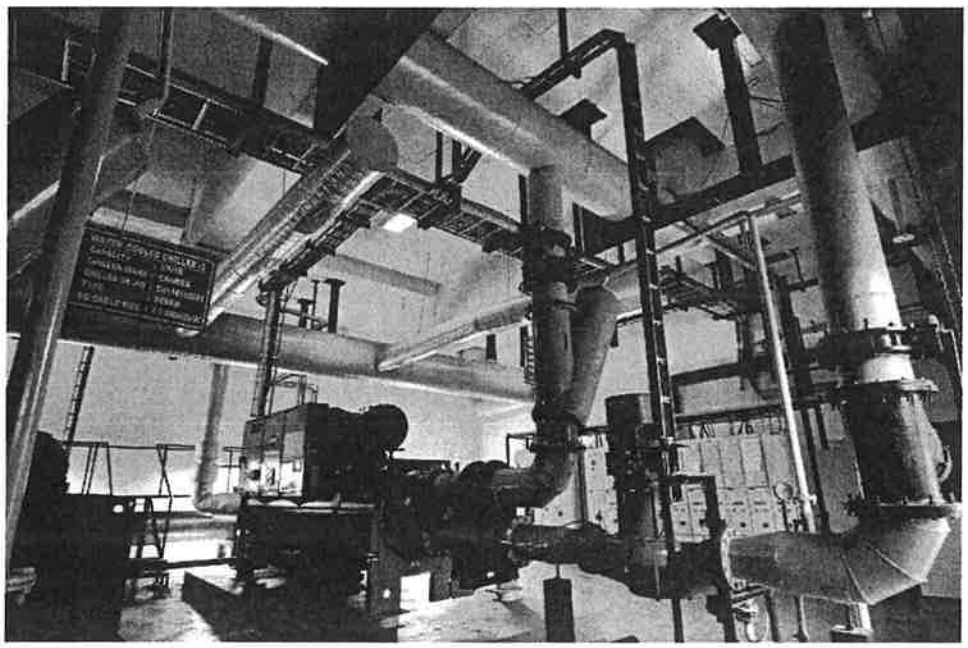

Inline centrifugal pump

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Performance Range

- Capacity: 8 to 6500 m^3/h
- Head: 5 to 150m
- Max. efficiency : 90%
- Motor power: 0.75 kW to 900 kW
- Motor speed: 2, 4, 6, 8, 10 pole
- Frequency: 50Hz, 60Hz

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PUMP CURVE, OPERATING POINT & SYSTEM CURVE

Pump Curve

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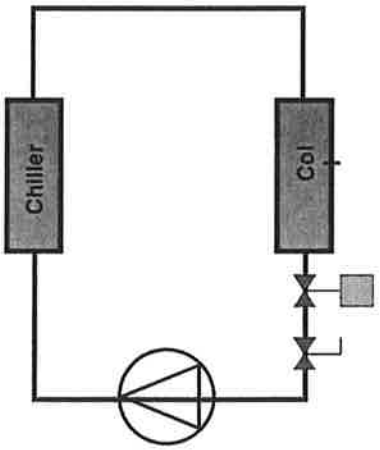
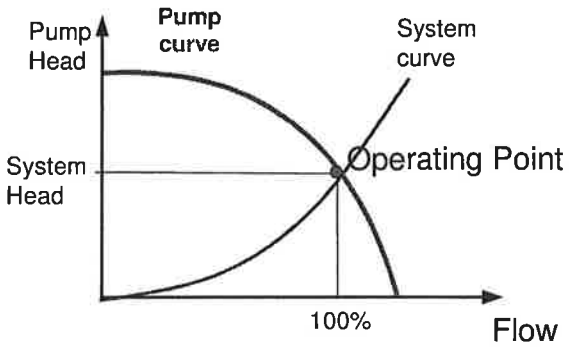
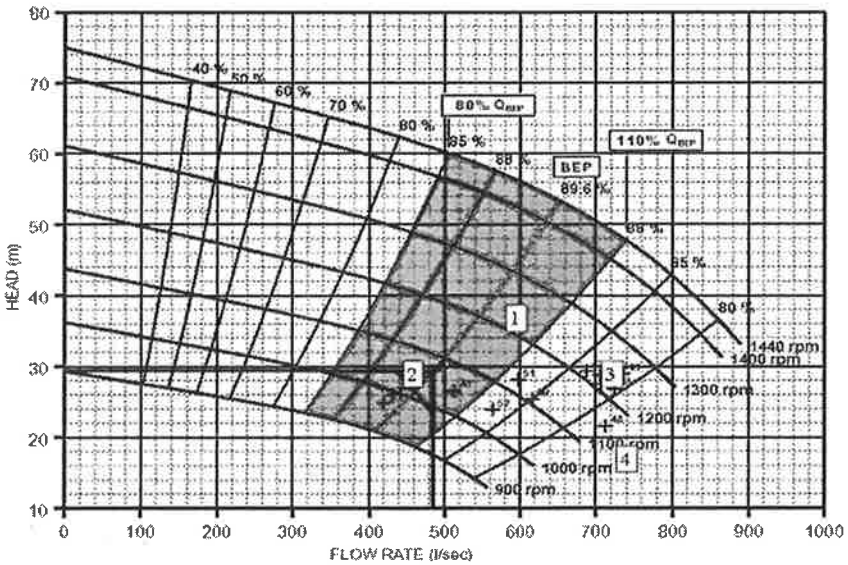




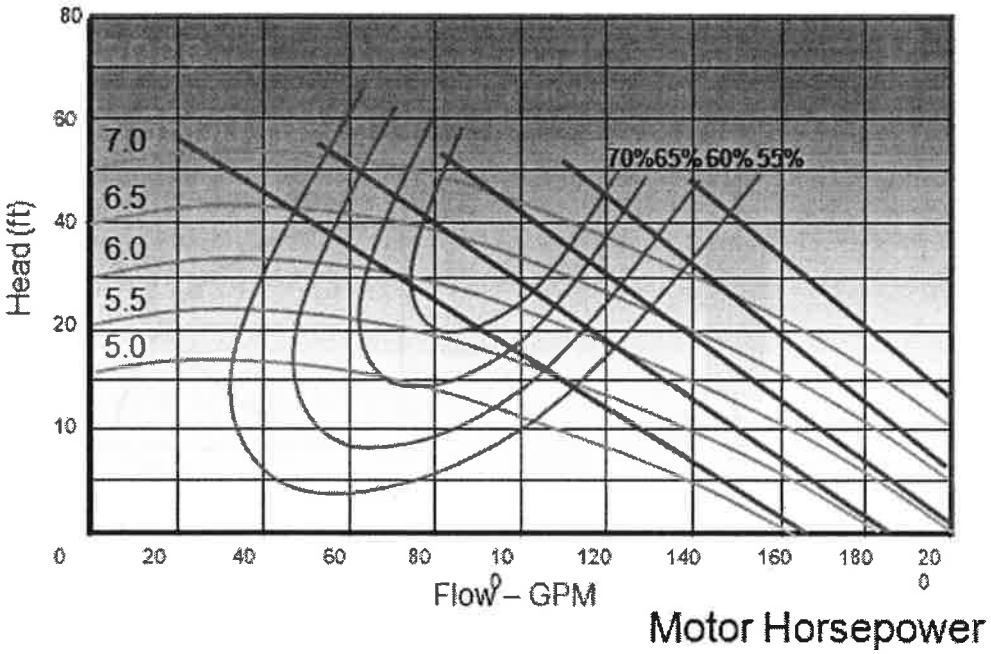
System Curves for a Pump Type

Measurement & Verification Pte Ltd



1 Pump Best Efficiency Point (BEP)
2 Ideal pump duty
3 Pump operating to the right of BEP
4 Pump cavitating (very far right of BEP)

System Curves for a Pump Type



EFFICIENCY

Motor Efficiency

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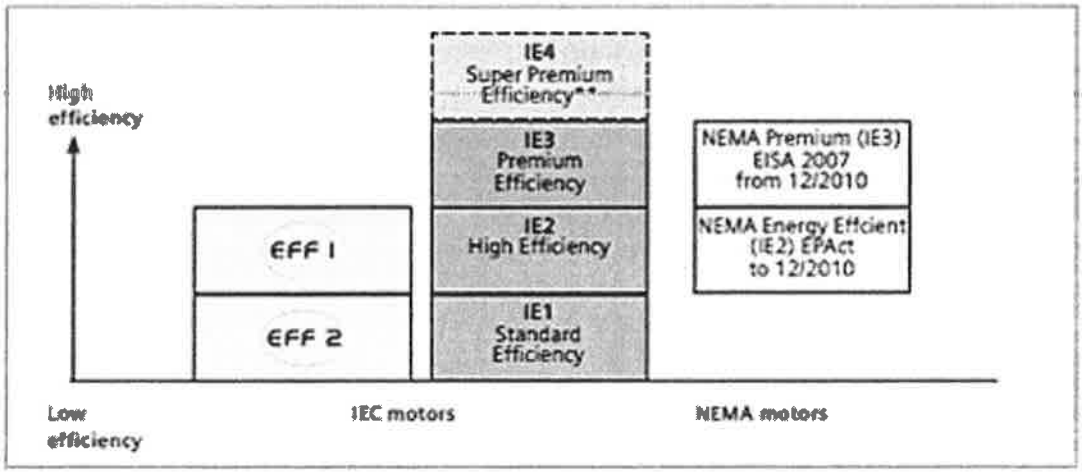

Motor Efficiency

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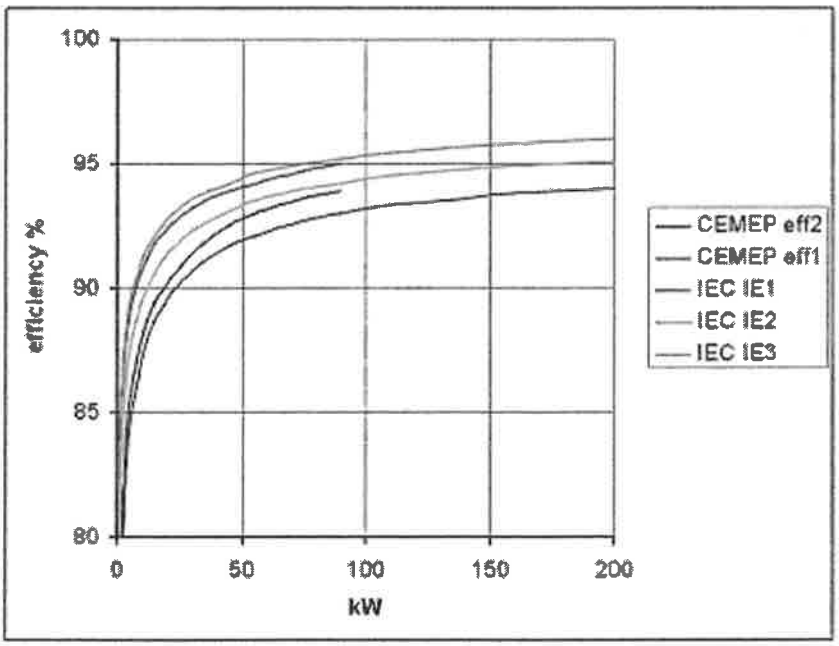

Motor efficiency

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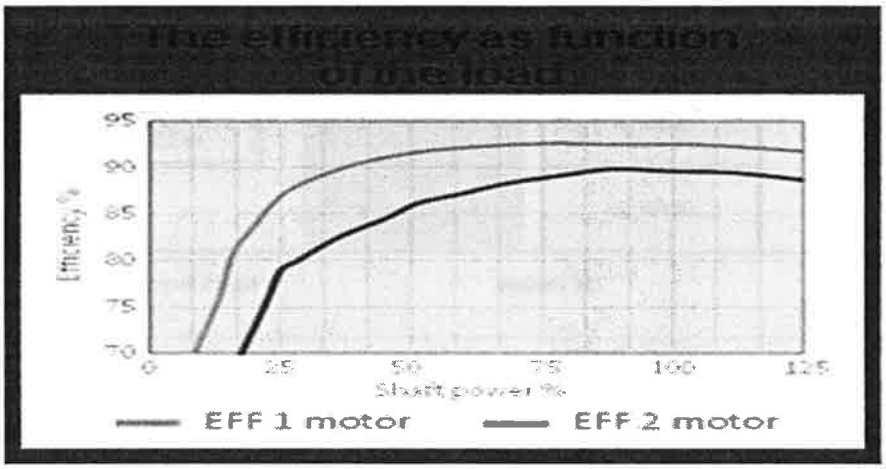
- The motor efficiency is dependent on the type of motor (EFF1 or EFF2) and the load of the motor (shaft power). A low load (below 25%) means a significant drop in efficiency.



Drive Efficiency

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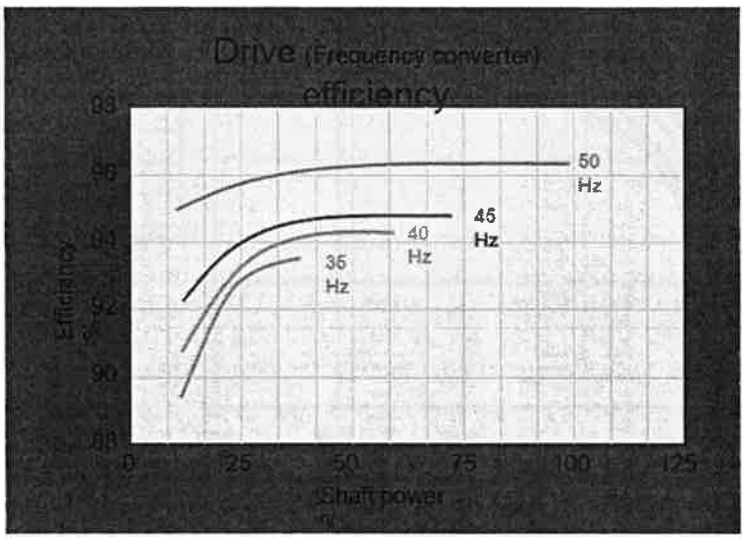
- The drive efficiency is dependent on the load of motor and the frequency output, but the efficiency down to 30% load is more or less constant.



Pump power

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$$\text{Pump power} = \frac{\text{Flow rate x Head}}{1000 \times \eta\text{pump} \times \eta\text{motor}}$$

Where:
Pump power: KW
Flow rate: L/s
Head: kpa
ηpump: Pump efficiency (%)
ηmotor: Motor efficiency (%)

Energy Efficiency Opportunities

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Efficiency Measures	Range of Savings (% of system energy)
Reduce Overall System Requirements	5-20%
Match Pump Size to Load	10-30%
Reduce or Control Pump Speed	5-50%
Component Purchase	1-3%
Operation & Maintenance	1-5%

Source: US Industrial Motor Systems Market Opportunities Assessment US Dept. of Energy, 1998

Pump vs. System Standards

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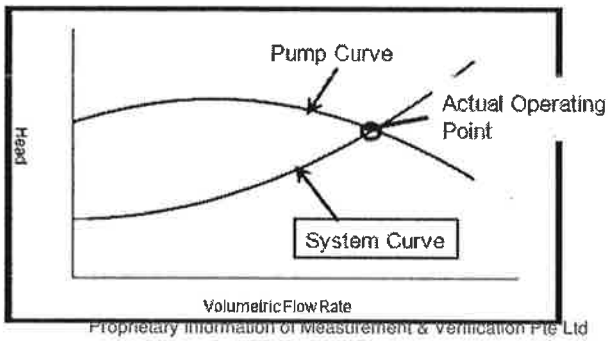
Pump Standards

- Pump design is subject to numerous standards
  - HI, API, ANSI, ISO.
- Pump hydraulic data is subject to numerous standards
  - Developed head, Efficiency, NPSHr

System Standards

- With few exceptions, there are no standards to guide system design
- Engineering contractors and owner/operators are allowed to choose (or ignore) how to calculate system hydraulics
  - Specified pump operating point not subject to standards



Proprietary Information of Measurement & Verification Pte Ltd

Tips for Energy Saving

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- Improve piping configuration
  - Use larger pipe sizes
  - Eliminate unnecessary turns, valves, accessories
  - Optimize pump inlet and outlet piping
- Minimize throttling
- Assess pumping system suitability for current application. Many installed systems are oversized, providing an opportunity to:
  - Install a smaller impeller or trim the existing one
  - Remove stages
  - Downsize pump
  - Install a smaller and/or a more efficient pump motor
  - Replace worn impellers

Tips for Energy Saving

Measurement & Verification Pte Ltd

- Reduce pump speed or install appropriate speed control devices
  - Install a slower speed motor
  - Change sheave diameters
  - Consider variable frequency drives
- Consider alternative pump configurations
  - Pony pumps
  - Multiple pumps in parallel
- Eliminate unnecessary uses
  - Schedule pumps to turn off whenever possible
  - Avoid recirculation through bypass lines
- Improve O&M practices
  - Establish basic system maintenance checklists and schedules
  - Establish a predictive maintenance program

THANK YOU

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COOLING TOWER

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Agenda

1. Types Of Cooling Tower
2. Cooling Tower Performance
3. Layout Considerations

TYPES OF COOLING TOWER

Definition of a Cooling Tower

"Heat transfer device, often tower-like, in which atmospheric air cools warm water, generally by direct contact (evaporation)."

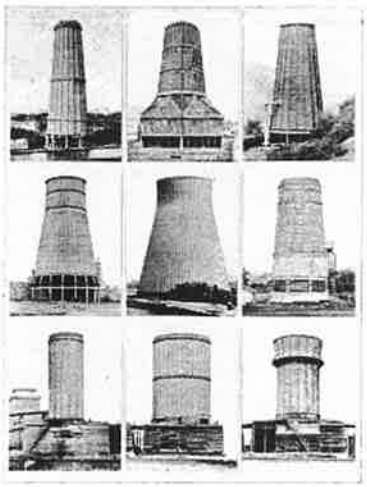
ASHRAE Terminology of HVAC&R

Cooling Tower Types

Natural Draught Tower



Old Cooling Towers



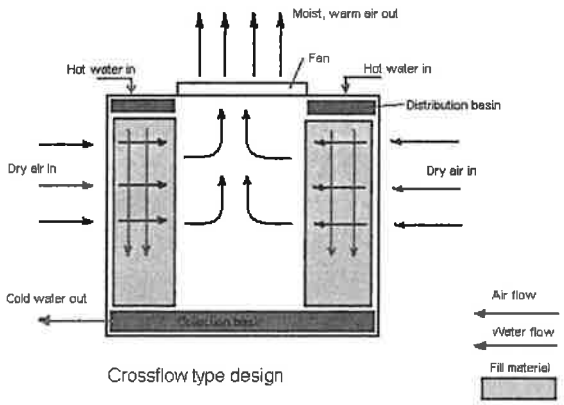
Induced Draft Cross-flow Tower

Air Movement

- Air is induced (drawn through) through the cooling tower infills
- Horizontally through the fill, across the downward fall of the water.

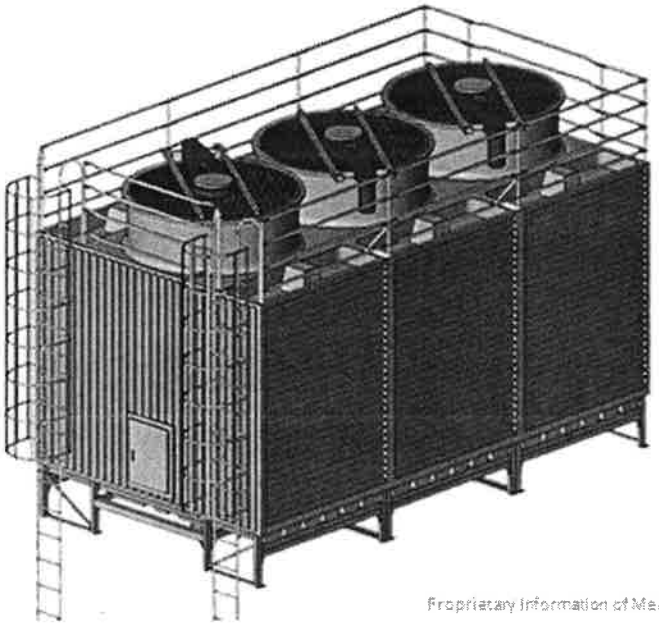
Water Distribution System

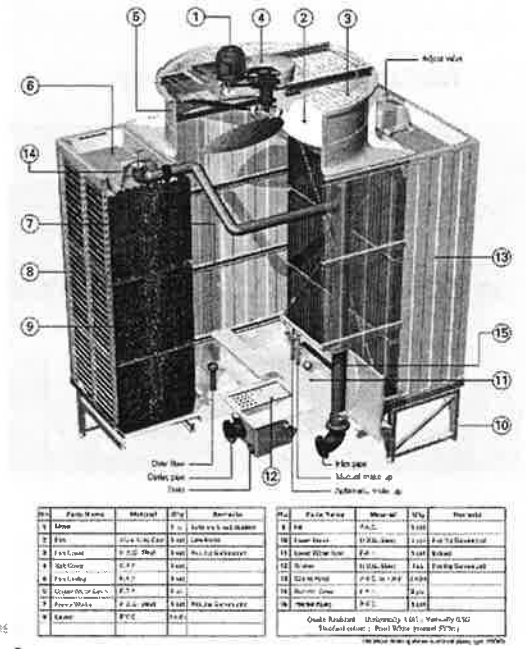
- Hot water is distributed to the fill by gravity through metering orifices in the floor of the inlet basin. There is no pressure spray distribution system.
- The internal pressure component of pump head is insignificant because maximum flow is achieved by gravity.



Crossflow type design

Induced Draft Cross-flow Tower





Proprietary Information of Measurem

Induced Draft Counter-flow Tower

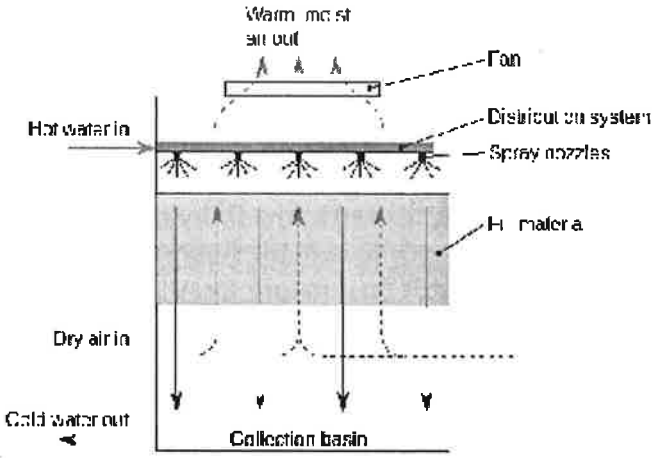
Air Movement

- Air is induced (drawn through) through the cooling tower infills.
- The air movement is vertically upward through the fill, counter to the downward fall of the water.

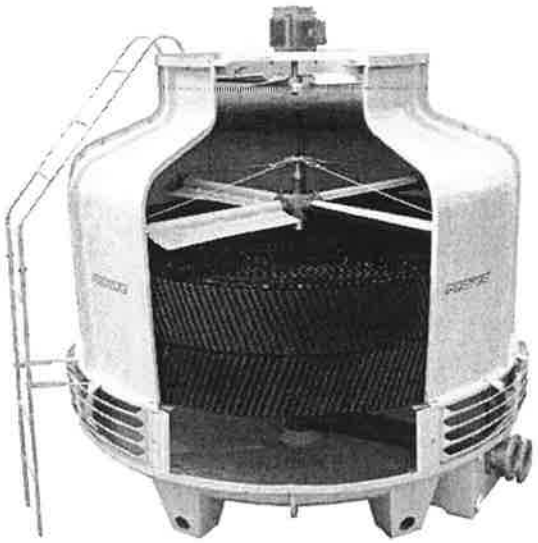
Water Distribution System

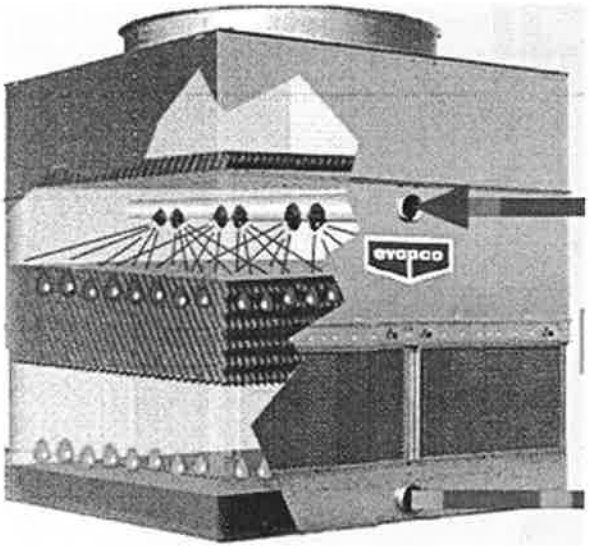
- Counterflow towers use a high pressure spray nozzle hot water distribution system to achieve water coverage of the fill.
- The nozzle spray pattern is sensitive to changes in water flow, and consequent change in nozzle pressure.

Counterflow towers typically have a smaller footprint than crossflow owers, but require additional height, static lift, and dynamic head to achieve the same cooling effect.



Induced Draft Counter-flow Tower





Forced Draft Counter-flow Tower

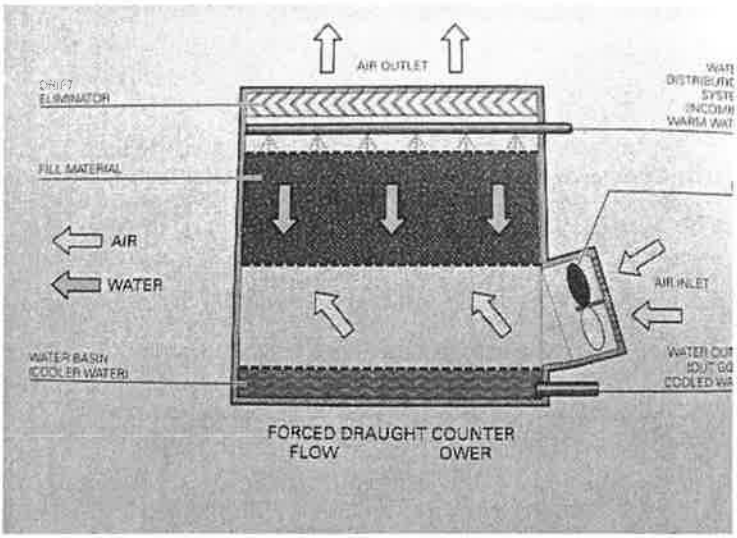
Air Movement

- Air is forced (pusedh through) through the cooling tower infills
- The air movement is vertically upward through the fill, counter to the downward fall of the water.

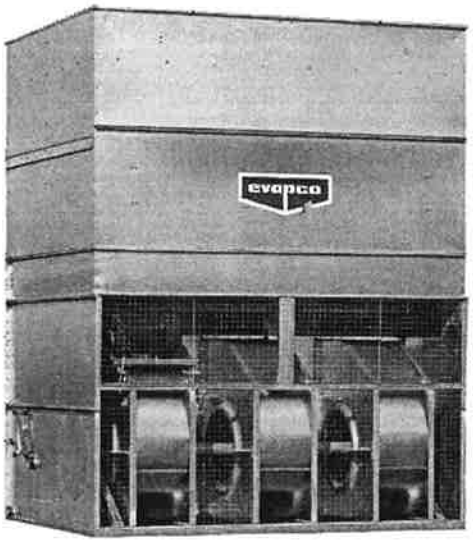
Water Distribution System

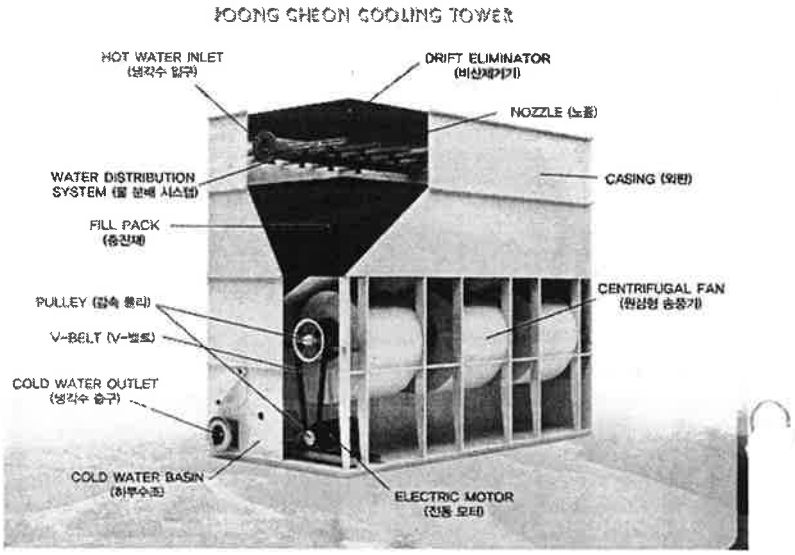
- Counterflow towers use a high pressure spray nozzle hot water distribution system to achieve water coverage of the fill.
- The nozzle spray pattern is sensitive to changes in water flow, and consequent change in nozzle pressure.

Normally used when inlet air intake is restricted and needs high static fans



Forced Draft Counter-flow Tower





Induced Draft Cross-flow Tower

Advantages	Disadvantages
Gravity flow hot water distribution	
• Low pumping head. • Lower first cost pumping systems. • Lower annual energy consumption and operating costs. • Accepts larger variation in water flow without adverse effect on the water distribution pattern (flat plate heat exchanger operation in winter). • Easy maintenance access to distribution nozzles.	• Low pressure head on the distribution pan may encourage orifice clogging and less water breakup at spray nozzle. • Exposure to air in the hot water basin may accelerate algae growth. • Larger footprint.
Horizontal air distribution system	
• The vertical air movement across the fill allows the coldest water to be in contact with the driest air maximizing tower performance.	• The resistance to upward air travel against the falling water results in higher static pressure loss and a greater fan horsepower than in crossflow towers. • The restricted louver area at the base with high velocity of inlet air increases the fan horsepower. • Tendency for uneven distribution of air through the fill with very little movement near the walls and center of the tower. • High inlet velocities are liable to suck airborne trash and dirt into the tower.

Counter - flow

Induced Draft Counter-flow Tower

Advantages	Disadvantages
Pressurized spray water distribution system:	
• Increased tower height accommodates longer ranges and closer approaches. • More efficient use of air due to finer droplet size from pressure sprays.	• Increased system pumping head requirements. • Increased energy consumption and operating costs. • Distribution nozzles difficult to inspect and clean. • Requires individual risers for each cell, increasing external piping costs.
Vertical air distribution system:	
• Low static pressure drop. • Reduced drift. • Reduced recirculation. • More air per fan horsepower. • Larger diameter fans can be used so that fewer cells are required for a given capacity. • Lower energy and operating costs.	• Larger louver surface area makes icing more difficult to control.

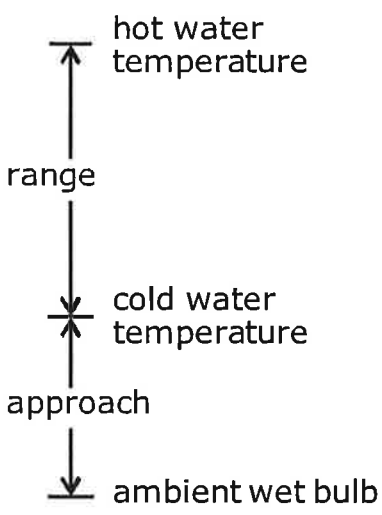
Cross-Flow

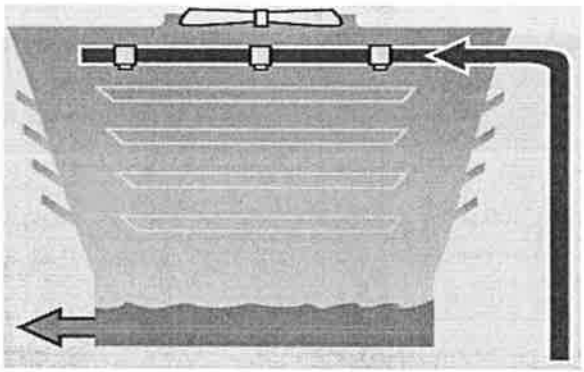
Conclusions and Recommendations

Crossflow cooling towers should be specified when the following criteria and limitations are important:	Counterflow cooling towers should be specified when the following criteria and limitations are important:
• To minimize pump head. • To minimize pumping and piping first costs. • To minimize operating costs. • When condenser water flow variance is expected. • When ease of maintenance is a concern.	• When space (footprint) is restricted. • When pumping is designed for additional pressure drop. • When icing is of extreme concern.

Cooling Tower Performance

Performance Factors





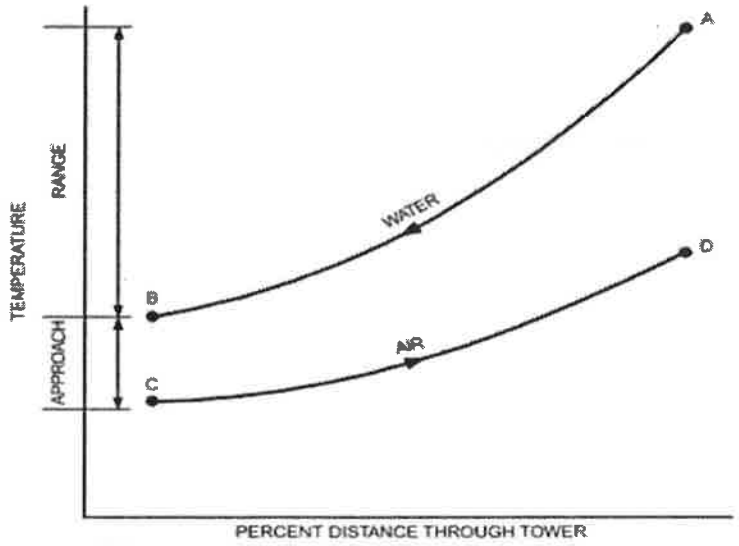
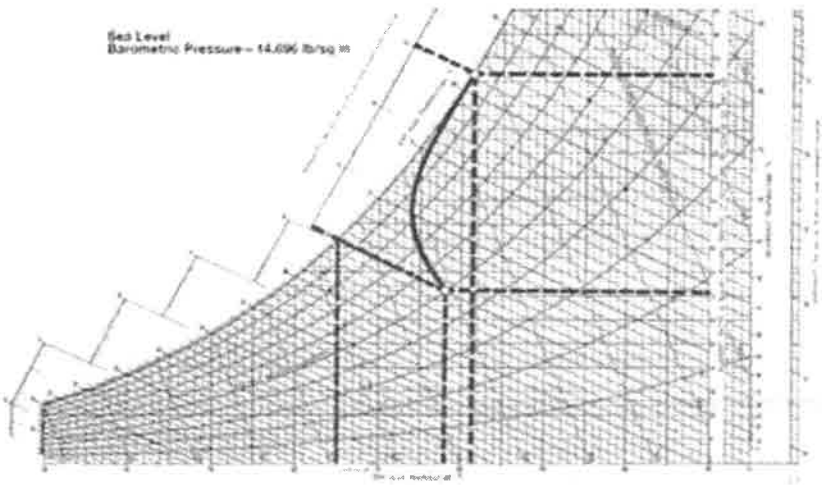


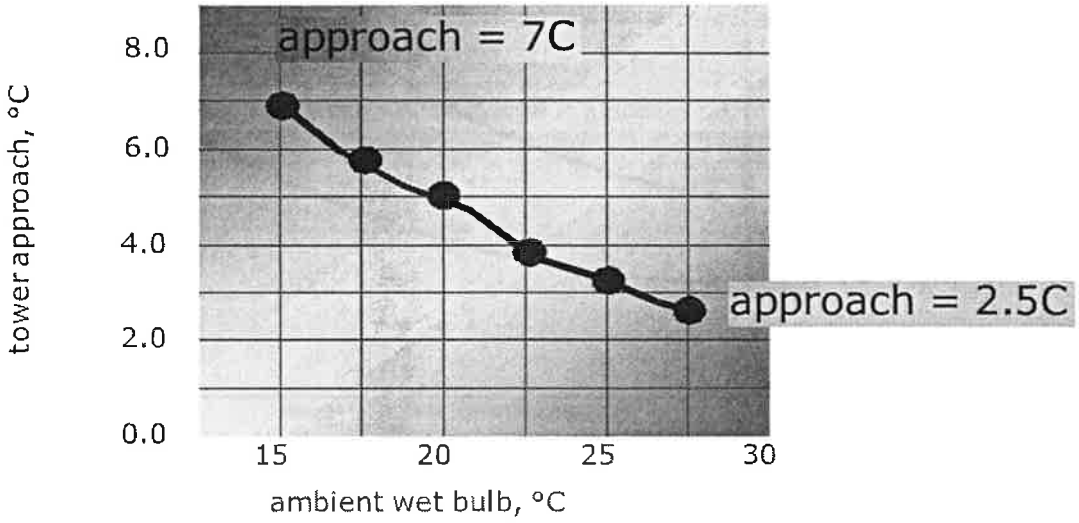
Figure 1. Temperature Relationship Between Water and Air in Counterflow Cooling Tower

7 L/RT water losses.



The CT air discharge air is typically saturated and at a higher WB and DB compared to the inlet air.
The increase in air enthalpy (approximated by WB) is equal to the heat absorbed from the CW.

Approach and Wet Bulb



Cooling Tower Rating

- CTI – 201: Rating Standard for Open Loop Propeller or Axial Fan Cooling

<u>Typical cooling tower rating condition for Singapore</u>

- Entering water temperature: 35.0C
- Leaving water temperature: 29.5C
- Ambient wet bulb temperature: 27.0C

LAYOUT CONSIDERATIONS

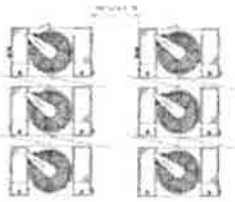


Figure 1

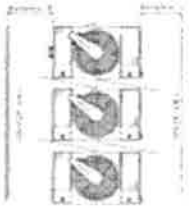


Figure 2

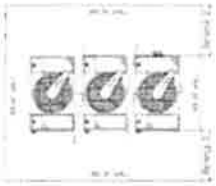


Figure 3

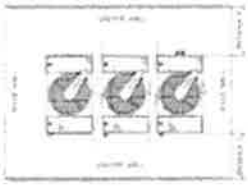


Figure 4

Cell	Figure 1	Figure 2		Figure 3	Figure 4	Figure 5
	Min. Distance Louver To Louver	Min. Distance Air Inlet To Louver Wall	Min. Distance Air Inlet To Solid Wall	Min. Distance For Cooling Tower In Enclosure (Solid Wall)	Min. Distance For Cooling Tower In Enclosure (Louver To Louver Wall)	Min. Distance For Cooling Tower In Enclosure (Louver Wall)
	Distance A	Distance B	Distance C	Distance D	Distance E	Distance F
1 Cell	1.5 Metre	1.5 Metre	1.5 Metre	1.5 Metre	1.5 Metre	1.5 Metre
2 Cells	2.5 Metre	1.5 Metre	2.5 Metre	2.5 Metre	1.5 Metre	1.5 Metre
3 Cells	3.5 Metre	1.5 Metre	3.5 Metre	3.5 Metre	1.5 Metre	1.5 Metre
4 Cells	5.0 Metre	1.5 Metre	5.0 Metre	5.0 Metre	1.5 Metre	1.5 Metre
5 Cells	5.0 Metre	1.5 Metre	5.0 Metre	5.0 Metre	1.5 Metre	1.5 Metre
6 Cells	5.0 Metre	1.5 Metre	5.0 Metre	5.0 Metre	1.5 Metre	1.5 Metre
7 Cells	5.0 Metre	1.5 Metre	5.0 Metre	5.0 Metre	1.5 Metre	1.5 Metre
8 Cells	5.0 Metre	1.5 Metre	5.0 Metre	5.0 Metre	1.5 Metre	1.5 Metre
9 Cells	5.0 Metre	1.5 Metre	5.0 Metre	5.0 Metre	1.5 Metre	1.5 Metre
10 Cells	5.0 Metre	1.5 Metre	5.0 Metre	5.0 Metre	1.5 Metre	1.5 Metre

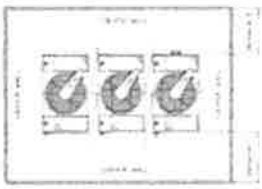


Figure 5

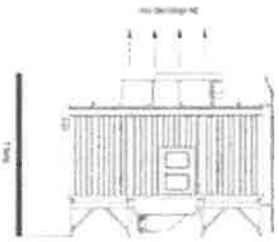


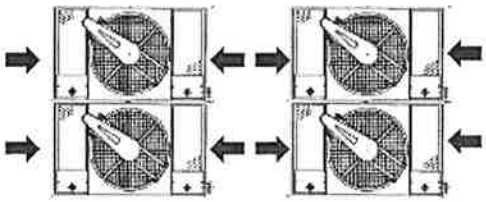
Figure 6

This is only a guide for a particular model of CT.

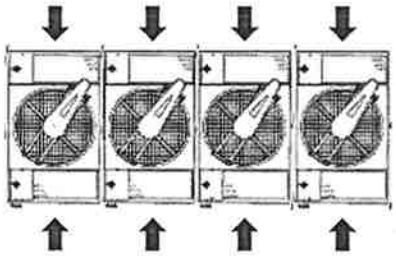
Requirements vary for different brands and models.

Consult OEM for different brands

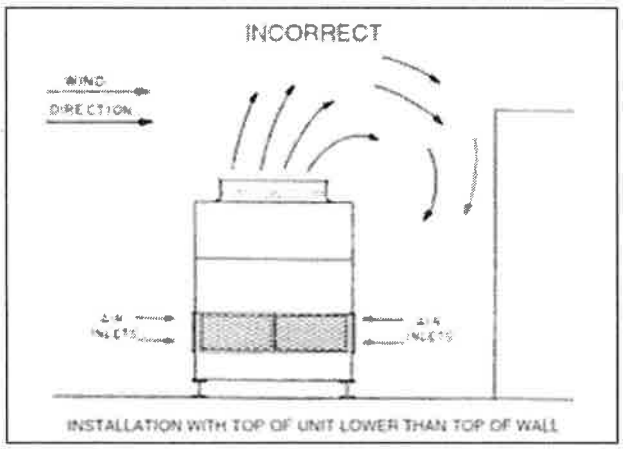
Intake air adjacent to each other

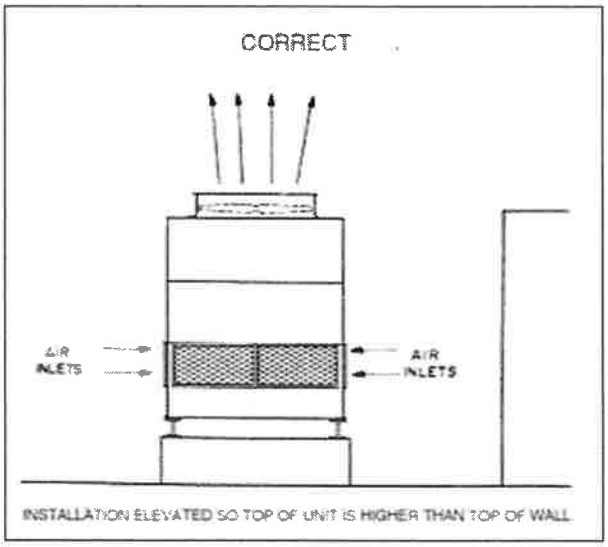


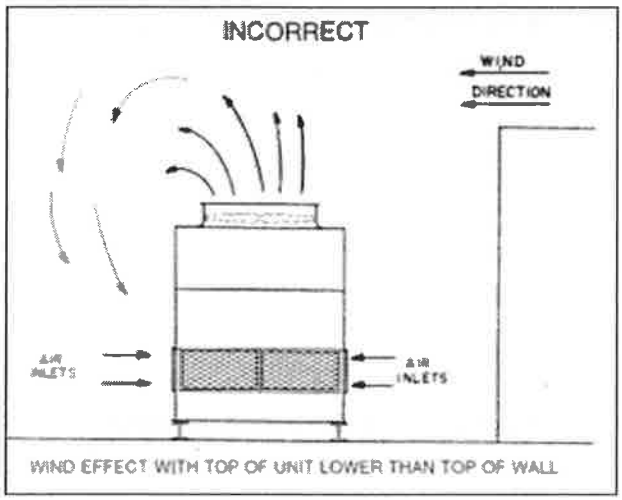
Intake air parallel to each other

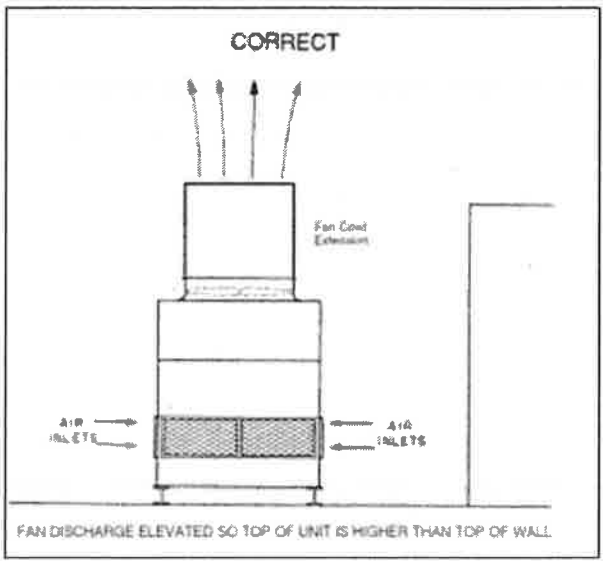


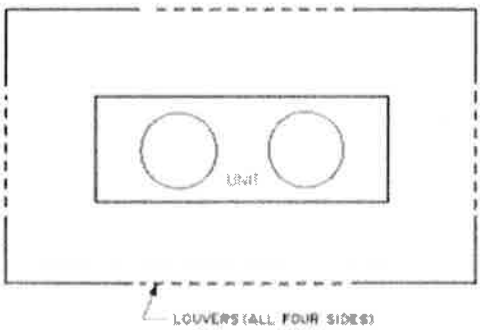
Preferred Layout











The inlet air will be drawn from the top and through the louvers.
To minimize the potential for recirculation, it is better to draw most of the air through the louvers.
Since the air will follow the path of least resistance, the pressure drop through the louvers will determine how much air is drawn from both areas.
Therefore, it is important that the louvers are designed for minimum pressure drop.
To achieve this goal, the velocity through the louvers should be maintained at or below 600 F.P.M., the louvers should have a minimum of 50% net free area, and the air inlets should face the louvers.



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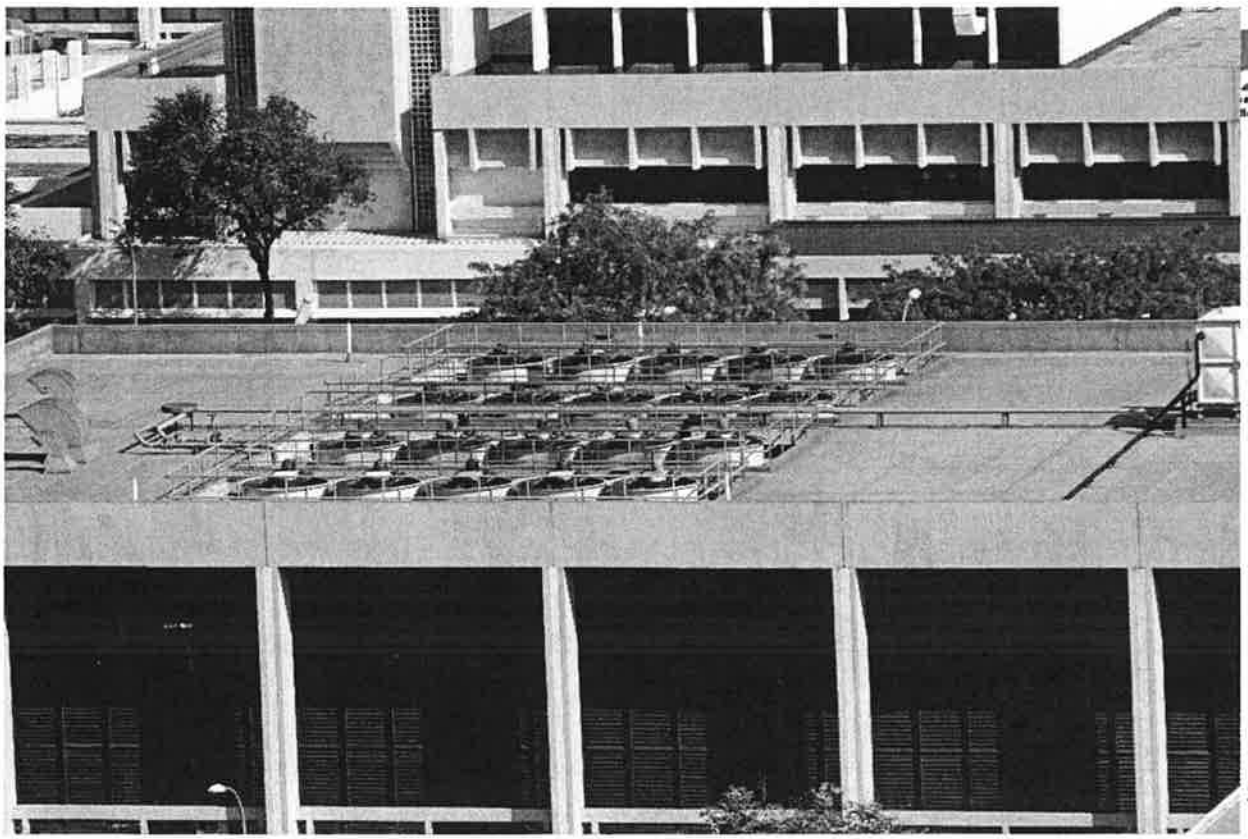
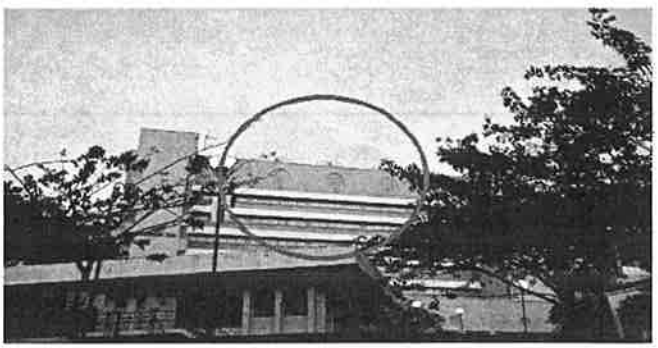
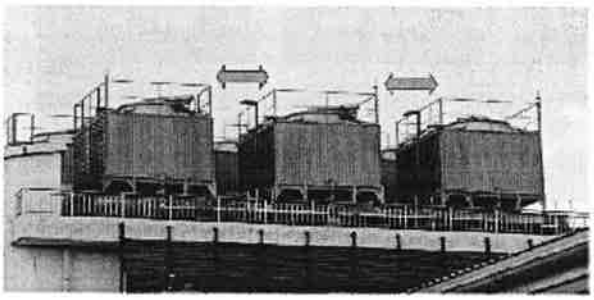


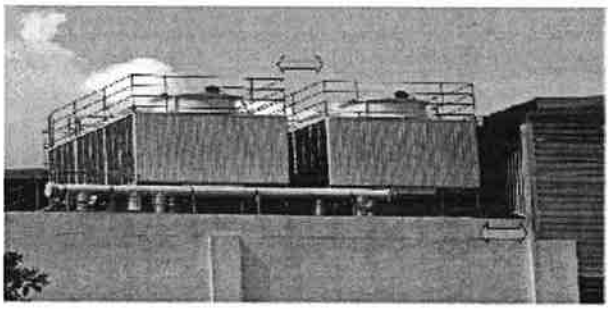
Cooling Tower Layout Considerations





Cooling Tower Layout Considerations









THANK YOU





AHU & FCU TECHNOLOGIES

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Agenda

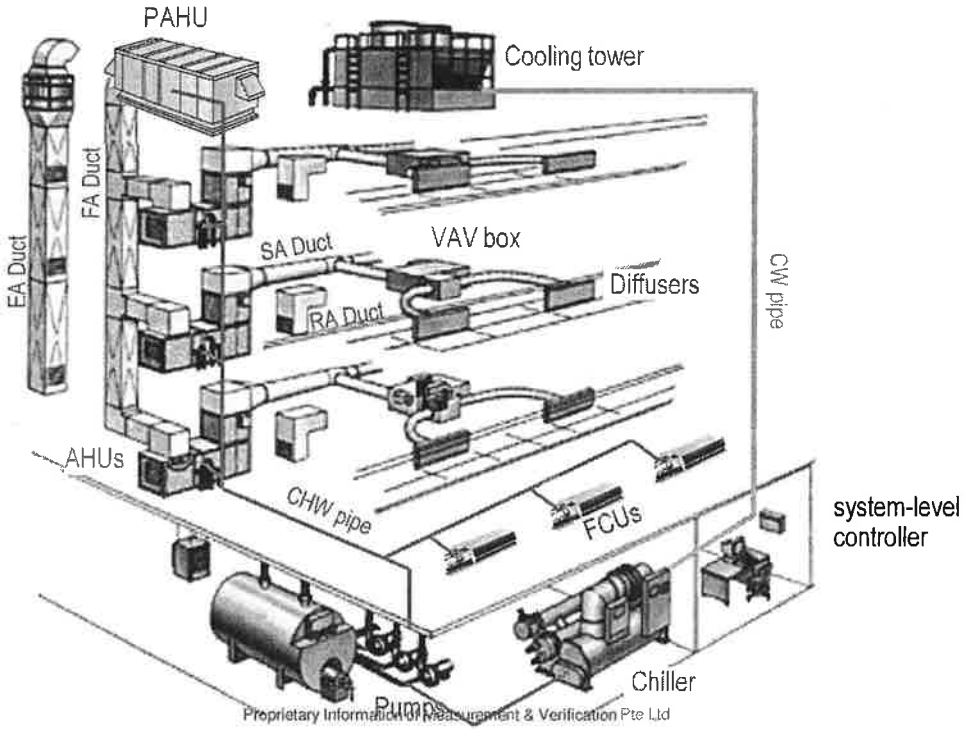
1. Overview of Air Side System
2. Air handling unit
3. Fan coil unit
4. Variable volume box

OVERVIEW OF AIR SIDE SYSTEM

Types of airside equipment

- **Air Handling Unit (AHU)**
  - Cool, dehumidify & filter recirculating air
  - Usually floor mounted in AHU room. However, increasingly are suspended from ceiling slab to save space.
  - Airflow: 2,000 to 90,000 CMH
  - External static pressure: Up to 1000pa
  - Supply air temperature: 12 to 15C
- **Pre-cooled Air Handling Unit (PAHU)**
  - Pre cool, dehumidify and filter outside air.
  - Unusually located at rooftop
- **Fan Coil Unit (FCU)**
  - Cool, dehumidify & filter recirculating air
  - Many types: Ducted, ceiling cassette, suspended, wall mounted
  - Airflow: 400 to 10,000 CMH
- **Variable Air Volume Box (VAV)**
  - Modulates the amount of cool air into the space to maintain comfort
- **Diffuser**
  - Distribute cool air evenly into space

Chilled water system components



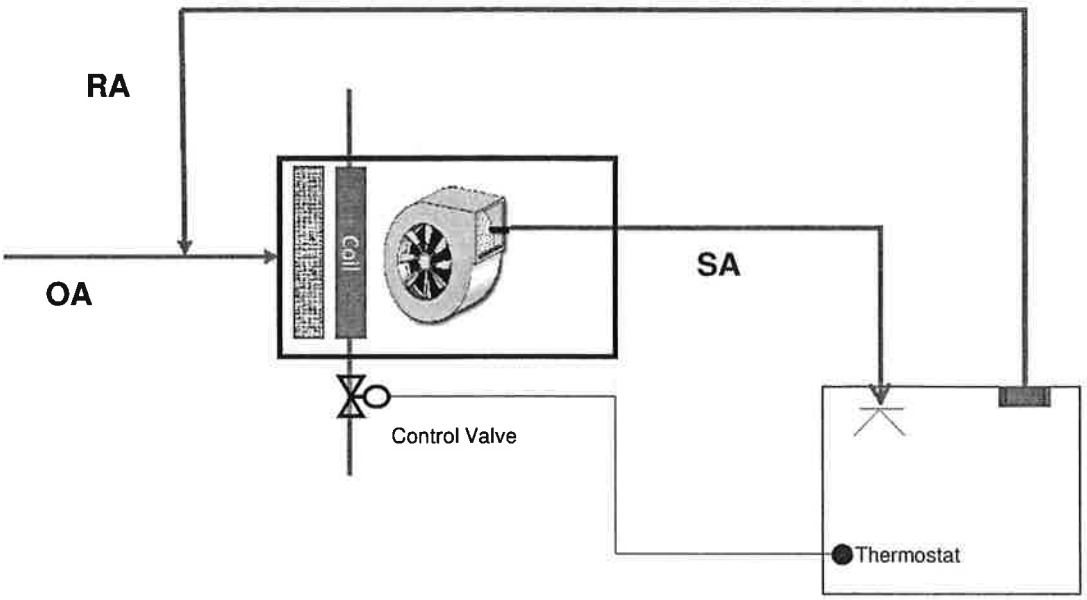
Proprietary Information of Measurement & Verification Pte Ltd

Types of airside systems

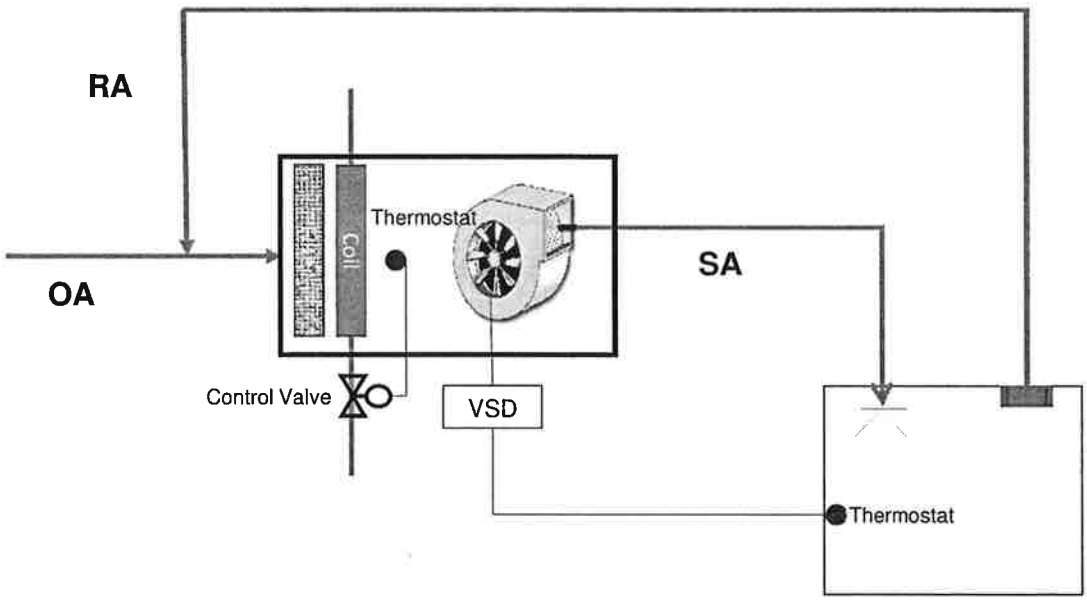
zones served by supply fan \ air volume delivered by supply fan		
	single zone constant volume	single zone variable volume
	multiple zone constant volume	multiple zone variable volume

Proprietary Information of Measurement & Verification Pte Ltd

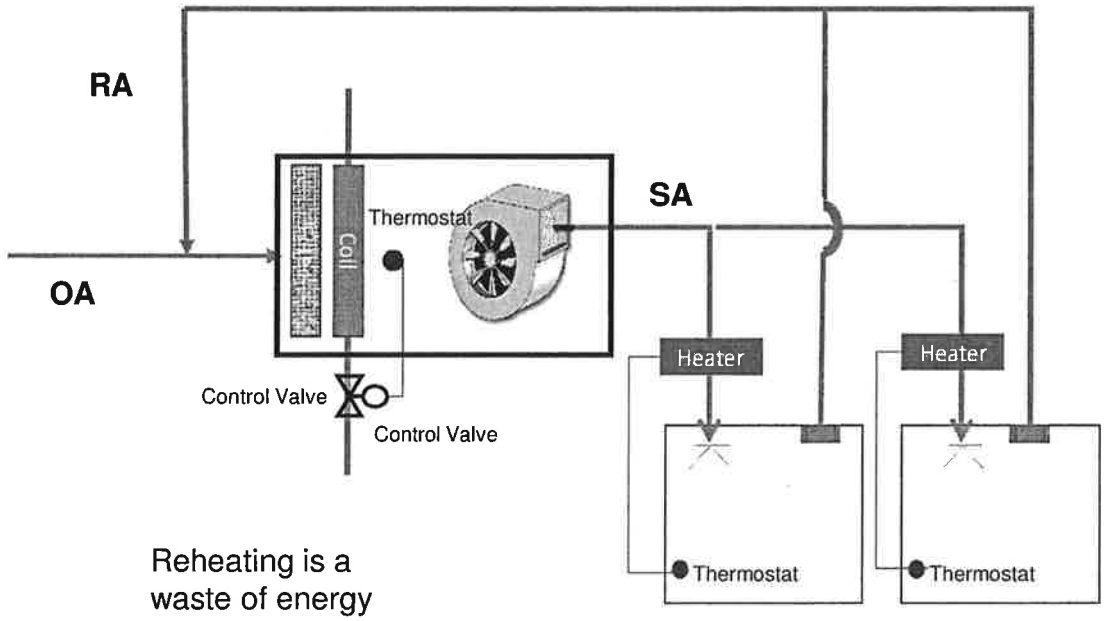
Single Zone, Constant Volume



Single Zone, Variable Volume

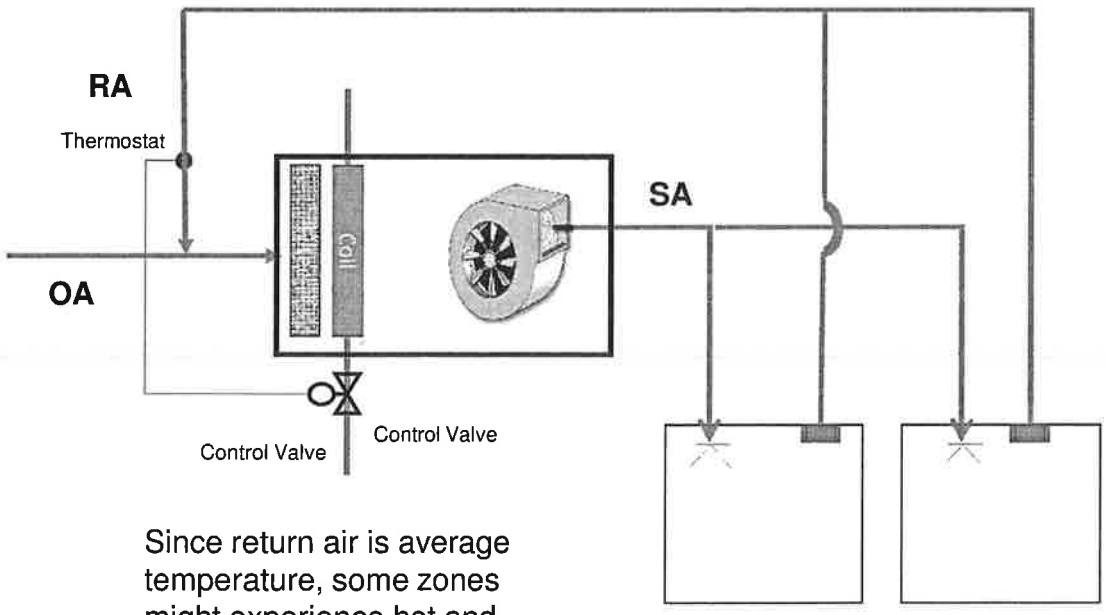


Multiple Zone, Constant Volume (With reheat)



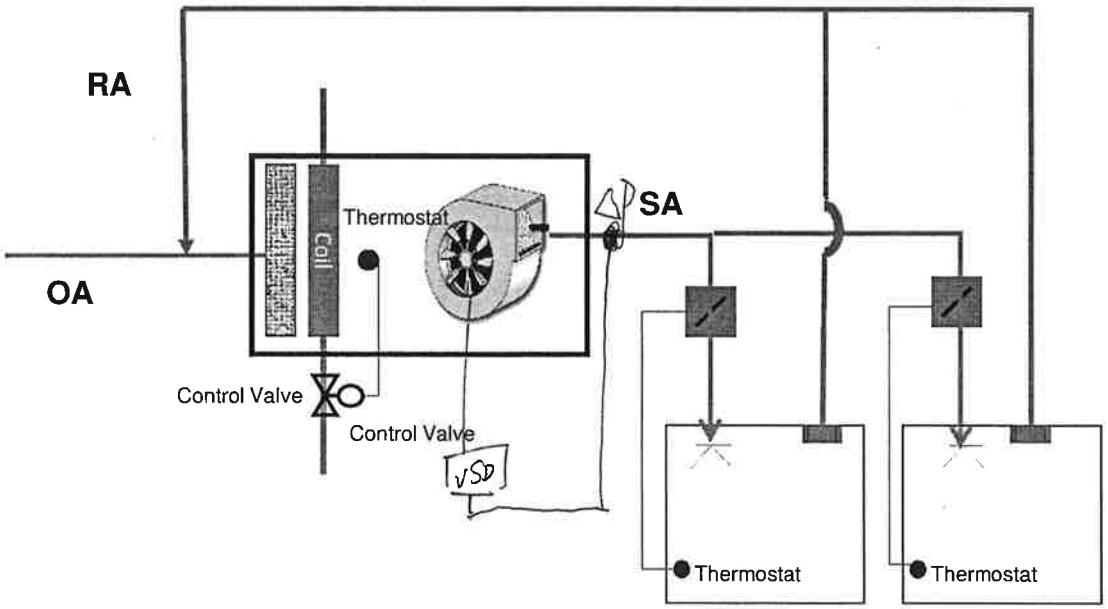
Reheating is a waste of energy

Multiple Zone, Constant Volume (Without reheat)



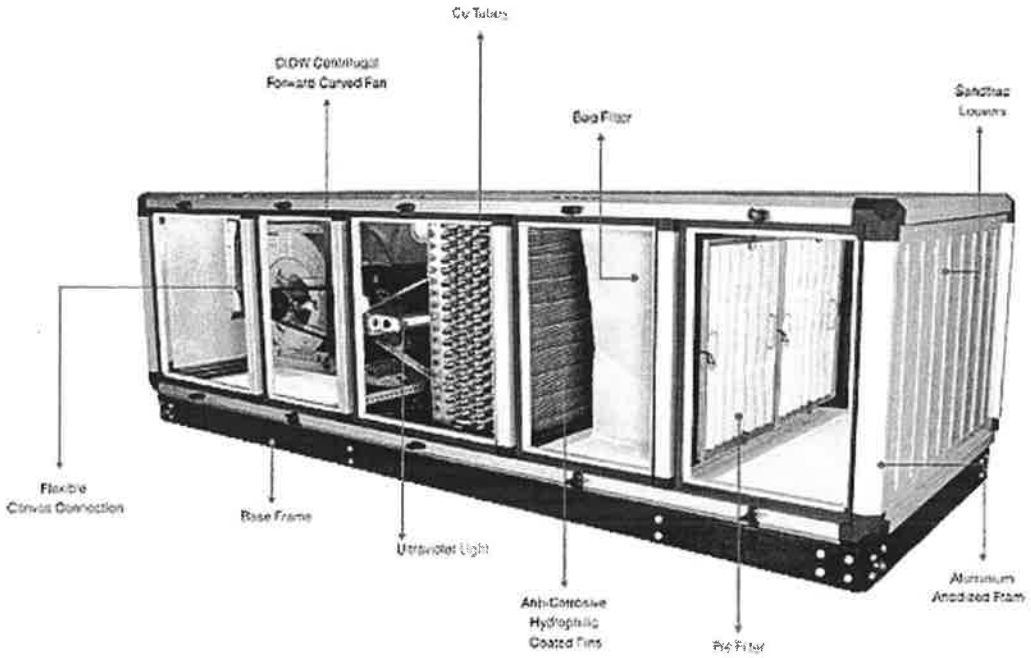
Since return air is average temperature, some zones might experience hot and cold spots.

Multiple Zone, Variable Volume

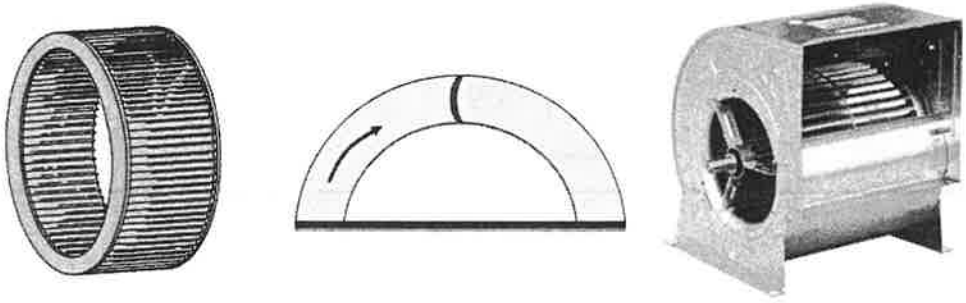


Air Handling Unit (AHU)

AHU

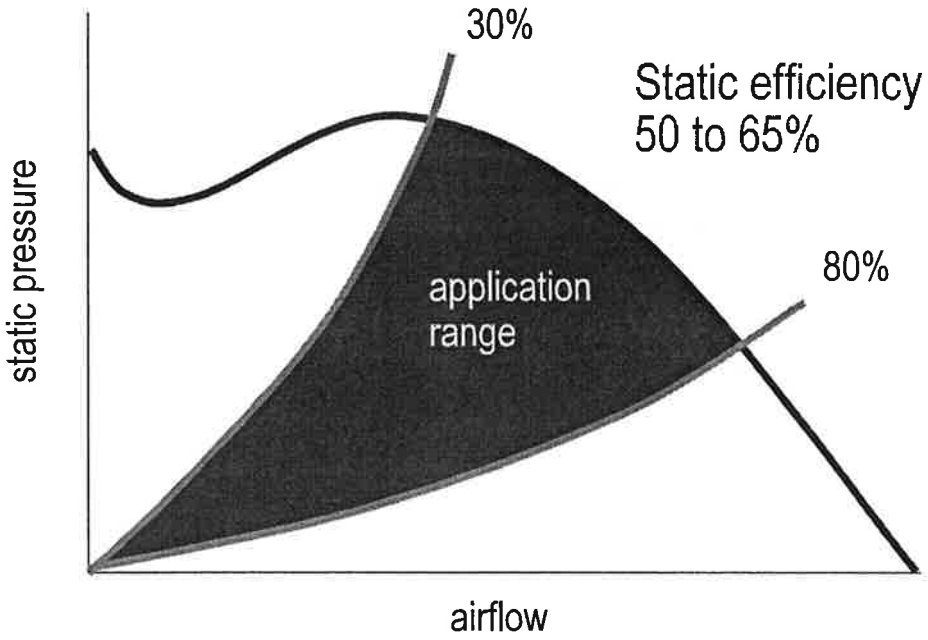


AHU centrifugal fan (FC fan)



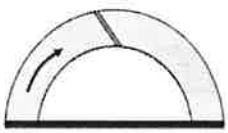
- Forward curved (FC) centrifugal fan has blades that are curved in the direction of wheel rotation.
- FC fans are operated at relatively low speeds and are used to deliver large volumes of air against relatively low static pressures.
- The inherently light construction of the forward curved fan wheel does not permit this wheel to be operated at the speeds needed to generate high static pressures.

AHU centrifugal fan (FC fan)



AHU centrifugal fan (BC fan)

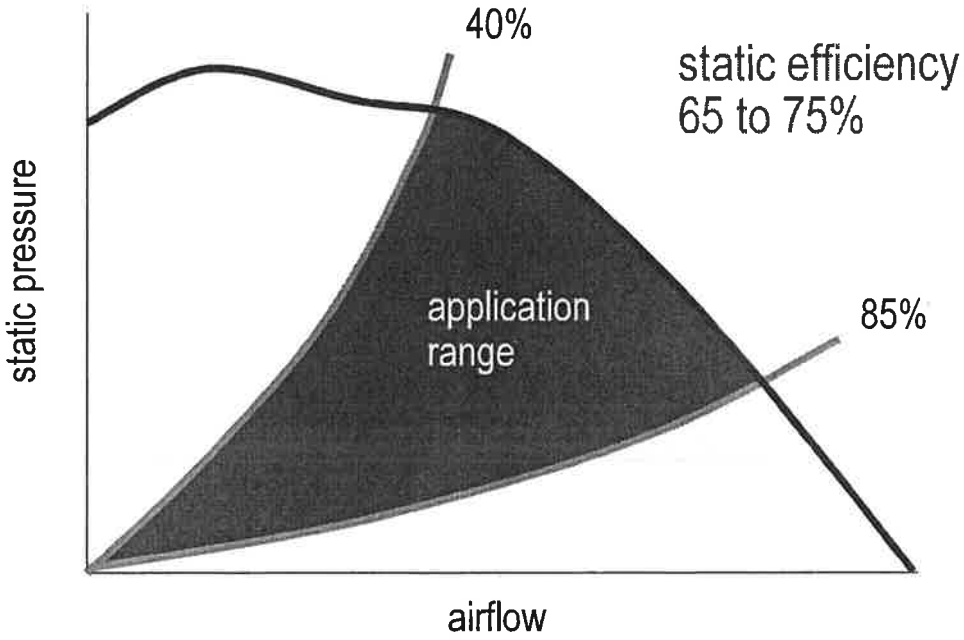




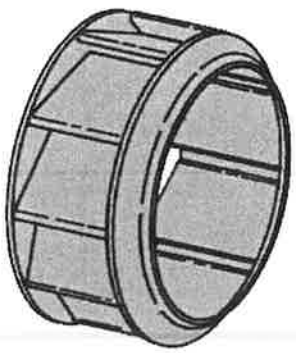


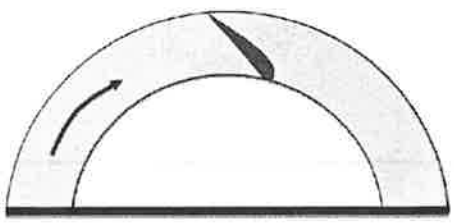
- Backward curved (BC) centrifugal fan design has blades that are slanted away from the direction of wheel rotation.
- BC fan operate at higher speeds than FC fans.
- Their rugged construction makes them suitable for moving large volumes of air in higher static-pressure applications.

AHU centrifugal fan (BC fan)



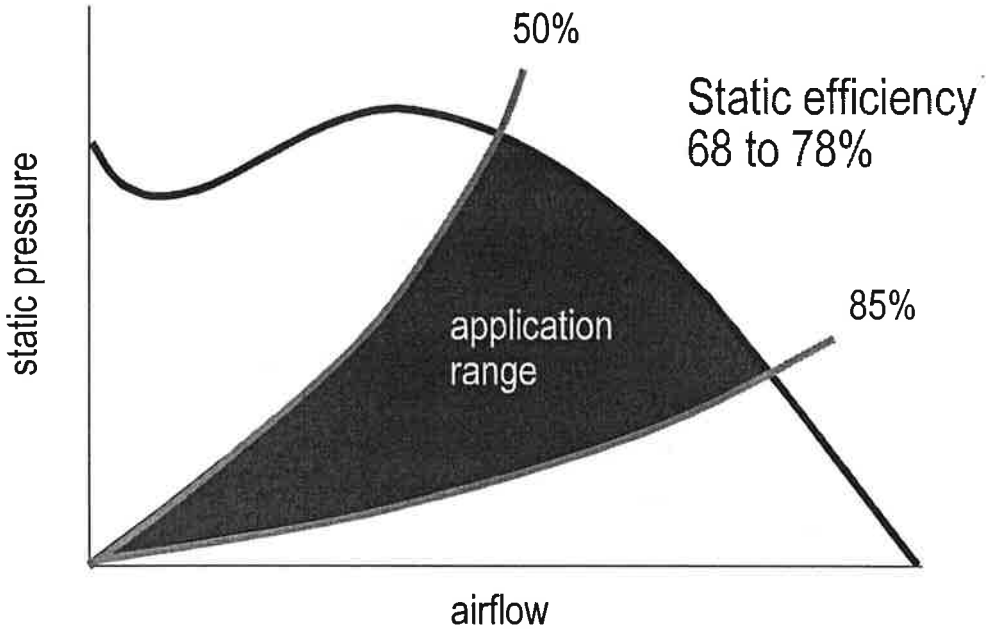
AHU centrifugal fan (Airfoil fan)



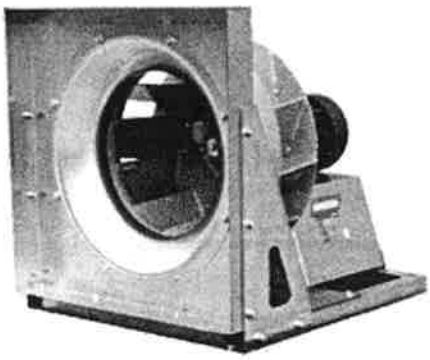


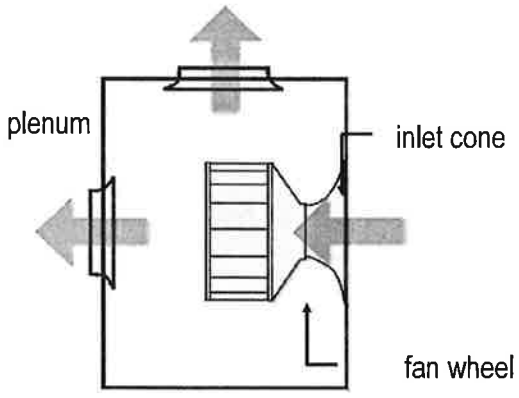
- A refinement of the BC fan changes the shape of the blade from a flat plate to that of an airfoil, similar to an airplane wing.
- The airfoil blade induces a smooth airflow across the blade surface, reducing turbulence and noise within the wheel. This results in increased static efficiency and reduced overall sound levels.

AHU centrifugal fan (Airfoil fan)



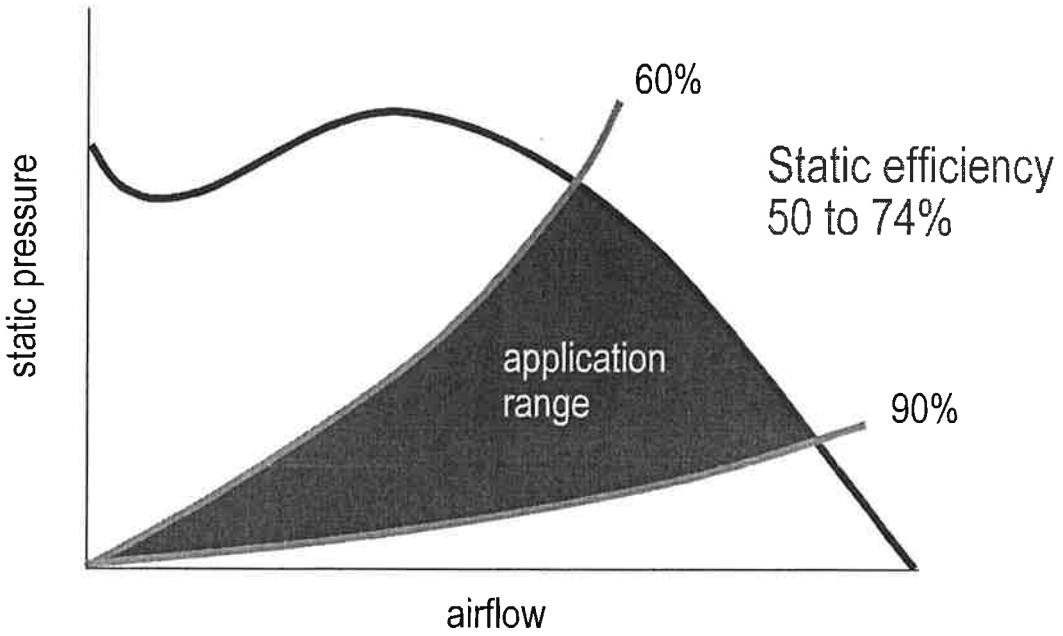
AHU Plenum fan





- Plenum fan consists of an unhoused fan wheel with airfoil fan blades and an inlet cone.
- The fan wheel pressurizes the plenum surrounding the fan, allowing the air to discharge in multiple directions. This fan can save space by eliminating turns in the ductwork and allowing multiple discharge ducts in different directions.
- Other advantages of plenum fan include being direct driven and being free of obstruction at suction and discharge.

AHU Plenum fan

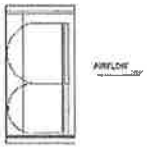
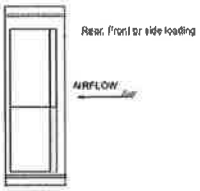


AHU filter rating

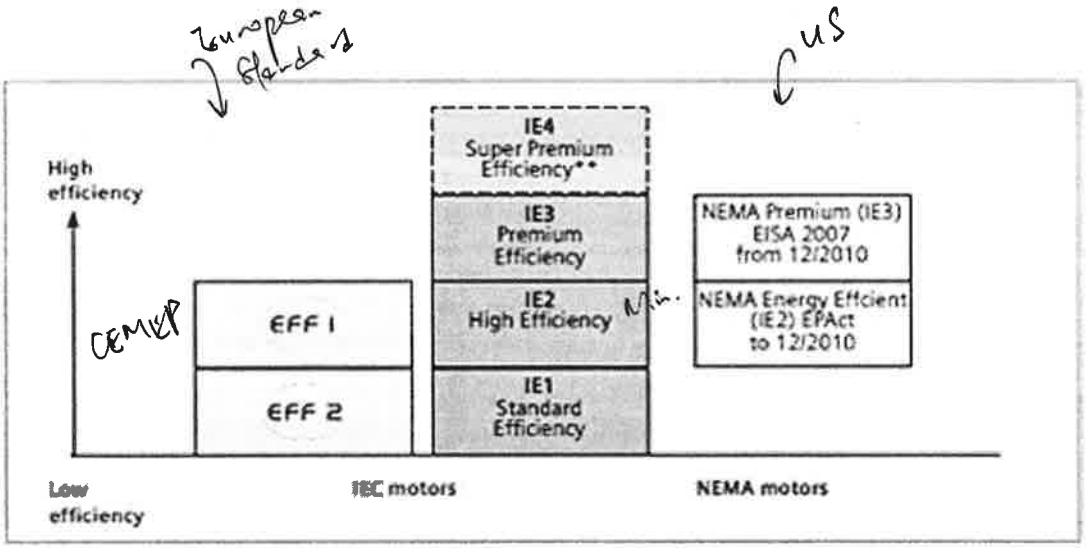
Ashrae 52.2 Filter rating: Minimum Efficiency Rating Value (MERV)

MERV VALUE	The filter will trap Average Particle Size Efficiency 0.3 - 1.0 Micron	The filter will trap Average Particle Size Efficiency 1.0 - 3.0 Micron	The filter will trap Average Particle Size Efficiency 3.0 - 10.0 Micron	Types of things these filters will trap
MERV 1	-	-	Less than 20%	Pollen, dust mites, standing dust, spray paint dust, carpet fibers
MERV 2	-	-	Less than 20%	
MERV 3	-	-	Less than 20%	
MERV 4	-	-	Less than 20%	
MERV 5	-	-	20% - 34%	Mold spores, hair spray, fabric protector, cement dust
MERV 6	-	-	35% - 49%	
MERV 7	-	-	50% - 69%	
MERV 8	-	-	70% - 85%	
MERV 9	-	Less than 50%	85% or better	Humidifier dust, lead dust, auto emissions, milled flour
MERV 10	-	50% - 64%	85% or better	
MERV 11	-	65% - 79%	85% or better	
MERV 12	-	80% - 89%	90% or better	
MERV 13	Less than 75%	90% or better	90% or better	Bacteria, most tobacco smoke, proplet Nuceli (sneeze)
MERV 14	75% - 84%	90% or better	90% or better	
MERV 15	85% - 94%	90% or better	90% or better	
MERV 16	95% or better	90% or better	90% or better	

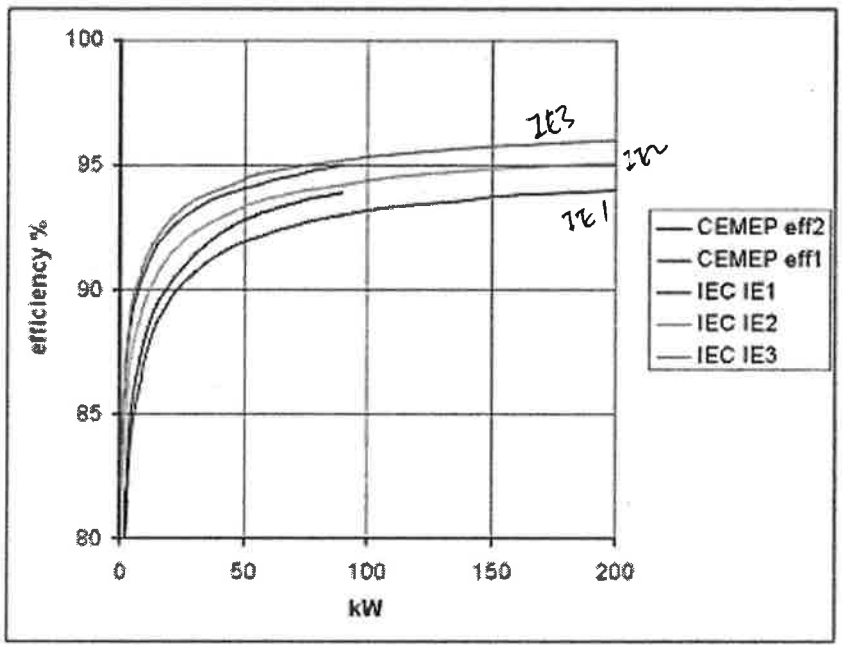
AHU filter types

Filter Types		MERV rating & pressure drop
	Flat Filter, washable / low eff media 50mm depth	MERV 1 to 4 Mid-life DP: 75pa
	Bag or Rigid Box Filter, med eff media 500mm depth	MERV 9 to 12 Mid-life DP: 150pa
	Bag or Rigid Box Filter High eff media 500mm depth	MERV 13 to 16 Mid-life DP: 150pa

Motor Efficiency

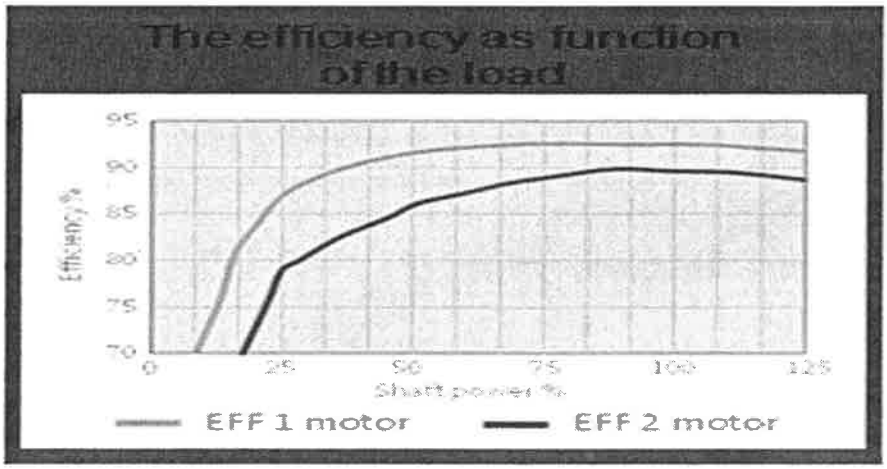


Motor Efficiency



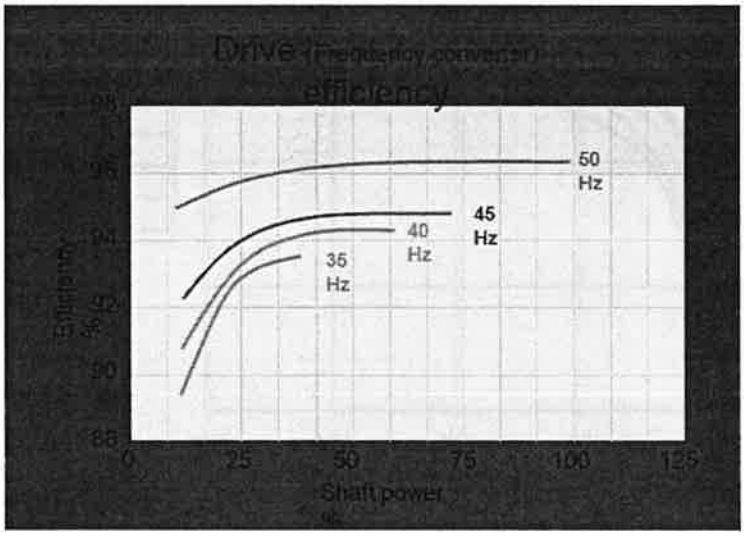
Motor efficiency

- The motor efficiency is dependent on the type of motor (EFF1 or EFF2) and the load of the motor (shaft power). A low load (below 25%) means a significant drop in efficiency.



Variable Speed Drive Efficiency

- The drive efficiency is dependent on the load of motor and the frequency output, but the efficiency down to 30% load is more or less constant.

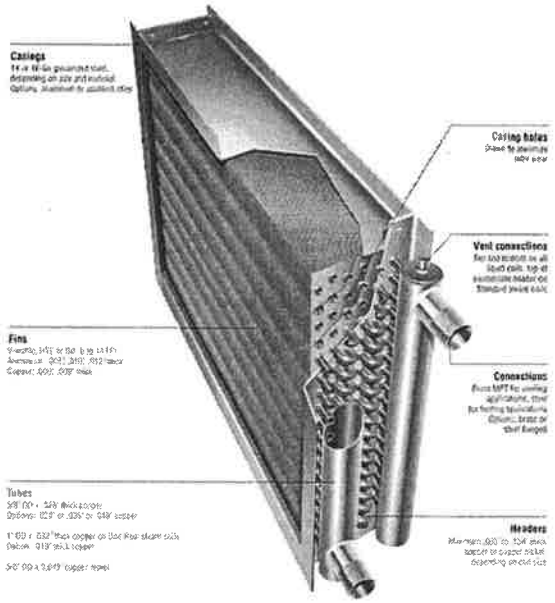


Fan Power

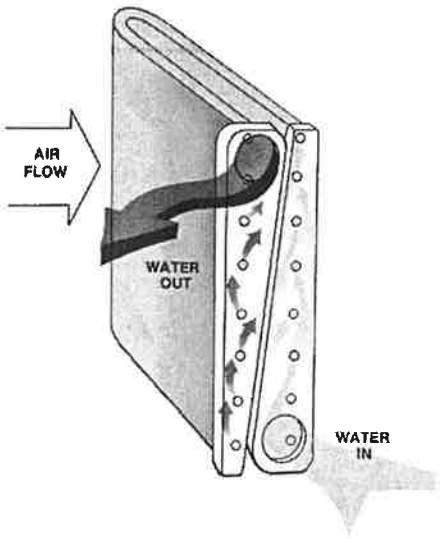
$$\text{Fan power} = \frac{\text{Flow rate x Total Pressure}}{3600 \times 1000 \times \eta\text{fan} \times \eta\text{motor}}$$

Where:
Fan power: KW
Flow rate: CMH
Total Pressure: Pa
ηfan: fan efficiency (%)
ηmotor: Motor efficiency (%)

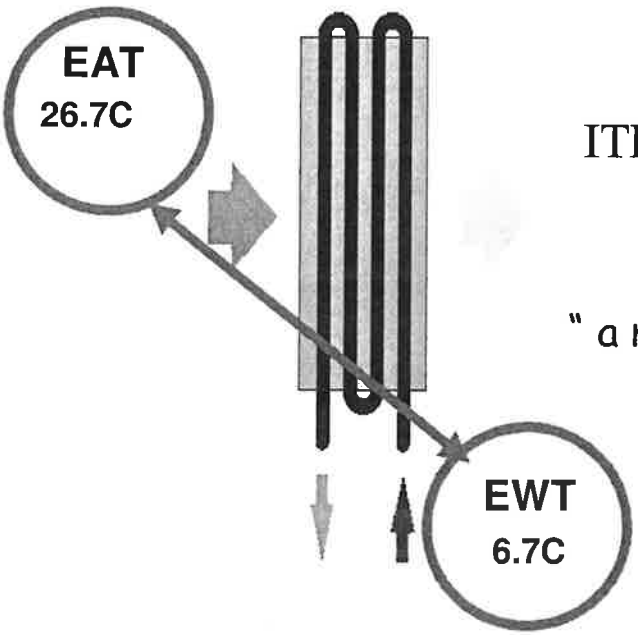
Cooling coils



Counter flow design



Coil initial temperature difference (ITD)

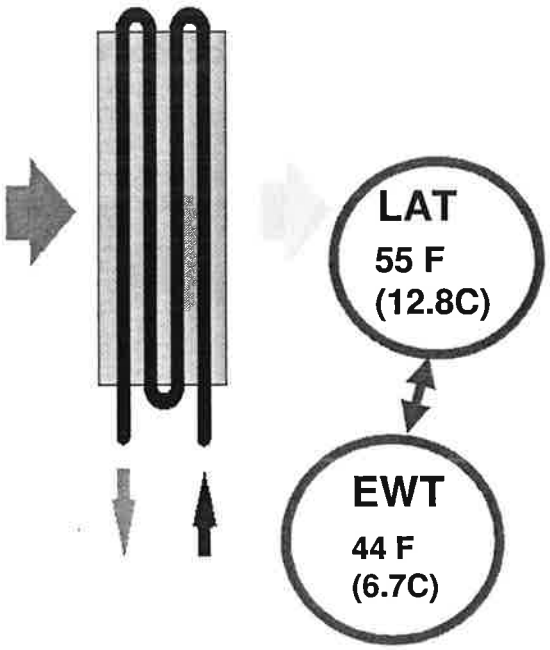


$$ITD = EAT - EWT$$

(hottest - coldest)

" a measure of coil capacity "

Coil Approach



APR = LAT - EWT

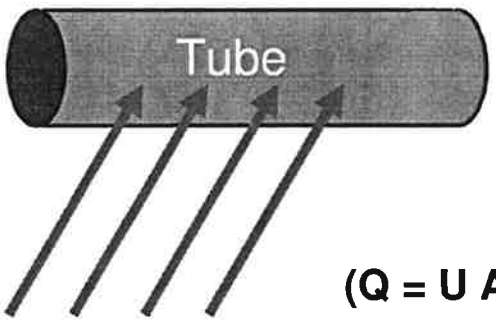
" a measure of coil efficiency "

(how close can you get?)

Coil heat transfer



$(Q = U\ A\ \Delta t)$



$(Q = U\ A\ \Delta t_{LM})$

Coil heat transfer

$$Q = UA \quad LMTD$$

Mostly hardware

Mostly application

Q: Heat transfer, Btu/H (KW)

U: Heat transfer coeff , Btu/H.ft^2.F (W/m^2.K)

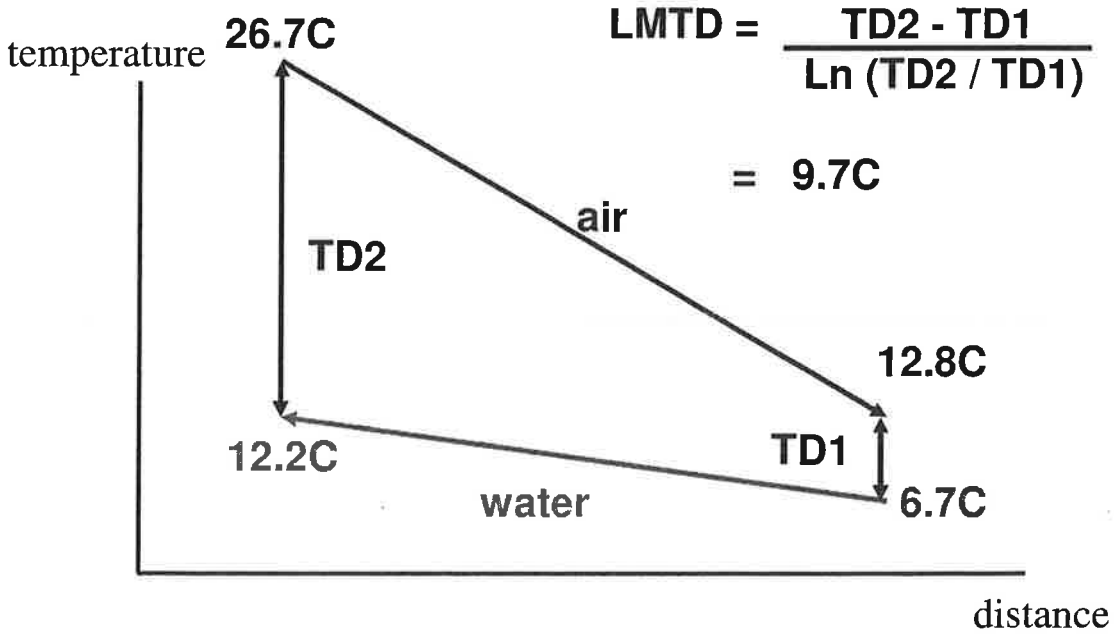
A: Surface area for heat transfer Ft^2 (m^2)

LMTD: Log-mean temperature diff, F (C)

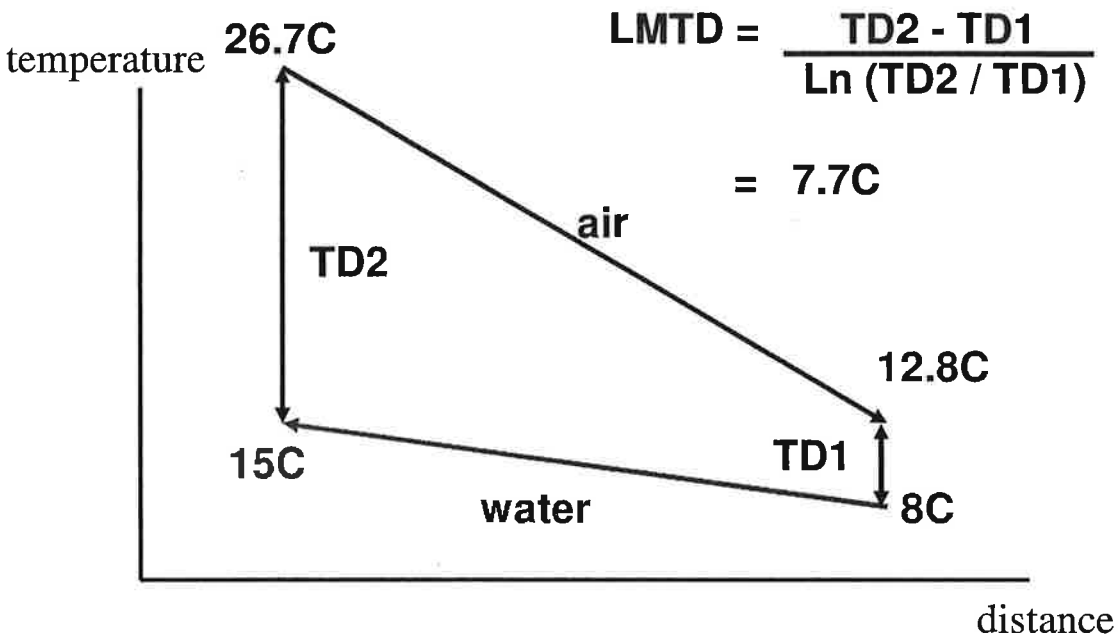
LMTD for standard CHW temp

$$LMTD = \frac{TD2 - TD1}{Ln\ (TD2 / TD1)}$$

$$= 9.7C$$

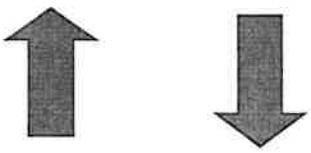


LMTD for high CHWS temp and high DT



LMTD for high CHWS temp and high DT

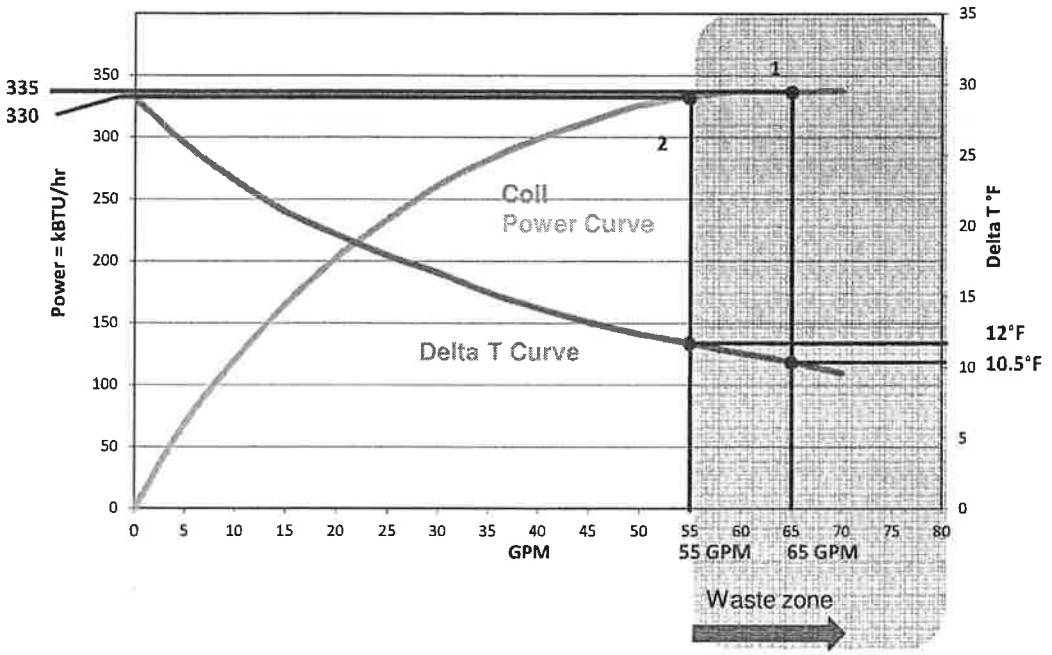
Q = UA LMTD



<u>To compensate for lower LMTD</u>
- Need for fins per feet (FPF)
- Need more rows

<u>Implications</u>
- Increase in cost
- Increase in air pressure drop
- Increase in water pressure drop

Cooling coil part load performance



Fan Coil Unit (FCU)

Fan Coil Unit (FCU)



Ceiling concealed ducted Type



Cassette type



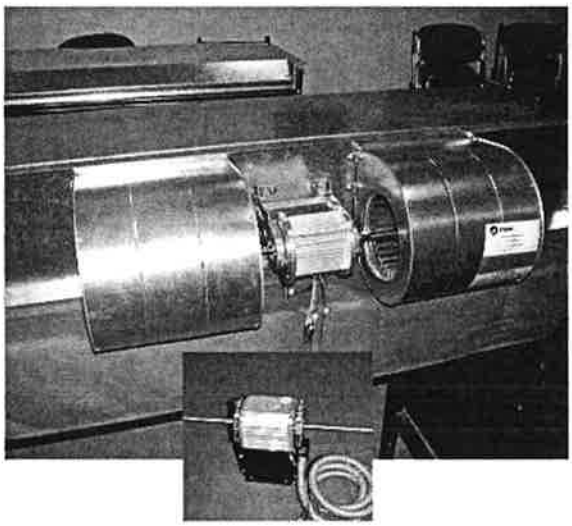
Ceiling suspended type



Wall type

VAV FCU with DC motor

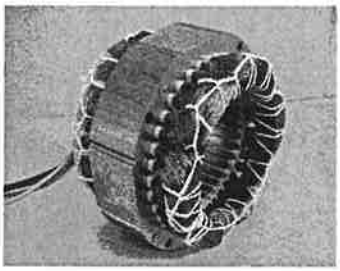
by itself variable speed PSC



Integrated signal driver to the DC (brushless) fan motor

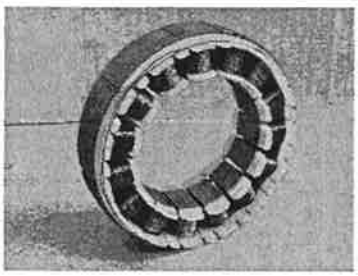
- *Substantial energy saving*
  - **Almost double efficiency than PSC motor**
- *Stepless control*
  - **Non-stage variable speed**
- *Quiet*
  - **Low electro-magnetic noise**
- *Comfortable*
  - **Constant temperature**
- *PI control*
  - **Fast response to the load changes**

VAV Fan Coil Unit -- Motor Coil



Traditional AC Motor

- Less reliable
- Lower energy efficient (35-50% FL eff)
- Lower 1st cost



DC brushless Motor

- More reliable
- Higher efficiency (65-70% FL eff)
- Higher 1st cost (aprrox 2 x vs ac motor)

Energy Efficiency comparison

Table 1. Simple payback for replacing three-speed permanent split-capacitor AC (PSC-AC) induction motors with brushless DC (BLDC) motors in fan–coils [a,b]

Nominal airflow, cfm	First cost (est), $ USD		Motor efficiency, %		Energy input, watts			Annual energy cost savings, $ USD	Simple payback, yr
	PSC-AC (std)	BLDC (opt)	PSC-AC	BLDC	PSC-AC	BLDC	Energy saved		
200			35	65	55.5	30.0	25.5	16.29	8.6
300			35	70	80.2	39.0	41.2	26.35	5.4
400	—	+70	35	70	96.4	49.1	47.3	30.29	4.6
600			50	70	115.2	79.1	36.1	23.12	6.0
800			50	70	124.3	93.2	31.2	19.95	7.0

[a] This comparison isn't exactly "apples to apples": The PSC-AC motors varied in size, with the smallest fan–coil using a 1/20 hp motor; but in each case, the replacement was a 1/3 hp BLDC motor.

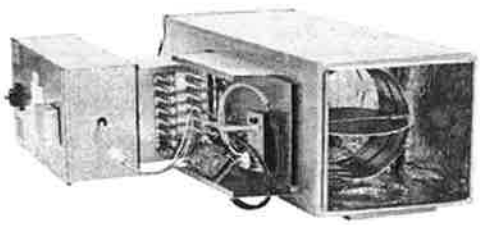
[b] Saving calculations are based on annual operation of 4,000 hours and an energy cost of $0.08/kWh.

Extracted from Trane *Engineers Newsletter* volume 33–4 (Lit. no. ADM-APN013-EN)

Variable Air Volume Box (VAV Box)

VAV box

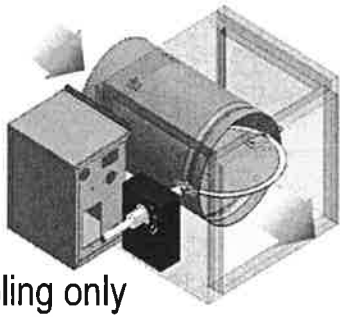
- VAV box provides a variable volume of air to match the cooling demand
- Designed to provide a constant supply air temperature to the space, whilst VAV box vary the amount of air (volume) to the space
- VAV boxes control the space temperature by modulating a damper
- The speed of the AHU fan/motor changes to match the system static pressure
  - Saves energy at reduced speeds



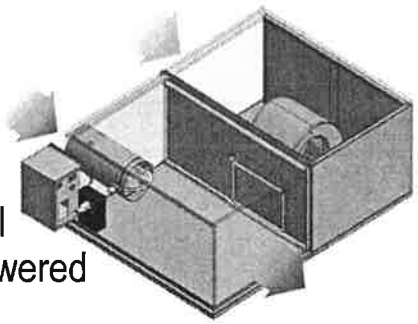


VAV controller

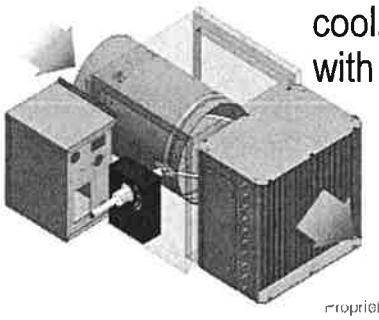
VAV box types



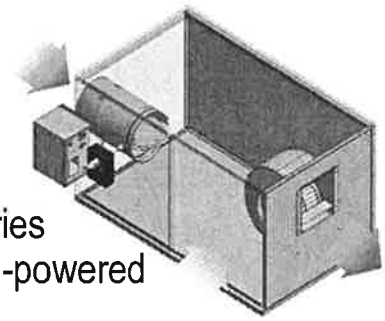
cooling only



parallel
fan-powered



cooling
with reheat



series
fan-powered

THANK YOU



Pipe & Duct Design

Steven Kang
Steven.kang@mnv.com.sg

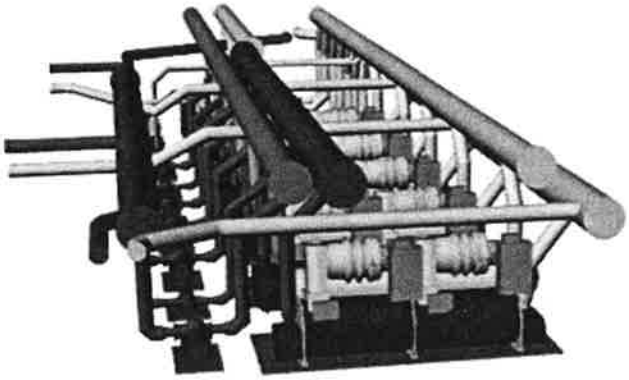
Agenda

Pipe design
1. Introduction to Pipe Design
2. Pipe Circuit Layout
3. Pipe Design Considerations
4. Pipe Pressure Drop & Sizing
5. Pump Head Calculation
6. Case Comparison

Duct design
1. Introduction to Duct Design
2. Duct Design Considerations
3. Duct Pressure Drop & Duct Sizing

INTRODUCTION TO PIPE DESIGN

Pipe Design is an art as much as it is a science



Why Good Ducting & Piping Design is Important?

1. Design optimized system to save energy & reduce CO2 emissions
2. Allows selection of high efficiency fan & pump to save energy
3. Ensure good air & water circulation for comfort
4. Minimize space requirements
5. Minimize noise
6. Minimize 1st cost

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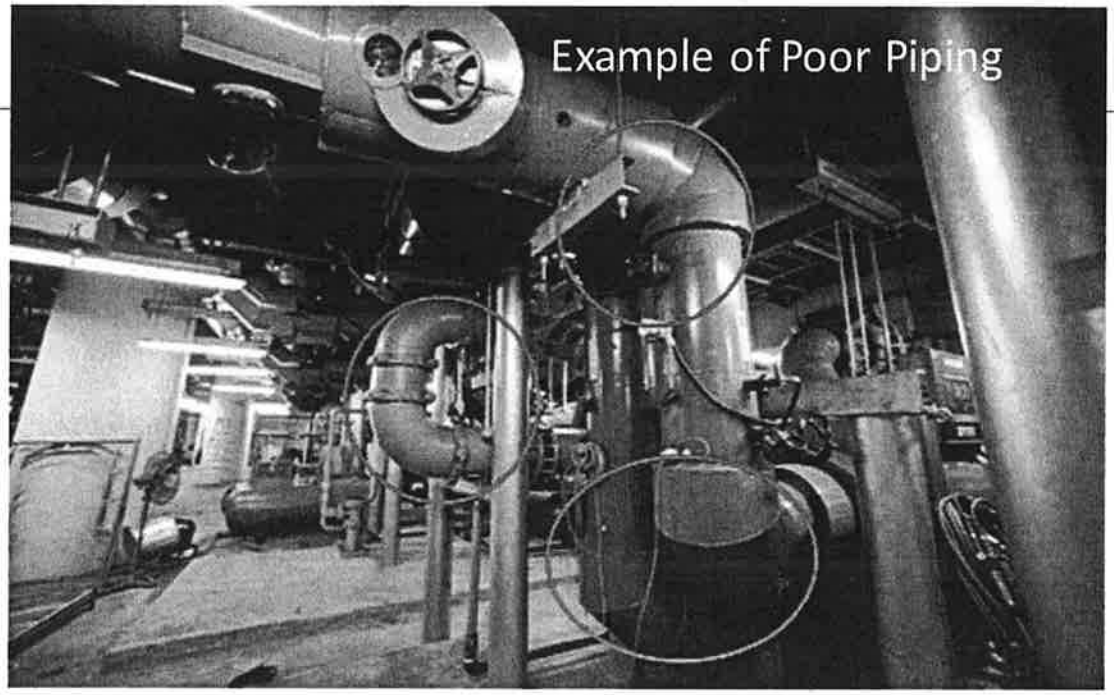

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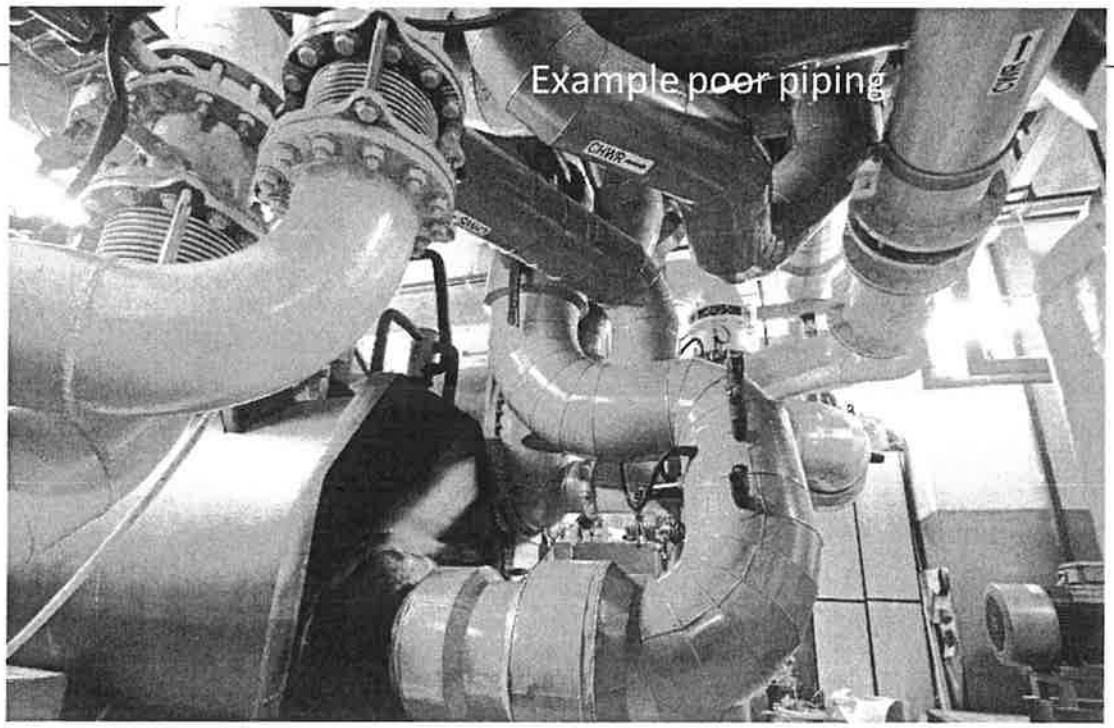
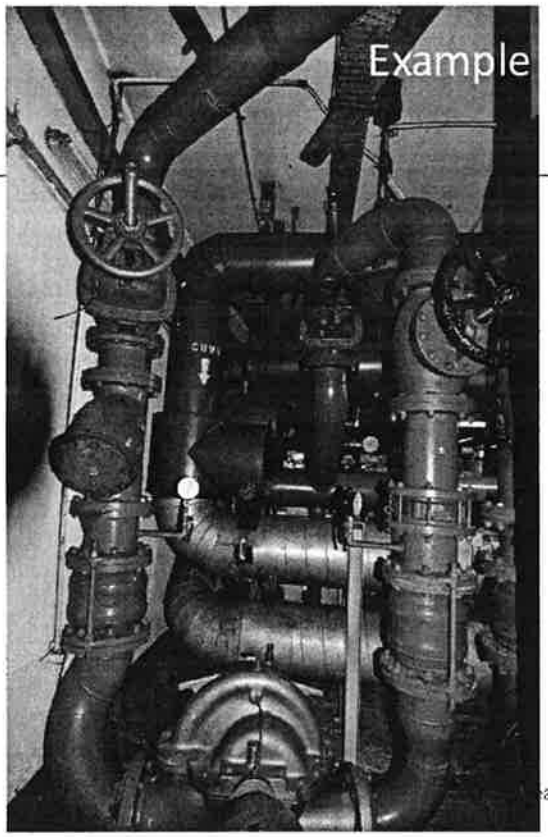
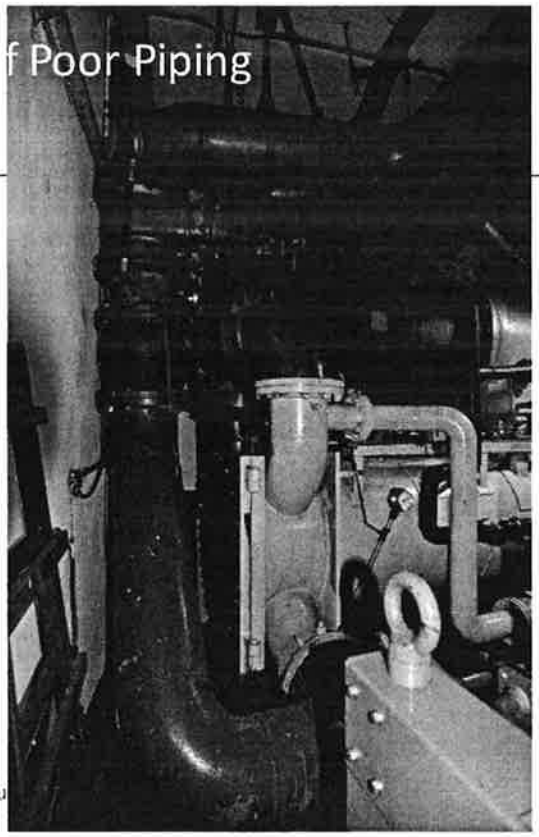


Example of Poor Piping





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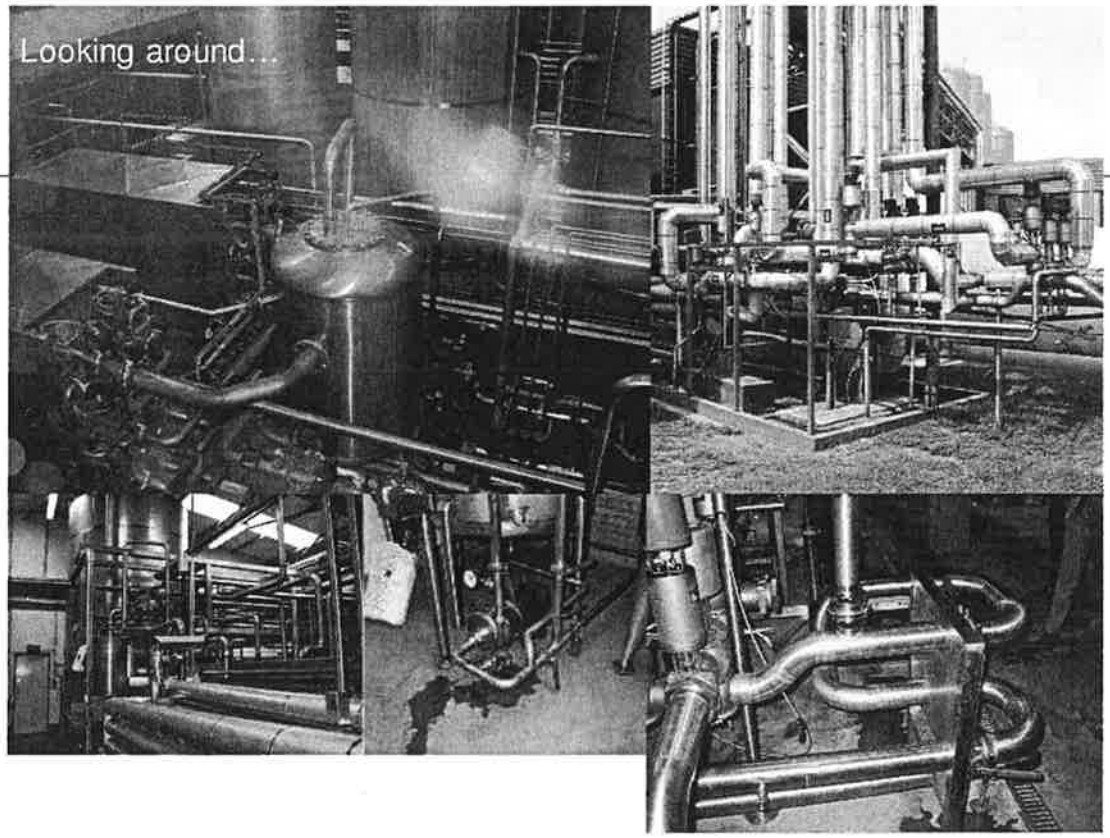
Looking around...



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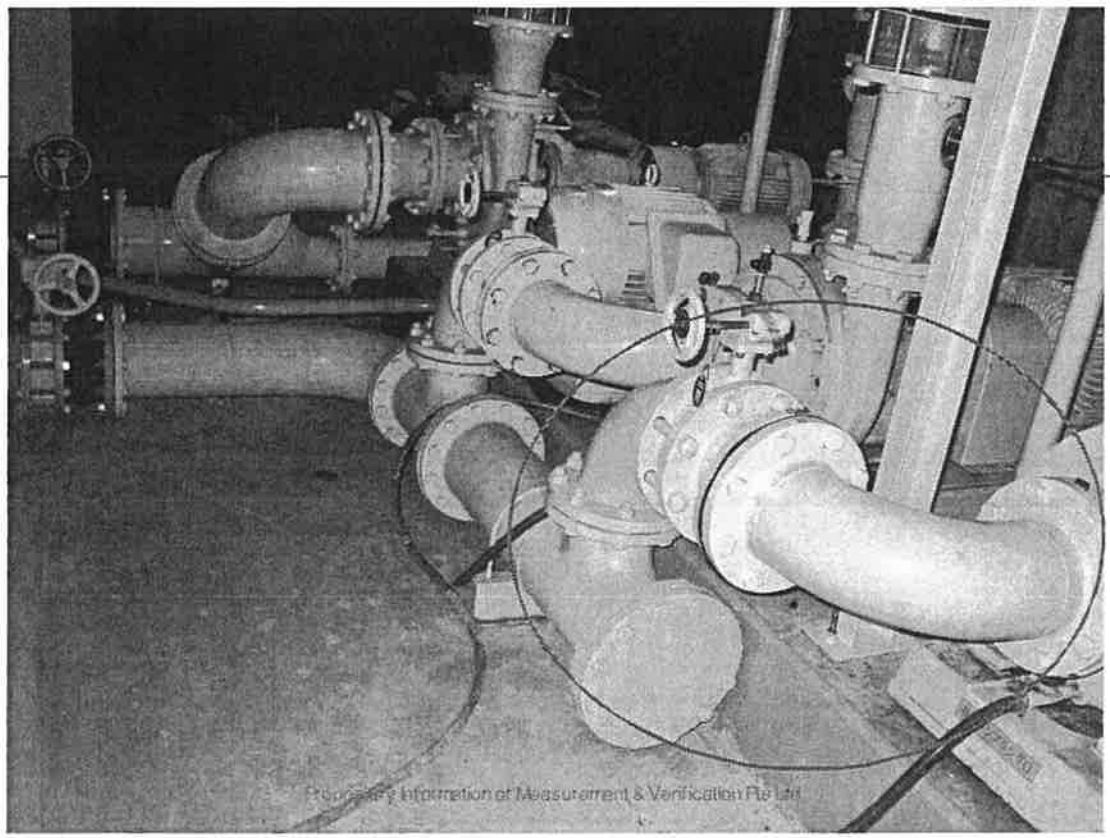

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New Design Mentality



1. Big Pipes, Small Pumps

New Design Mentality

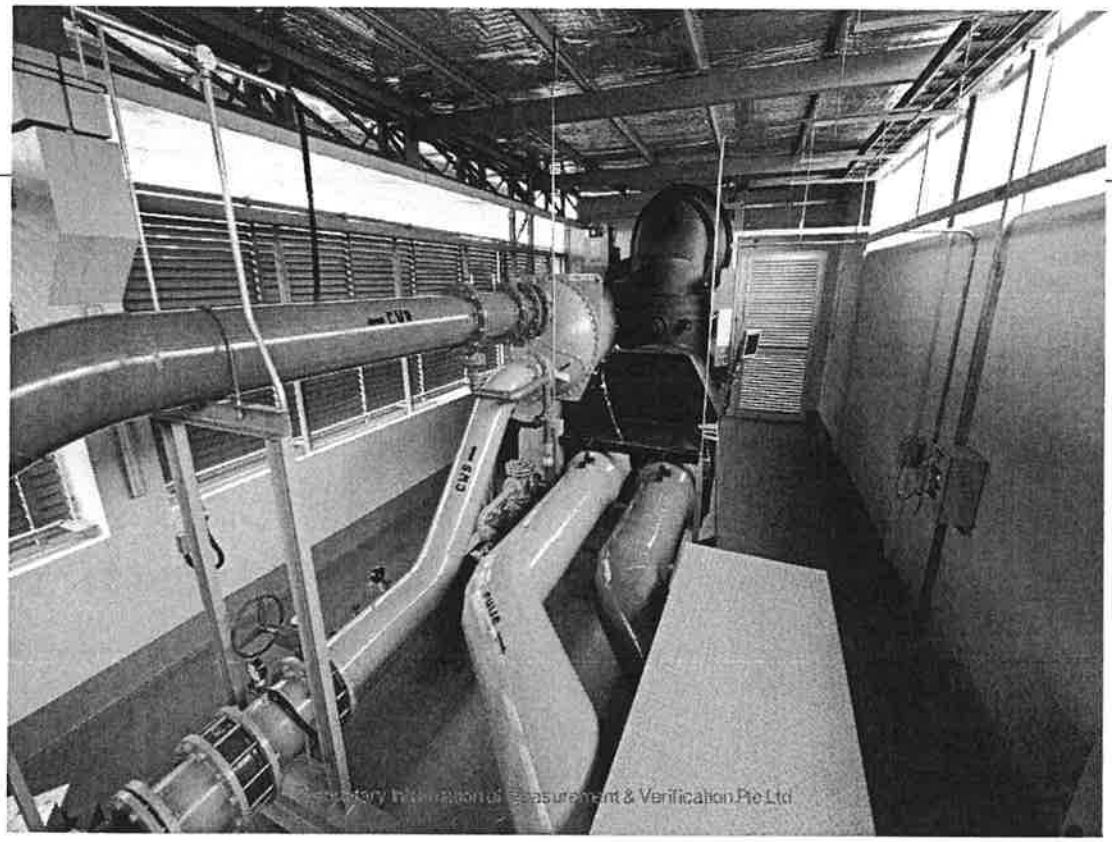


2. Lay out the pipes first, then the equipment

Fat, short, straight pipes

Not skinny, long, crooked pipes!

No new technologies, just two design changes



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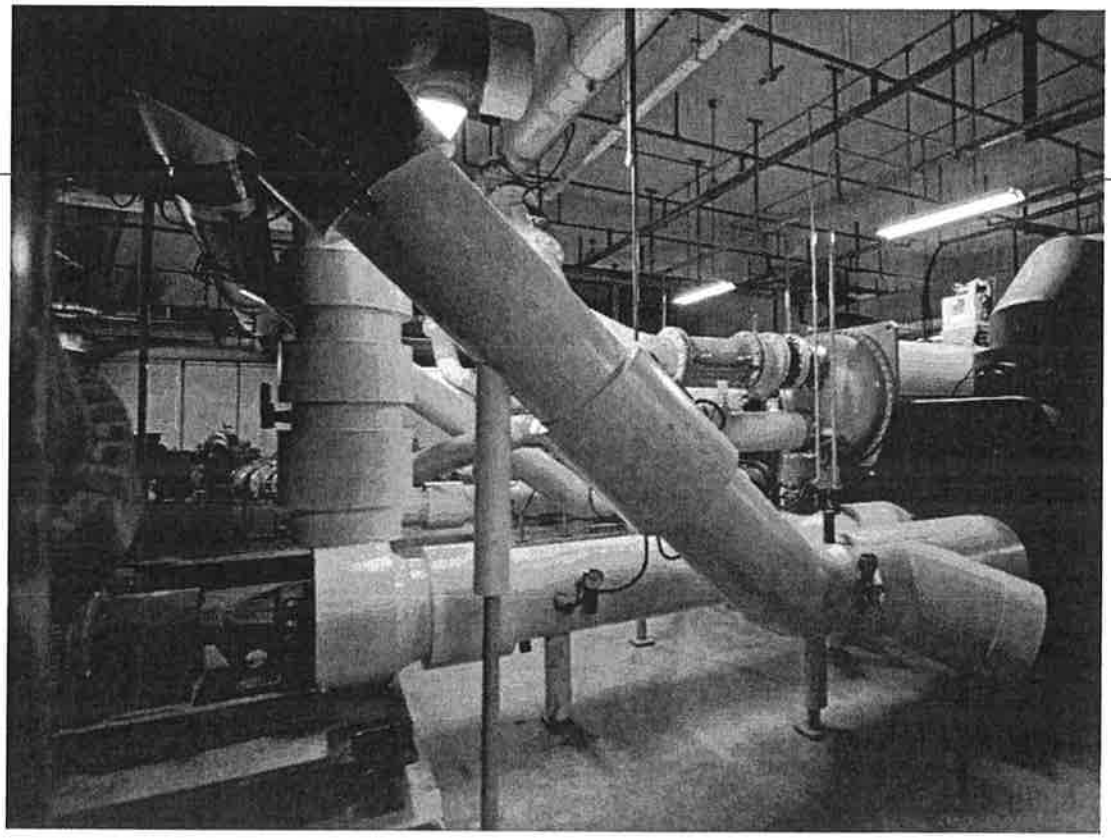

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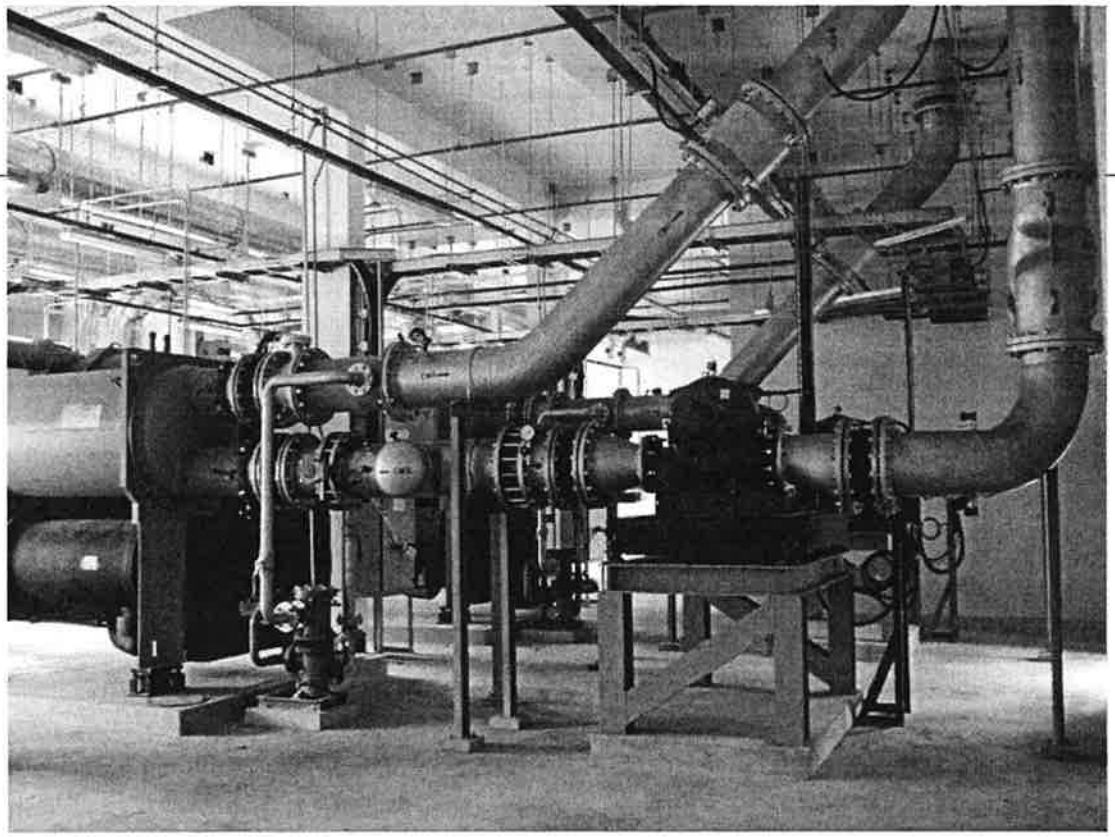

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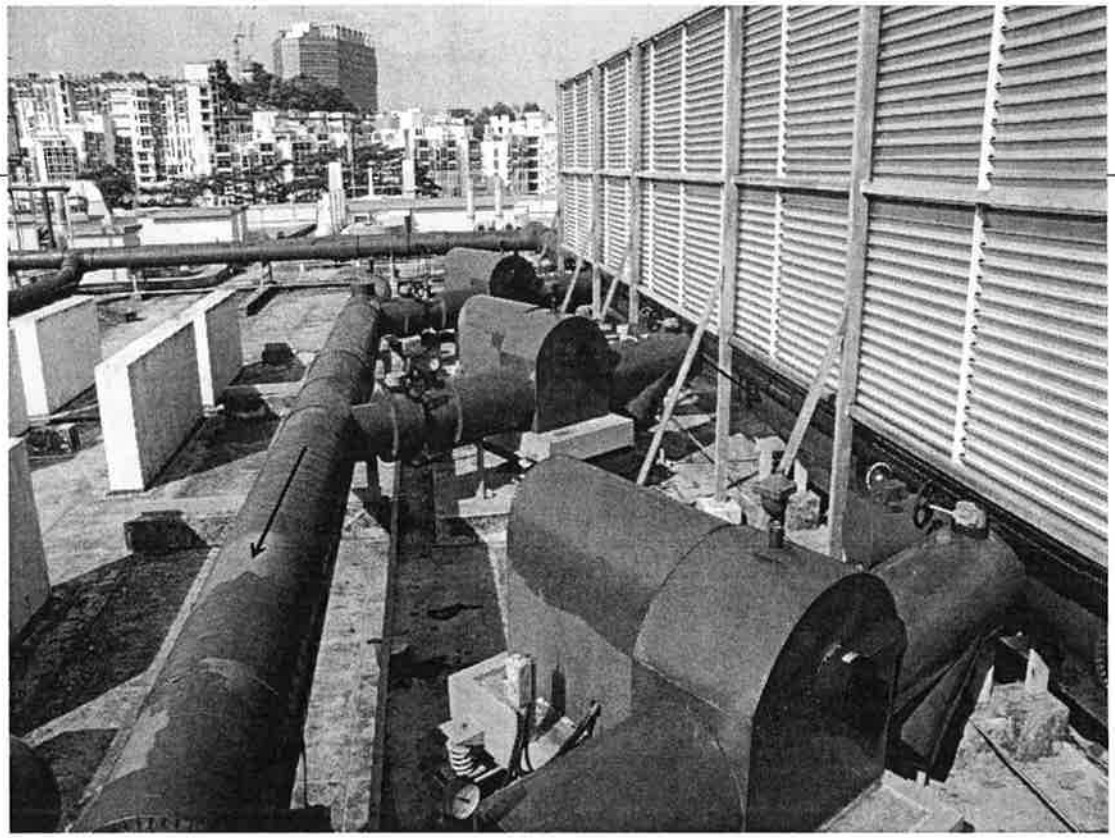

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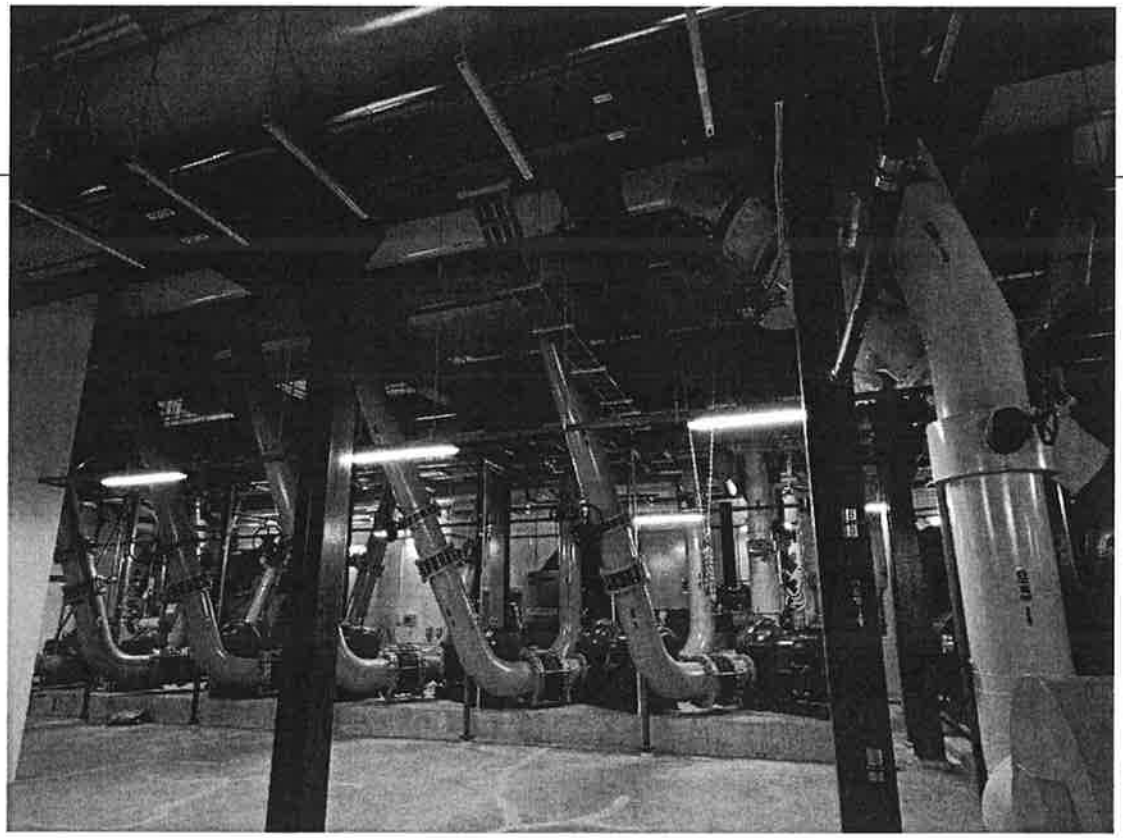

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PIPE CIRCUIT LAYOUT

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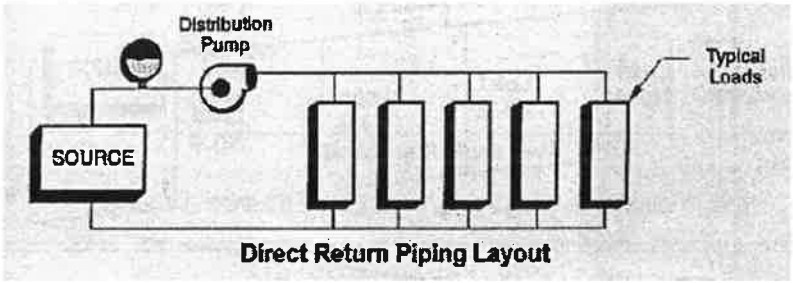
Pipe Circuit Layout

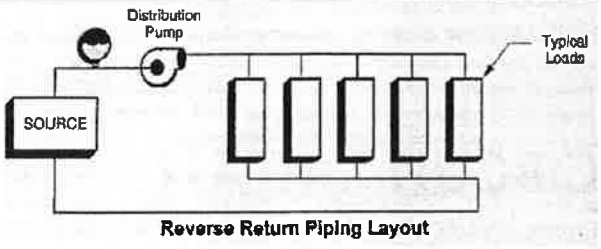


Direct Return Piping Layout

- The length of supply and return piping through the sub-circuits is unequal. It may cause unbalanced flow rates and require careful balancing to provide each sub-circuit with design flow

- Widely applied by using control valves, providing balancing devices, and minimizing distribution piping pressure drop.

Pipe Circuit Layout



Reverse Return Piping Layout

- This layout provides nearly equal total lengths for all terminal circuits by using the extra piping to balance pressure drops between nearby and faraway coils. The closest coil has the shortest supply piping and the longest return piping which the farthest coils has the opposite, requires longer piping length which would drive the cost higher

- Not suitable for bigger chilled water loops having different sub-circuit head losses.

PIPE DESIGN CONSIDERATIONS

Pipe Design Considerations

Good piping design always balances first cost against future operating cost, taking the erosion, noise generation, installation into consideration.

- Location of pipe in the system: critical circuit that determines pump head
- First costs of installed piping
- Pump energy cost: pump motor efficiency, distribution system type, type of pump control, annual flow profile
- Erosion considerations: high velocities can contribute to hastening of pipe wall deterioration
- Noise considerations: control noise generation from turbulence, cavitation, release of entrained air, and water hammer
- Physical constraints
- Budget constraints

Pipe Design Considerations

Code requirements

Type	Load Condition	Min eff	Units	Remarks
CHWP	Full load	349	KW / m^3/s	
		0.053	KW / RT	=(349KW/m^3/s) x(2.4gpm/RT) x (0.0000631 m^3/s / gpm)
	Part load	Motors exceeding 15KW must modulate to 50% of design flow @ no more than 30% of design KW		
CWP	Full load	301	KW / m^3/s	
		0.057	KW / RT	=(301KW/m^3/s) x (3gpm/RT) x (0.0000631 m^3s / gpm)
	Part load	Motors exceeding 15KW must modulate to 50% of design flow @ no more than 30% of design KW		

PIPE PRESSURE DROP & PIPE SIZING

Pipe Pressure Drop Equations

Darcy-Weisbach Equation

Predicts the pressure drop or " head loss" of a steady-state, incompressible fluid flowing through a length of pipe.

$$H = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right)$$

where...
H = head loss, ft wg
f = friction factor, dimensionless
L = actual pipe length plus equivalent length for valves and fittings, ft
V = fluid velocity, fps
D = pipe diameter, ft
g = gravitational acceleration constant, 32.1740 ft/s^2

Pipe Pressure Drop Equations

Moody formula

Uses the Reynolds number and the roughness factor of the pipe, when known, to determine the friction factor.

$$f = 0.0055\left[1 + \left(20{,}000\,\frac{\varepsilon}{D} + \frac{1{,}000{,}000}{Re}\right)^{0.33}\right]$$

where...
f = friction factor, dimensionless
ε = pipe roughness, ft
D = pipe diameter, ft
Re = Reynolds number, dimensionless

$$Re = \frac{D \times V \times \rho}{\mu}$$

where...
D = diameter, ft
V = fluid velocity, fps
ρ = density, lb/cu ft
μ = dynamic viscosity, lb/ft-sec

Pipe Friction Calculation

1. For simplicity, a lot of pressure loss charts for different pipes are presented in pipe design handbooks.

   Ps = Length * Pressure drop per length

2. ASHRAE general design guide :

- Pipe friction loss used is between 1 and 4 ft of water per 100 ft of pipe, wider ranges may be used in specific designs if certain precautions are taken
- Pipe velocity limit is 4 fps for 2 in. pipe and smaller. Velocities in excess of 4 fps can be used in piping of larger size.

3. It is designer's responsibility to determine pipe size, maximum velocity, and the friction loss.

1 to 2 ft / 100 ft

ASHRAE Standard 90.1-2007 Addendum

Operating hr/yr	<=2,000 hr/year		>2,000 and ≤4,400 hr/year		>4,400 and ≤8,760 hr/year	
Nominal Pipe Size (in.)	Other	Variable Flow/ Variable Speed	Other	Variable Flow/ Variable Speed	Other	Variable Flow/ Variable Speed
2 1/2	120	180	85	130	68	110
3	160	270	140	210	110	170
4	350	530	260	400	210	320
5	410	620	310	470	250	370
6	740	1,100	570	860	440	680
8	840	1,300	650	970	510	770
10	1,800	2,700	1,300	2,000	1,000	1,600
12	2,500	3,800	1,900	2,900	1,500	2,300
Maximum Velocity for Pipes Over 12 in. Size	8.5 fps	13.0 fps	6.5 fps	9.5 fps	5.0 fps	7.5 fps

Table 2: From Standard 90.1-2007, Table 6.5.4.5 "Piping System Design Maximum Flow Rate in Gallons Per Minute."

Recommended max flow rate in gpm

CHART 3—FRICTION LOSS FOR CLOSED PIPING SYSTEMS

Schedule 40 Pipe

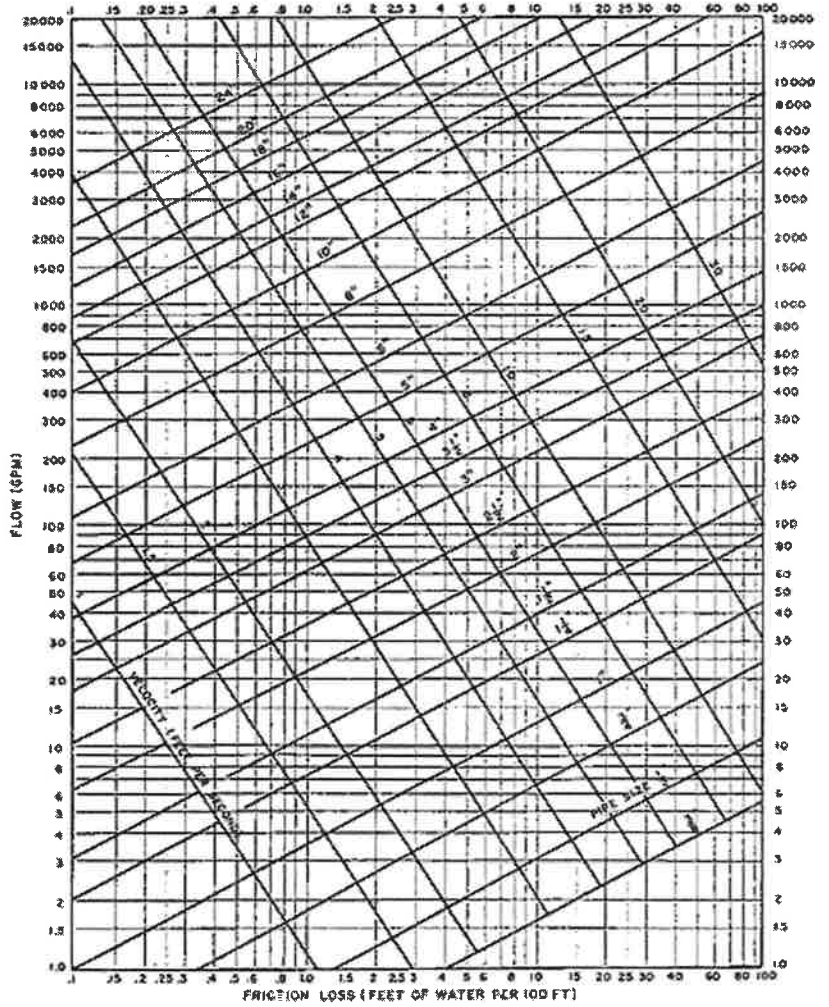
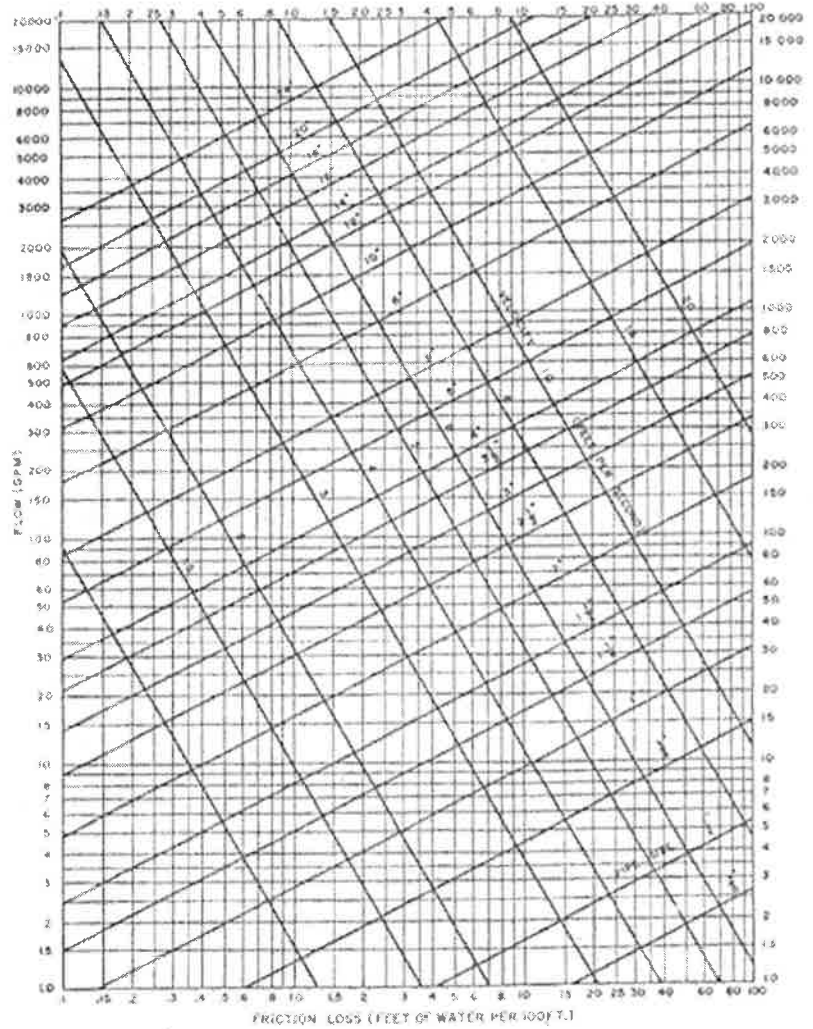


CHART 4—FRICTION LOSS FOR OPEN PIPING SYSTEMS

Schedule 40 Pipe



Pipe Fitting Pressure Drop

1. Equivalent length: determine the pressure drop for the fittings in a length of pipe, convenient to use, ASHRAE published equivalent lengths for various fittings and tabulated against diameter and velocity.

   Pf = Equivalent length * Pressure drop per length

2. Loss coefficient : dependent solely on velocity which is governed by the actual internal diameter of the particular fitting

IP Units

$$Pf = \frac{K \times Q^2}{D^4 \times 385}$$

Pf : Friction loss, FT WG
K: Friction factor
Q: Flow rate, GPM
D: Pipe diameter, in

SI Units

$$Pf = \frac{800000 \times K \times Q^2}{D^4}$$

Pf : Friction loss, KPa
K: Friction factor
Q: Flow rate, L/S
D: Pipe diameter, mm

Equivalent Length Table

Nom. Pipe Size (inches)	Angle Valve	Ball Valve	Check Valve	Gate Valve	Globe Valve	Elbow	Strainer
3/8	6.0	1.4	5.0	0.6	17.0	0.9	n/a
1/2	7.0	1.6	6.0	0.7	18.0	1.0	3.0
3/4	9.0	2.0	8.0	0.9	22.0	1.4	4.0
1	12.0	2.6	10.0	1.0	29.0	1.7	5.0
1-1/4	15.0	3.3	14.0	1.5	38.0	2.3	9.0
1-1/2	18.0	4.0	16.0	1.8	43.0	2.6	10.0
2	24.0	5.0	20.0	2.3	55.0	3.3	14.0
2-1/2	29.0	6.0	25.0	2.8	69.0	4.1	20.0
3	35.0	7.5	30.0	3.2	84.0	5.0	40.0
4	47.0	10.0	40.0	4.5	120.0	6.7	60.0
5	58.0	13.0	50.0	6.0	140.0	8.2	80.0
6	70.0	16.0	60.0	7.0	170.0	10.0	110.0
8	85.0	20.0	80.0	9.0	220.0	13.0	150.0
10	105.0	25.0	100.0	12.0	280.0	16.0	190.0
12	130.0	30.0	120.0	13.0	320.0	19.0	250.0
14	155.0	34.0	135.0	15.0	360.0	23.0	n/a
16	180.0	38.0	150.0	17.0	410.0	25.0	n/a
18	200.0	42.0	165.0	19.0	460.0	29.0	n/a
20	235.0	50.0	200.0	22.0	520.0	33.0	n/a
24	265.0	60.0	240.0	25.0	610.0	40.0	n/a

Units: FT

from 1985 *ASHRAE Handbook of Fundamentals*, Chapter 34

note
- ⇒ All valves, regardless of type, are assumed to be in the full-open position.
- ⇒ Losses for check valves apply to both swing-type and ball-type designs.
- ⇒ Losses for angle valves and globe valves do not apply designs with needle-point seats.
- ⇒ Losses for elbows apply to 90-degree "long" designs with a radius-to-diameter ratio approximating 1.5.

Loss Coefficient K Charts

Table 1 *K* Factors—Threaded Pipe Fittings

Nominal Pipe Dia., in.	90° Standard Elbow	90° Long-Radius Elbow	45° Elbow	Return Bend	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve	Bell Mouth Inlet	Square Inlet	Projected Inlet
3/8	2.5	—	0.38	2.3	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
1/2	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
3/4	1.7	0.92	0.35	1.7	0.90	2.1	10	0.28	6.1	3.7	0.05	0.5	1.0
1	1.5	0.78	0.34	1.5	0.90	1.8	9	0.24	4.6	3.0	0.05	0.5	1.0
1-1/4	1.3	0.65	0.33	1.3	0.90	1.7	8.5	0.22	3.6	2.7	0.05	0.5	1.0
1-1/2	1.2	0.54	0.32	1.2	0.90	1.6	8	0.19	2.9	2.5	0.05	0.5	1.0
2	1.0	0.42	0.31	1.0	0.90	1.4	7	0.17	2.1	2.3	0.05	0.5	1.0
2-1/2	0.85	0.35	0.30	0.85	0.90	1.3	6.5	0.16	1.6	2.2	0.05	0.5	1.0
3	0.80	0.31	0.29	0.80	0.90	1.2	6	0.14	1.3	2.1	0.05	0.5	1.0
4	0.70	0.24	0.28	0.70	0.90	1.1	5.7	0.12	1.0	2.0	0.05	0.5	1.0

Source: Engineering Data Book (Hydraulic Institute 1979).

Table 2 *K* Factors—Flanged Welded Pipe Fittings

Nominal Pipe Dia., In.	90° Standard Elbow	90° Long-Radius Elbow	45° Long-Radius Elbow	Return Bend Standard	Return Bend Long-Radius	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve
1	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	—	4.8	2.0
1-1/4	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	—	3.7	2.0
1-1/2	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	—	3.0	2.0
2	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
2-1/2	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
3	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
4	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
6	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
8	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
10	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
12	0.24	0.13	0.16	0.24	0.13	0.08	0.50	5.7	0.05	2.1	2.0

Source: Engineering Data Book (Hydraulic Institute 1979).

PUMP HEAD CALCULATION

Pump head Calculation

$$H_p = H_L + H_E + H_F$$

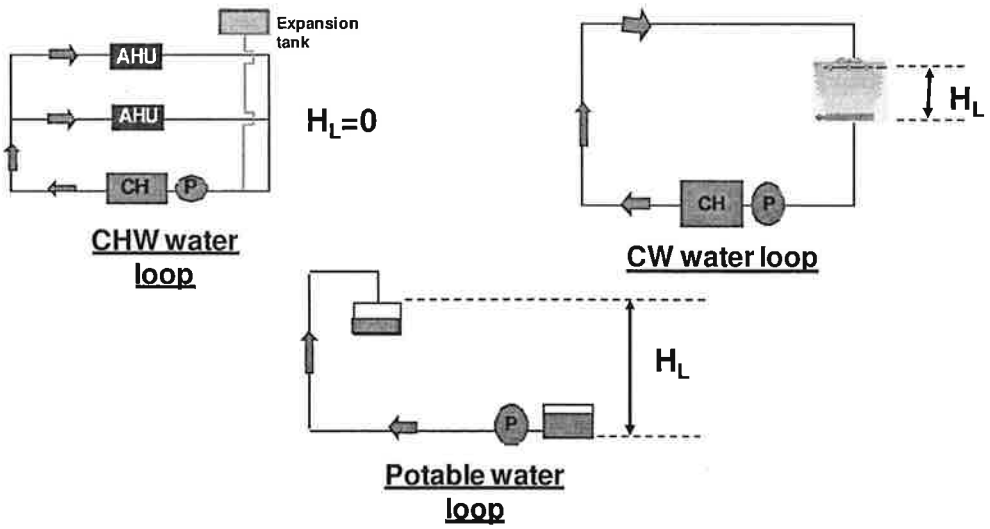
H_p: Pump head

H_L: Static lift

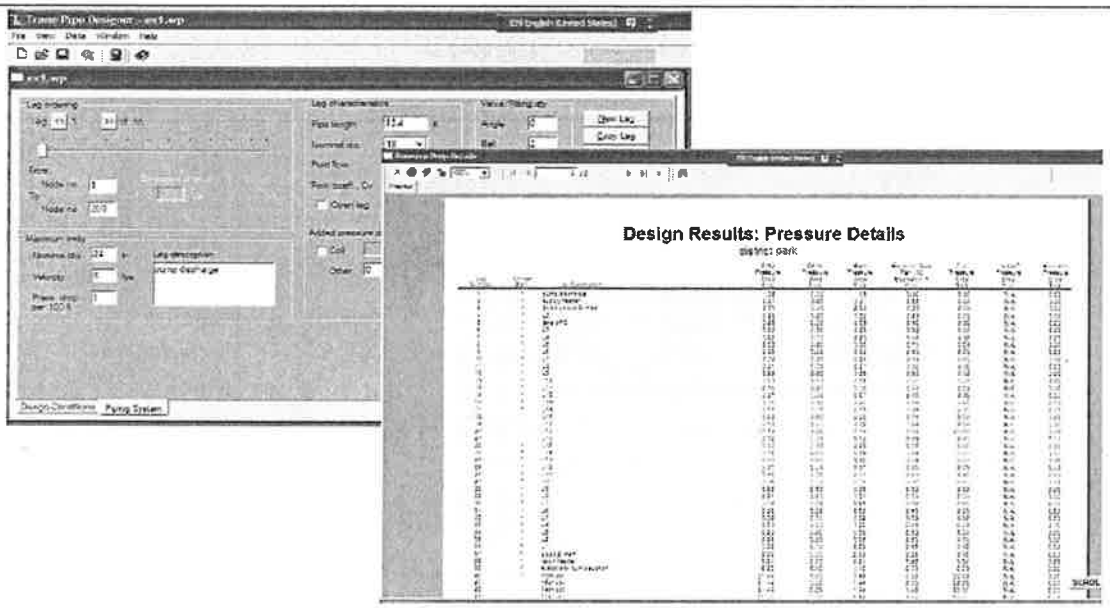
H_E: Equipment friction loss at design flow

H_F: Pipe and fitting friction loss at design flow

Pump head calculation



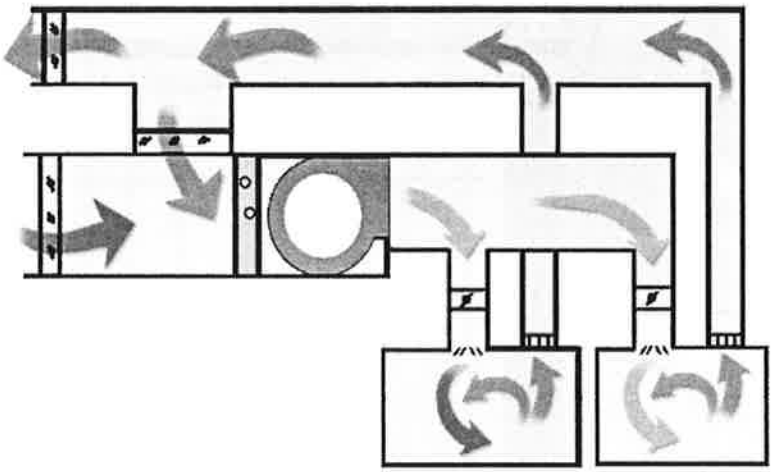
Computer Software Pipe Designer



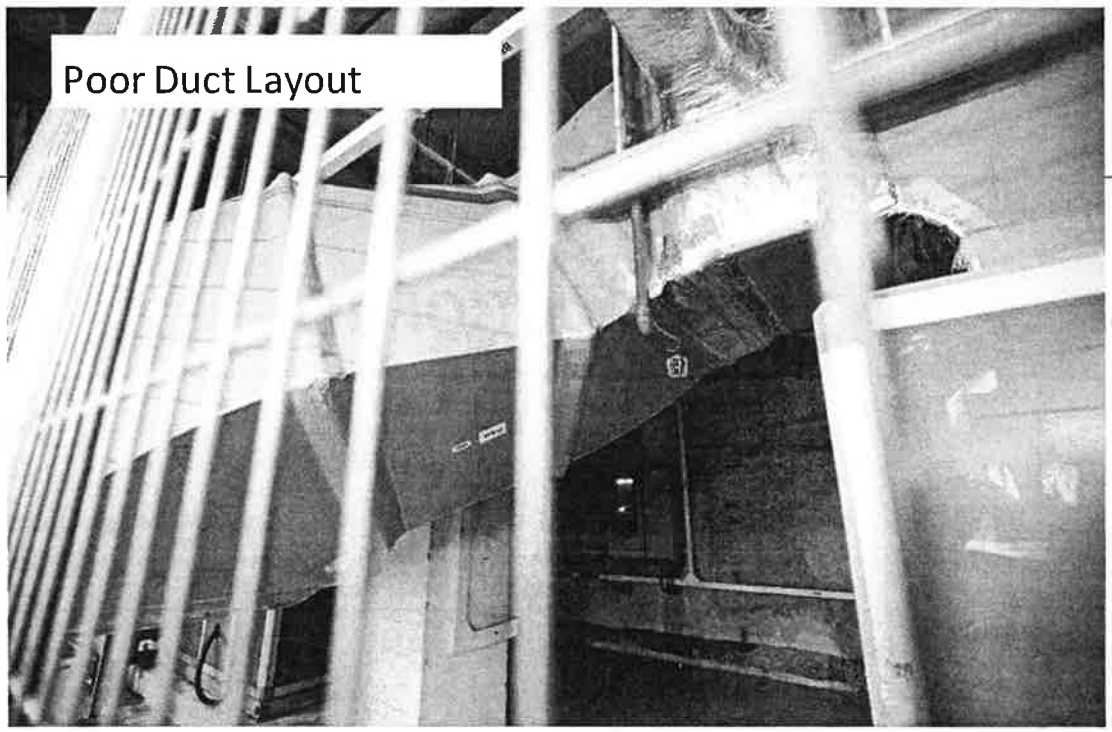
INTRODUCTION TO DUCT DESIGN

Introduction to Duct Design

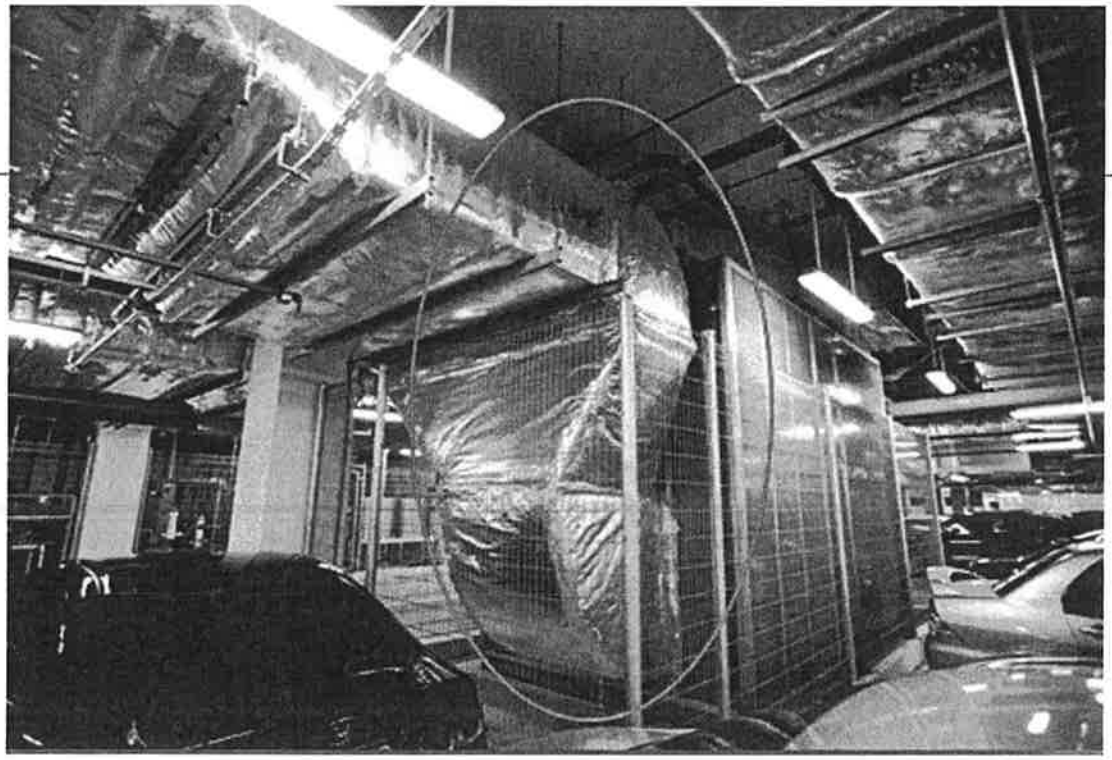
Duct Design is an art as much as it is a science

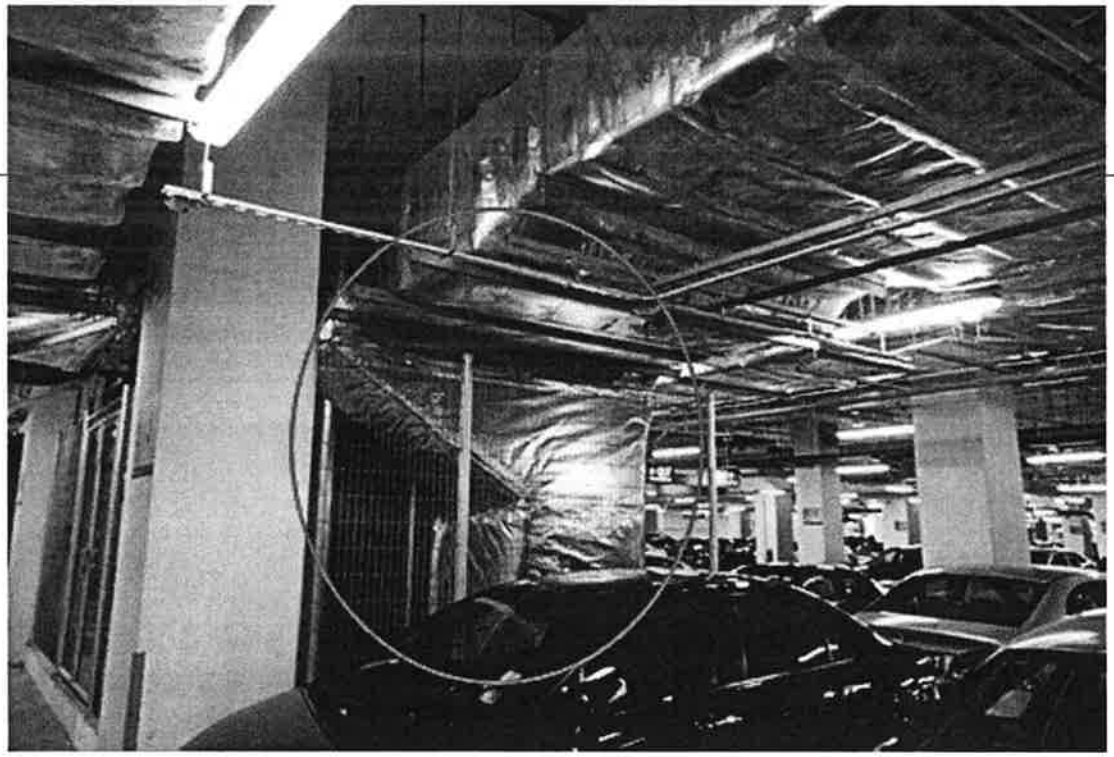


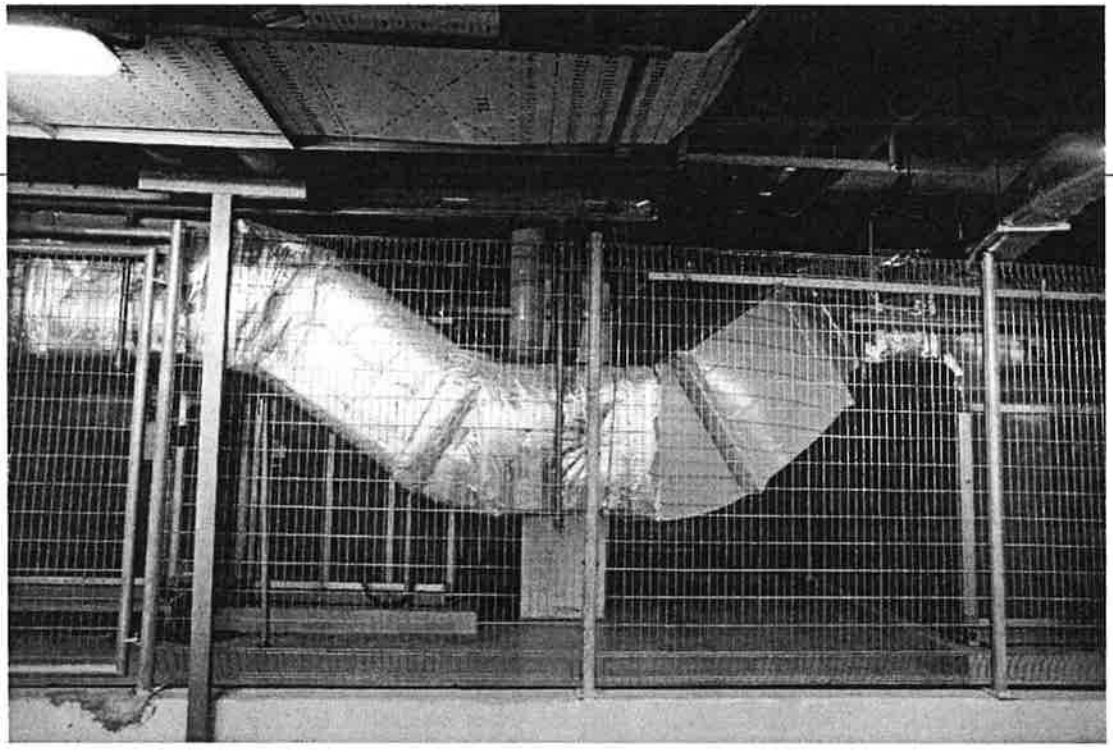
Poor Duct Layout

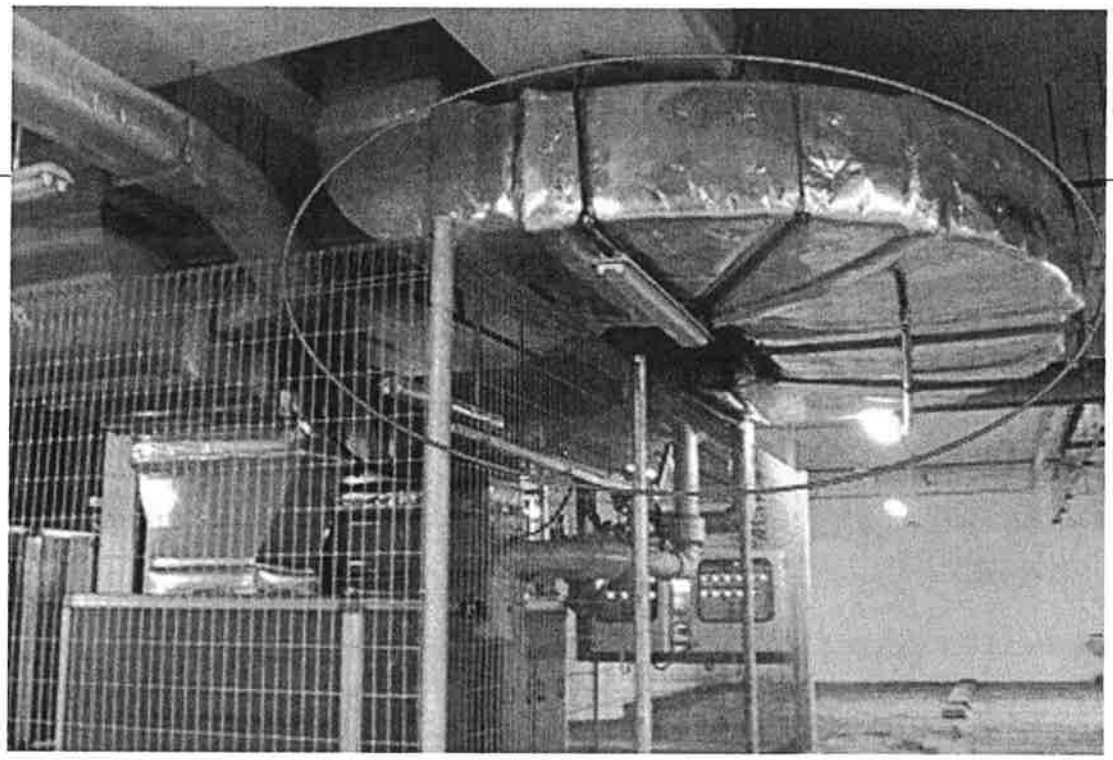


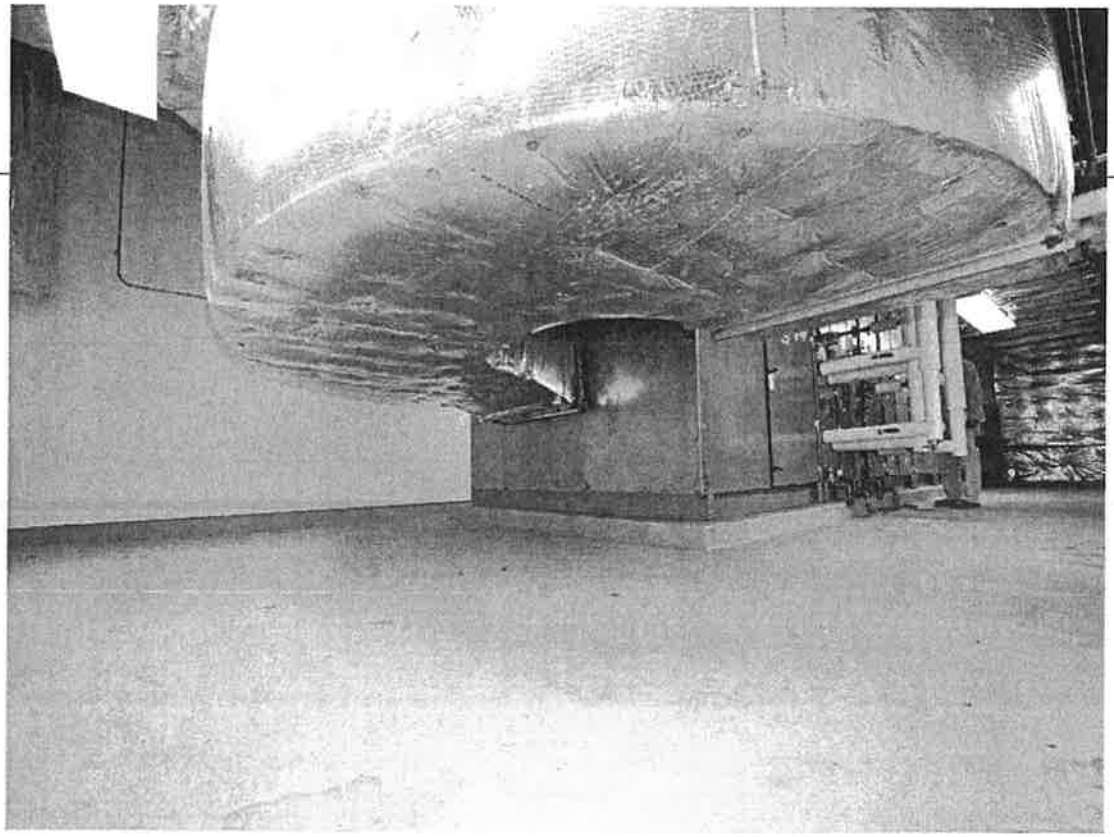


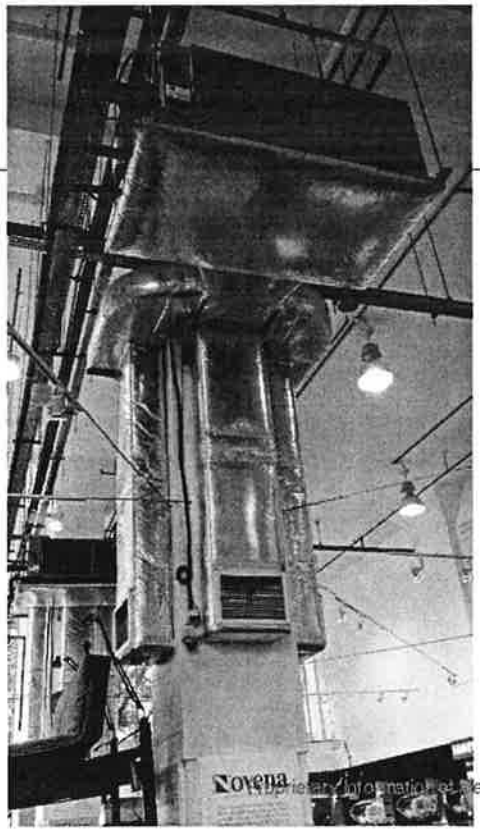


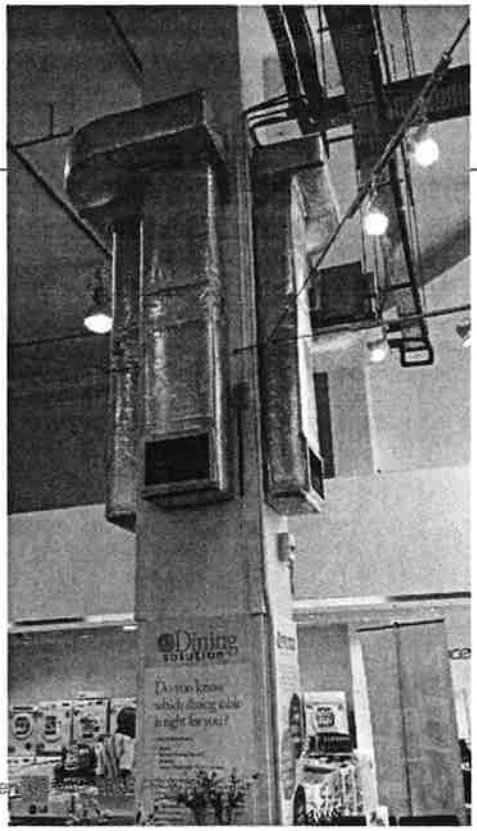








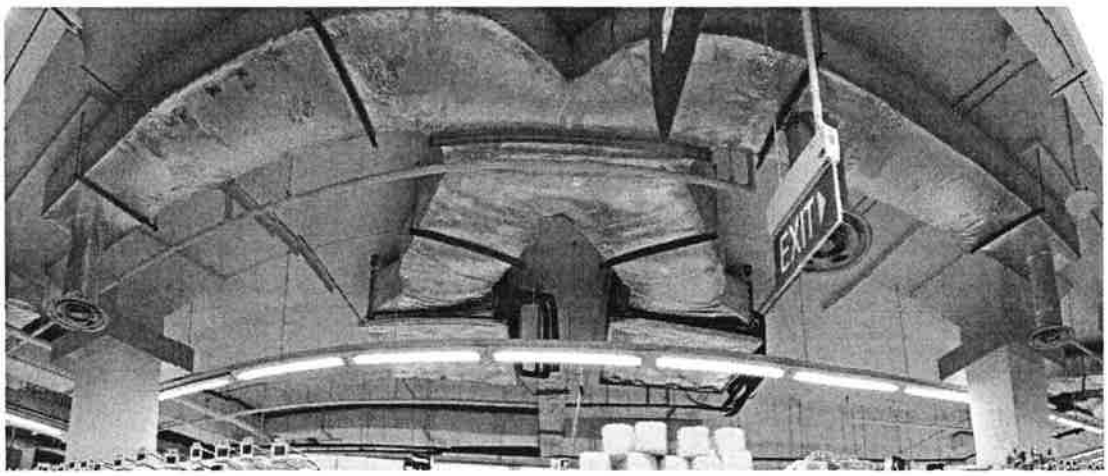




Measurement & Verification Pte Ltd

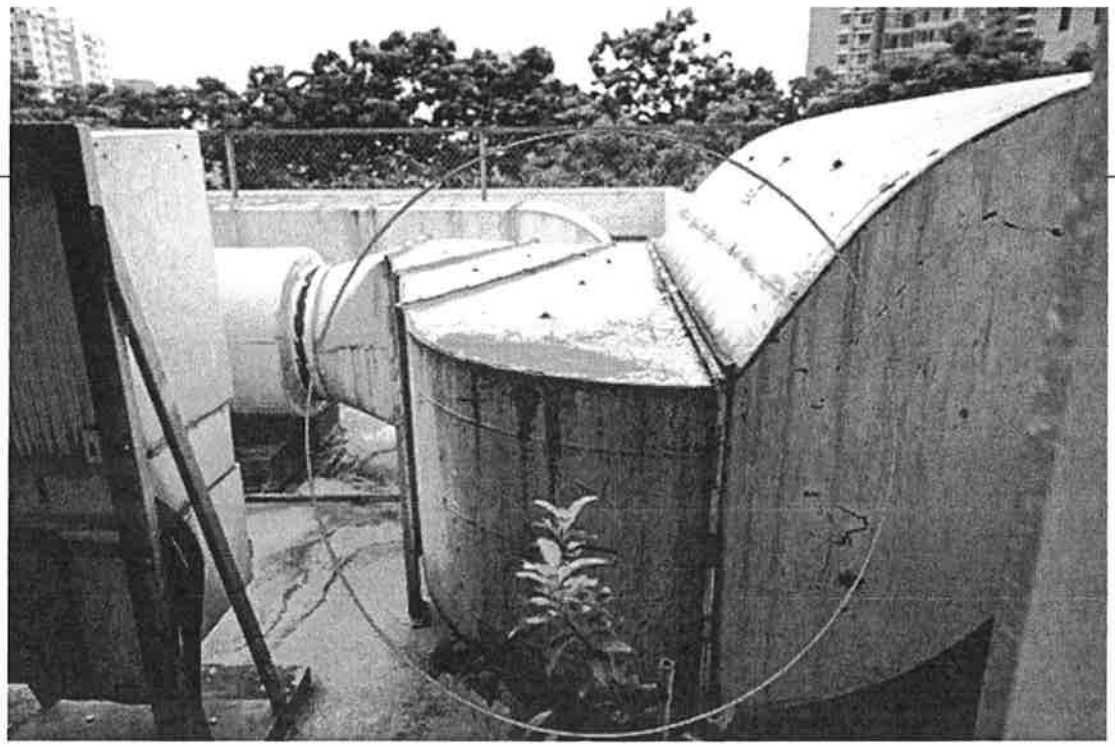
Proprietary Information of Measurement & Verification Pte Ltd

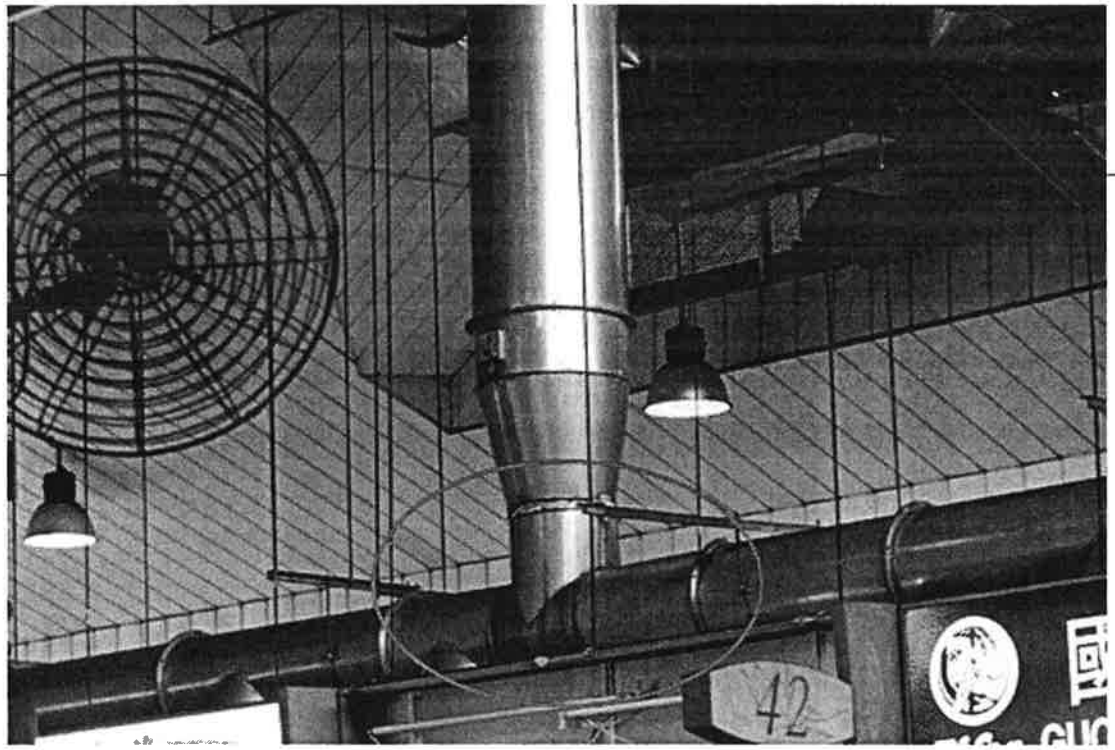
Measurement & Verification Pte Ltd



Proprietary Information of Measurement & Verification Pte Ltd







DUCT DESIGN CONSIDERATIONS

Duct Design Considerations

1. Go straight

The most important rule of all. The straighter the duct system, the lower both energy and first costs will be.



2. Reduce pressure losses

Avoid duct fittings that have high pressure losses and consequently high noise generation especially in high velocity systems.

Avoid consecutive fittings because they can dramatically increase pressure drop.

3. Plan your layout

2-1/2 duct diameters straight duct after fan connection to reduce fan system effect losses

Avoid duct locations where access for installation is restricted. This normally results in poor installation and consequently excess leakage from poorly installed joints, inadequate support etc

Recognize that in many cases a smooth elbow with internal insulation will give effective and cheap noise control.

Duct Design Considerations : Code requirements

SS553-2009				
Type	Load Condition	Min Eff	Units	Remarks
Constant volume system	Full load	1.7	KW / m^3/s	
		0.47	W / CMH	
Variable volume system	Full load	2.4	KW / m^3/s	
		0.67	W / CMH	
	Part load	Motor >= 11kw must able to modulate to 50% design flow at less than 30% of design KW		

Duct Design Considerations
Green Mark for New Building v4.1

1-2C Airside energy efficiency (Max 6pts)

Fan System Motor Nameplate Power OR **Fan System Input Power**

Baseline Air Distribution System Type	Allowable Motor Nameplate Power	
	($kW/m^3/s$)	(W/CMH)
AHUs/FCUs ≥ 4kW (Constant Volume)	1.7	0.47
AHUs ≥ 4kW (Variable Volume)	2.4	0.67
Fan systems with nameplate motor power < 4 kW	No baseline	

Baseline Air Distribution System Type	Allowable Fan System Input Power[a]	
	($kW/m^3/s$)	(W/CMH)
AHUs/FCUs ≥ 4kW (Constant Volume)	1.5	0.42
AHUs ≥ 4kW (Variable Volume)	2.1	0.58
Fan systems with nameplate motor power < 4 kW	0.6	0.17

Duct Design Considerations
Green Mark for Existing Building v3.0

(c) Air Distribution system:
- Air Handling Units (AHUs)
- Fan Coil Units (FCUs)

Baseline – Fan power limitation in air conditioning system

Allowable nameplate motor power	
Constant volume	Variable volume
0.47 W/CMH	0.74 W/CMH

DUCT PRESSURE DROP & DUCT SIZING

Terminology and Equations:

Static Pressure (SP) - The amount of energy (static) required to overcome the frictional compression of air in a duct system.

Velocity Pressure (VP) - The amount of kinetic energy (dynamic) required to move the airflow through a duct system.

Total Pressure (TP) - The total amount of energy required to overcome both the frictional losses (SP) and dynamic losses (VP) imposed by the resistance of airflow in a duct system.

Equations:

$$VP := \rho \cdot \left(\frac{\text{Velocity}}{1097}\right)^2 \quad \text{(in w.g.)}$$

ρ = The density of the air

$$\text{Velocity} := \frac{\text{Airflow}}{\text{Area}} \quad \begin{matrix}\text{(cfm)} \\ \text{(cross-sectional)}\end{matrix}$$

$$TP = SP + VP$$

Δ = Change

$$\Delta TP = \Delta SP + \Delta VP$$

Pressure drop along ductwork

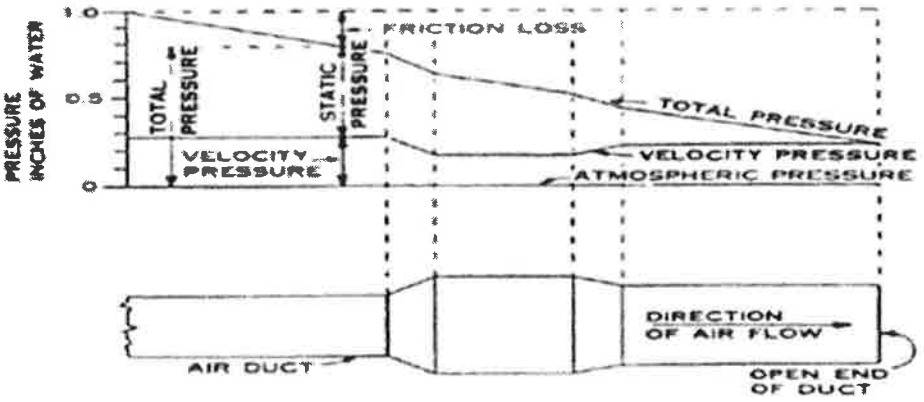
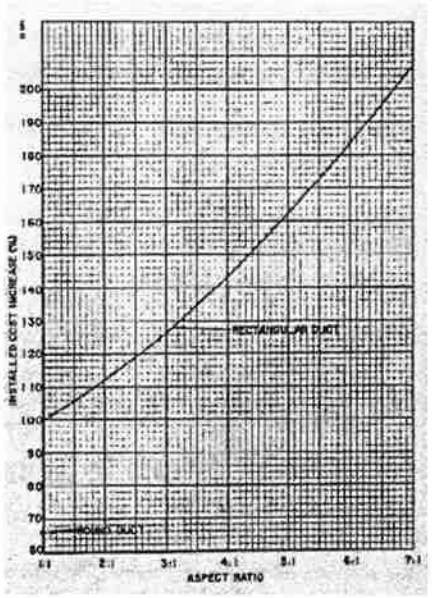


FIGURE 9-J

ACTUAL CHANGES IN PRESSURE WITH CHANGES IN DUCT AREA

Aspect Ratio

- Duct should have aspect ratio (W/H) as close to 1 as possible
- Duct with low aspect ratio have
- Lower friction rate (Lower operating cost)
- Lower perimeter to area ratio (Lower material cost)



Friction loss

Due to fluid viscosity, momentum exchanges between
boundary layer and the main flow, occur along the entire duct length

Pf = f * L

Pf: friction losses Pa

f: friction loss per meter Pa/m (from friction chart)

L: duct length m

Friction chart: based on standard air flowing through round galvanized ducts with an absolute roughness of 0.0003 ft.

For rectangular ducts: Circular equivalent diameter De

$De = 1.30(ab)^{0.625}/(a+b)^{0.250}$

a: length one side of duct

b; length adjacent side of duct

Fitting losses

Due to the change of air flow direction or area when flowing through equipment or fittings

Calculation methods:

1. Exact losses from catalogs
2. Equivalent length estimate
3. Fitting loss coefficients

   $Pc = C * \rho v^2/2g$

   Pc: fitting losses Pa

   C: fitting loss coefficients (from fitting database)

   $\rho v^2/2g$: Velocity pressure Pa

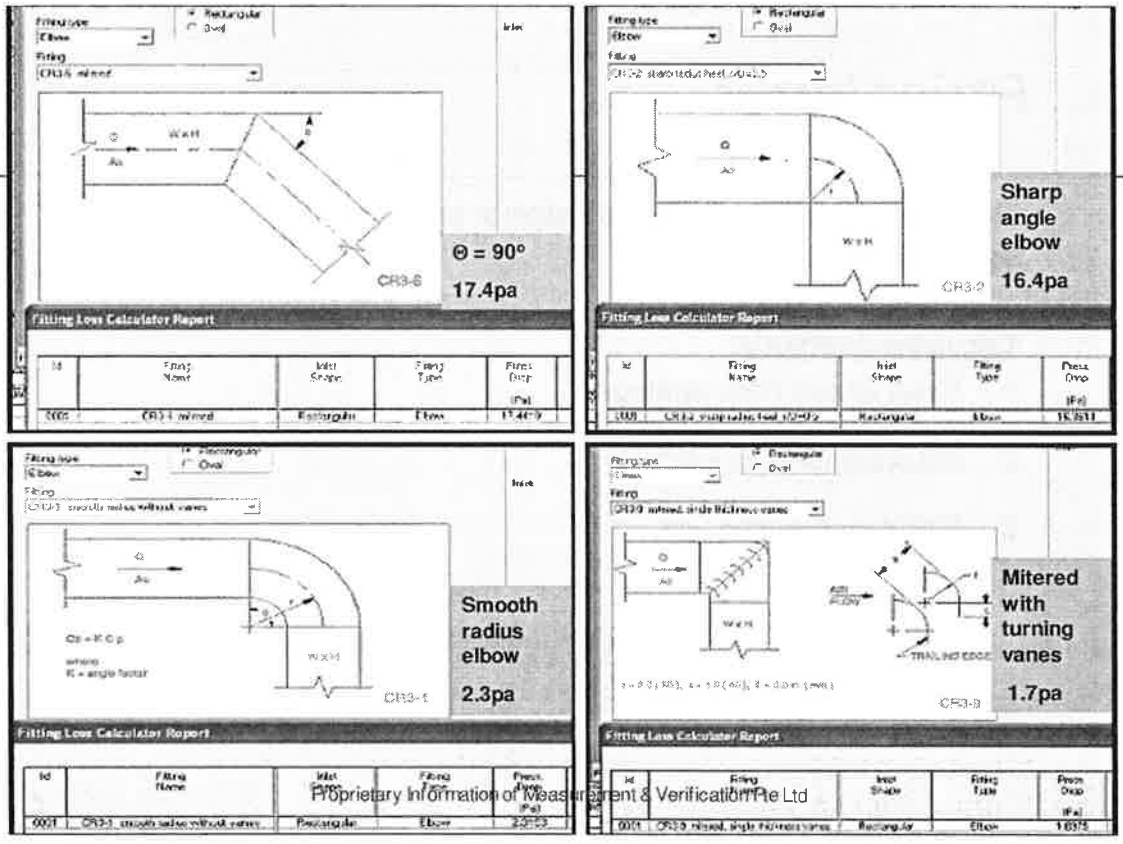
Difference for different elbows

Comparison of Loss Coefficients of Abrupt Turn Fittings.

FITTING	90° ELBOWS 8" Dia.	LOSS COEFFICIENT[1]
Die Stamped		0.10
5-Piece		0.22
Mitered with Turning Vanes		0.52
Mitered with Throat Gore		0.85
Mitered with Heel Gore		1.04
Mitered		1.24

[1] For Elbows: $\Delta p = K \times VP$

* Extracted from ASHRAE 2002 Duct Fitting Database



Duct Sizing Method

1. Velocity Method

Assign velocities to each duct based on recommended velocity tables and then choose the proper duct size, typically apply to small systems, such as homes and small offices

2. Equal Friction

Each segment of the duct work system is sized on using the same friction rate or loss per unit length for the entire system. This method is far superior to the velocity reduction since it require less balancing for symmetrical duct layout.

3. Static Regain

The static regain method is to size the duct network so that the decrease in velocity pressure of each branch just offset the friction loss in the succeeding section of the duct. The static pressure will then be in theory be the same at each terminal and at each branch throughout. This method is applicable for supply duct only and is most suitable for VAV system.

Velocity Method

1. LOW VELOCITY SYSTEM

NC of area being served	Main supply duct (m/s)	Branch supply duct (m/s)	Return air duct (m/s)
50	10.0	8.0	6.0
45	9.0	7.0	6.0
40	8.0	6.0	5.0
35	6.5	5.0	4.0
30	5.0	4.0	3.0

2. HIGH VELOCITY SYSTEM

Air quantity carried by duct l/s (CMH)	Max. Velocity m/s
Above 5000 (18,000)	20.0
5000 to 3000 (18,000 to 10,800)	17.5
3000 to 1500 (10,800 to 5,400)	15.0
1500 to 500 (5,400 to 1,800)	12.5

Equal Friction

Attempts to keep the pressure drop per unit length (generally 100 ft.) of duct relatively constant throughout the system.

- Calculation revolves around the design friction rate.
- Typical friction rates range from 0.05 in. w.g. per 100 ft. to 0.15 in w.g. per 100 ft. (0.4pa/m to 1.2pa/m)

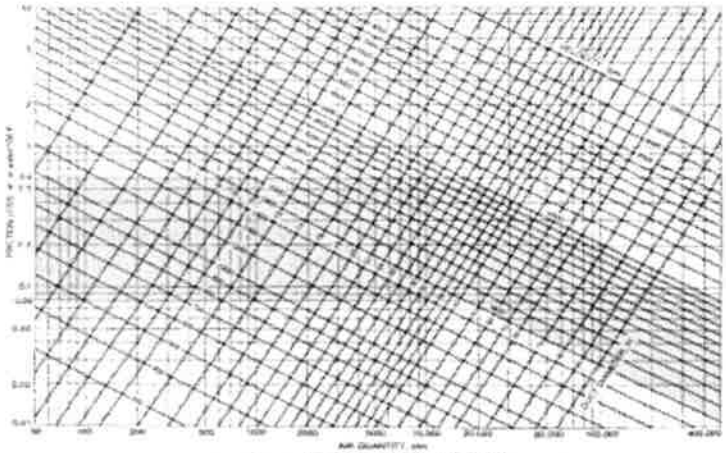
Design Friction Rate = 0.047

Determined by first trunk section... unless overridden

Section	Length	Actual Friction Rate	Airflow	Velocity	Diameter	Actual SP Loss	Height	Width
0001	20	0.047	3740	991		0.0093	24	24
001A	1	0.034	3740	875	28	0.0003		
0005	27	0.031	2929	794	26	0.0085		
0009	27	0.036	2035	771	22	0.0098		
0013	37	0.029	1059	672	18	0.0108		
0015	45	0.030	550	597	14	0.0137		
0023	1	0.472	550	1576	8	0.0047		

Equal Friction

The shade area of the friction chart is the suggested ranged of friction rate and air velocity. Friction rate is designed by balancing between energy cost and duct cost.



Static Regain

- Complicated design requiring computer software design programes, used for large, complex, high velocity systems
- Static regain factor : 75%(considering the splitting of mass at junctions and the dynamic loss between junctions, ranging from 0.5-0.95)
- Calculation equation:

$$\Delta P = R(\frac{\rho V_1^2}{2g} - \frac{\rho V_2^2}{2g})$$

- R : Static regain factor
- ΔP: static pressure regain between junctions

Static Regain

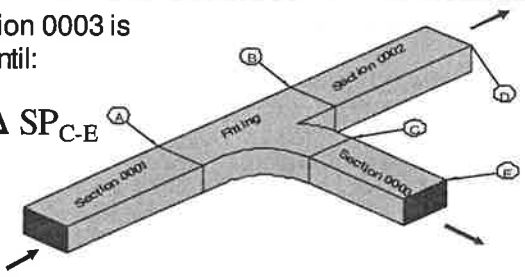
- Attempts to maintain a constant static pressure throughout the duct system (a balanced system).
- Calculation is limited by parameters such as minimum velocity, fan size, and sometimes fitting selections (rectangular).
- Fan outlet size is critical
- Typical minimum velocity = 500 fpm....when decreased the method can actually be limited in it's effectiveness

Size of section 0002 is increased (converting velocity pressure to static pressure) until:

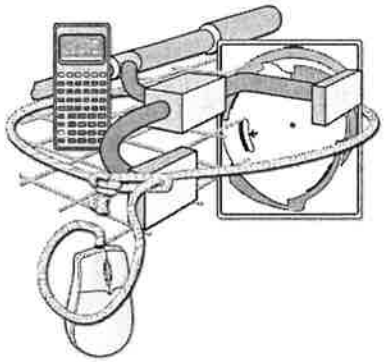
$$\Delta SP_{A-B} = \Delta SP_{B-D}$$

Size of section 0003 is increased until:

$$\Delta SP_{A-C} = \Delta SP_{C-E}$$



Softwares are Available for Duct Design



VariTrane™ Duct Designer

- Models supply duct systems from fan to diffuser or VAV box
- Uses the ASHRAE fitting database
- Models round, rectangular, and flat-oval duct

Noise Control Considerations

- Fan Selection: to reduce the sound power at the fan source.
- Pressure drop: lowering the system pressure drop allows for lower fan speed which lowers sound power levels.
- Duct Velocities: Keep to limits.
- Avoid take-offs near fan: Take-offs near to the fan normally needs dampers that are near full throttle position resulting in noise.
- Duct insulation and attenuator: attenuate sound generated by the noise sources during transportation. A smooth elbow with internal insulation will often give effective and cheap noise control

THANK YOU



Measurement & Verification Pte Ltd

CHILLER PLANT MEASUREMENT & VERIFICATION

Steven Kang
Steven.kang@mnv.com.sg

Agenda

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1. Chiller Plant Measurement & Verification
2. Temperature Measurement
3. Flow Measurement
4. Power Measurement
5. Total Uncertainty and Heat Balance

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CHILLER PLANT MEASUREMENT & VERIFICATION

Green Mark requirements

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- GM V4.1 for New Buildings and GM V3.0 for Existing Buildings.
- A pre-requisite for permanent instrumentation to measure chiller plant efficiency to accuracy of +/- 5% plus additional requirements below:

1) Location and Installation of instruments to meet manufacturer's recommendations.
2) Data acquisition system to have minimum A/D resolution of 16bit.
3) All data logging with capability to trend at 1 minute time interval.
4) Flow meters are to be provided for chilled water and condenser water loop and shall be of ultrasonic / full bore magnetic type or equivalent.
5) Temperature sensors are to be provided for chilled water and condenser water loop and shall have an end-to-end measurement uncertainty not exceeding +/- 0.05C over the entire measurement or calibration range. All thermo-wells shall be installed in a manner that ensures that the sensors can be in direct contact with the fluid flow. Provision shall be made for each temperature measurement location to have 2 spare thermo well located at both side of the temperature sensor for verification of the measurement accuracy.
6) Dedicated power meters are to be provided for each of the following equipment: Chillers, chilled water pumps, condenser water pumps and cooling towers.

Measuring chiller plant efficiency

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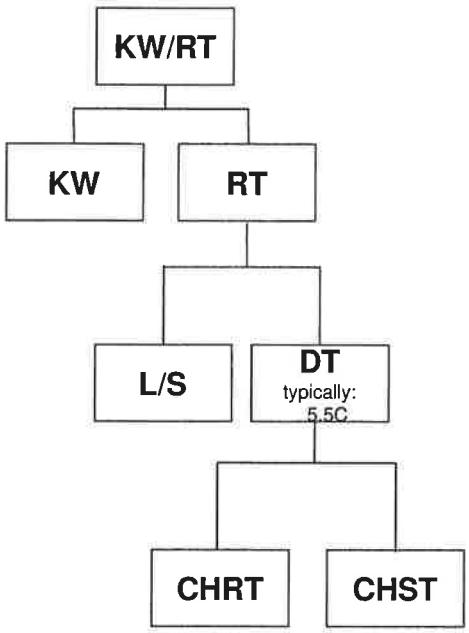

Instrument Accuracy

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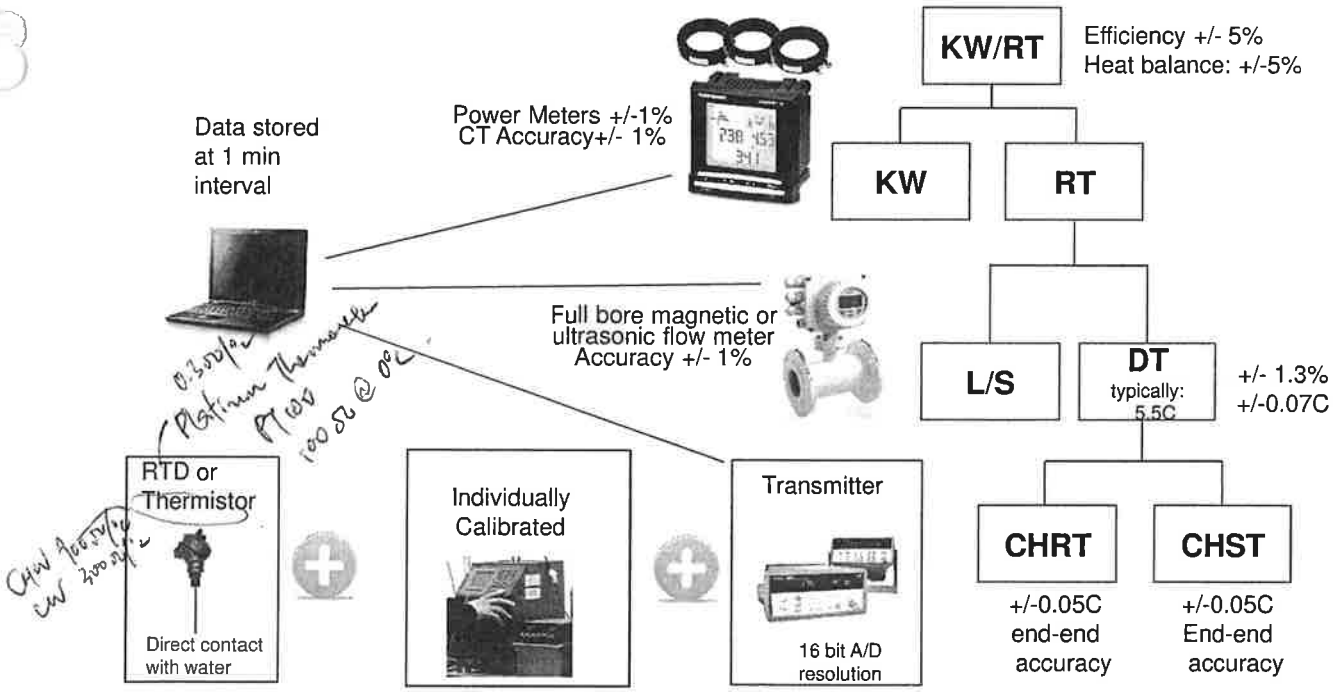

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TEMPERATURE MEASUREMENT

Temperature Measurement

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	Resistant temperature detectors (RTD)	Thermistors
Principle of operation	It exploits the predictable change in electrical resistance of some materials with changing temperature. Most commonly used material is Platinum and they are commonly called PRT or PT.	Like RTD, it also exploits the predictable change in electrical resistance with temperature. Thermistors are made from certain metal oxides whose resistance decreases with increasing temperature.
Accuracy	They have positive temperature coefficient. It is very accurate and nearly linear. Characterised by Callendar-van Dusen equation $R = r\ (\Omega\ @\ 0C) \times (1 + aT + bT^2)$ Resistance of PT (100 Ω at 0C) is relatively low so lead wire resistance becomes significant. 3 or 4 wire PT is therefore required. Exhibit a small change in resistance for a change in temperature. (eg 0.3 Ω / C) With proper calibration, uncertainties of better than 0.01°C can be achieved.	They have negative temperature coefficient. It is very accurate but non linear. Characterised by Steinhardt equation $1/T = a + b(LnR) + c(LnR)^3$ Resistance is high (eg 10,000 Ω at 25C) so lead wire becomes insignificant. 2 wire thermistor is usually adequate. Exhibit a large change in resistance for a change in temperature. (eg 300 Ω / C for CW and 900 Ω / C for CHW) With proper calibration, uncertainties of better than 0.01°C can be achieved.
Cost	High	Medium

Temperature Measurement Chain

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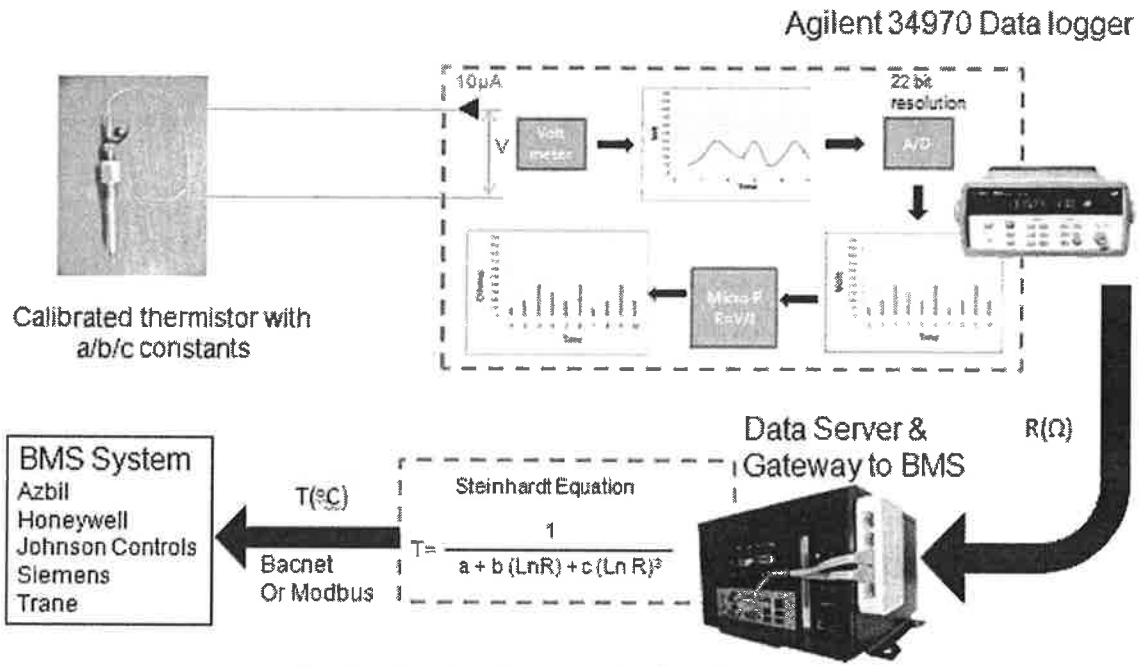

Thermistors Direct Contact with Fluid

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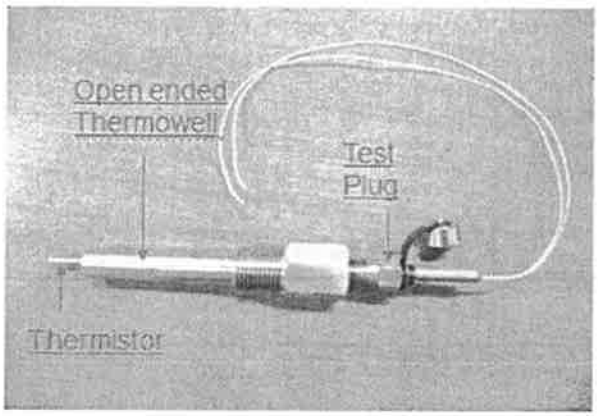
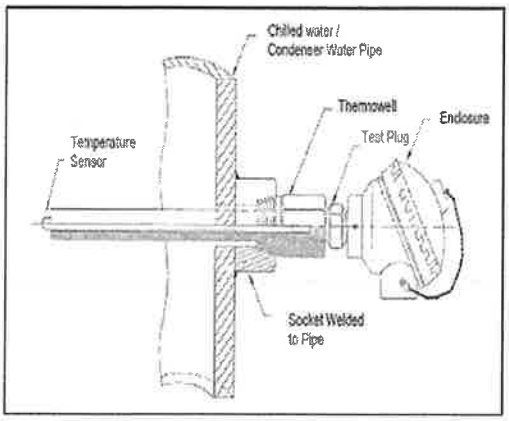
- To improve accuracy and response time.





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FLOW MEASUREMENT

Flow Measurements

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	Full bore magnetic flow meter	Ultrasonic clamp on flow meter
Principle	The principle of operation is based on the Faraday Law which states that a voltage is created as conductor moves through a magnetic field. The fluid is the conductor in this case. The voltage created is proportional to the velocity of the fluid.	The principle of operation is based on sound transit time in a moving fluid. The sound transit time in the direction of flow is faster than the sound transit time in the opposite to the direction of flow.
Accuracy	0.5-1%	1-2%
Price	Depends on the diameter Diameter 200mm (8"): S$3-4k Diameter 700mm (28"): S$10-12k	S$ 5-6k
Brands	ABB Endress & Hauser Siemens	GE Panametrics Endress & Hauser Siemens

Full Bore Electromagnetic Flow Meter

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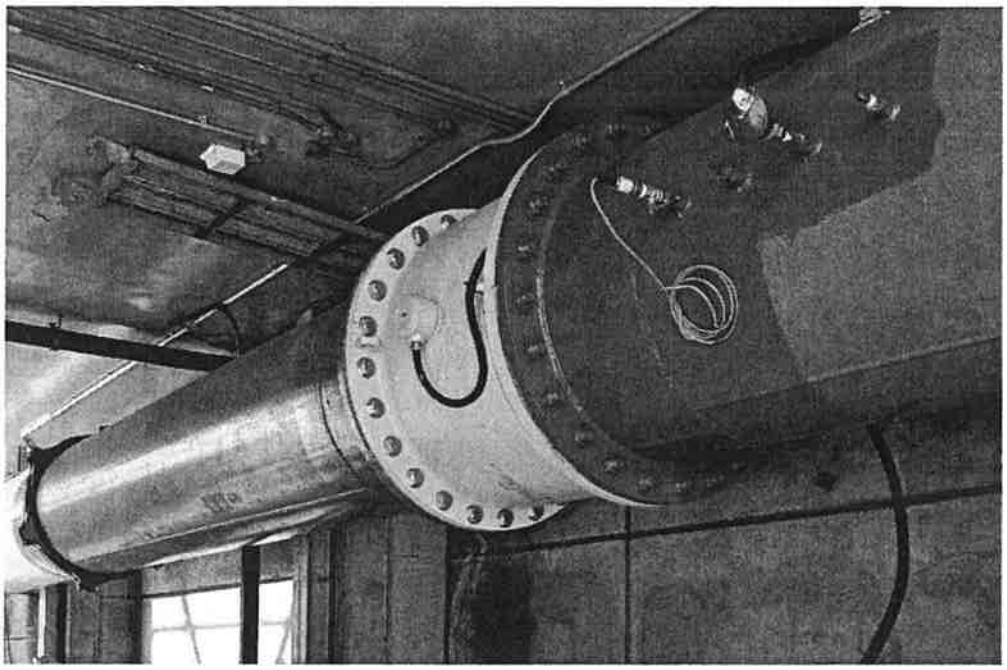
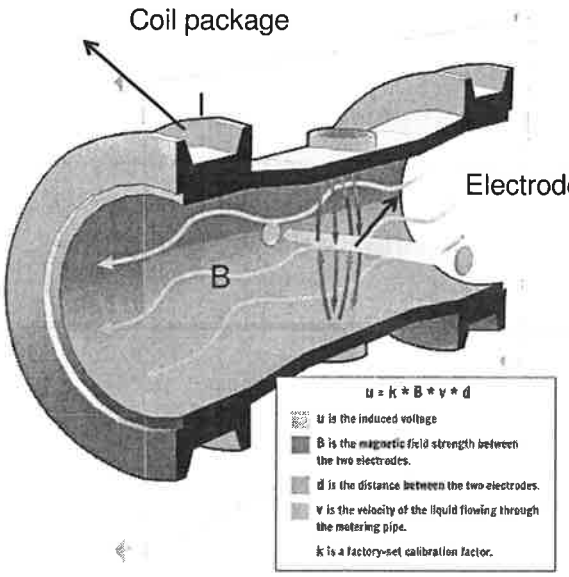


Full Bore Electromagnetic Flow Meter

Measurement & Verification Pte Ltd



The principle of operation for the magnetic flowmeter is based on the Faraday's Law of Electromagnetic Induction.

An electrical current (I) is applied to a coil package inside the flowmeter. As a result, a magnetic field (B) is created across the metering pipe.

When a conductive liquid flows through the magnetic field, a small voltage (V) is induced. This voltage is proportional to the velocity of the flow and is accurately measured by two stainless steel electrodes mounted opposite each other inside the metering pipe. The two electrodes are connected to an advanced electronic input circuitry which processes the signal and in turn feeds it to the a microprocessor inside the electronics module. The microprocessor then calculates the volumetric flow and controls the various outputs on the terminal board.

Ultrasonic Clamp-on Flow Meter

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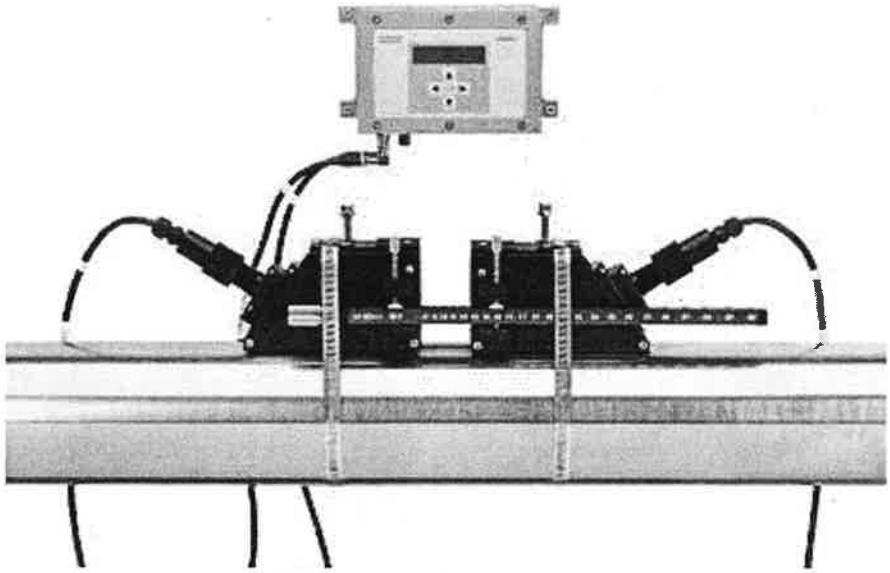

Ultrasonic Clamp-on Flow Meter

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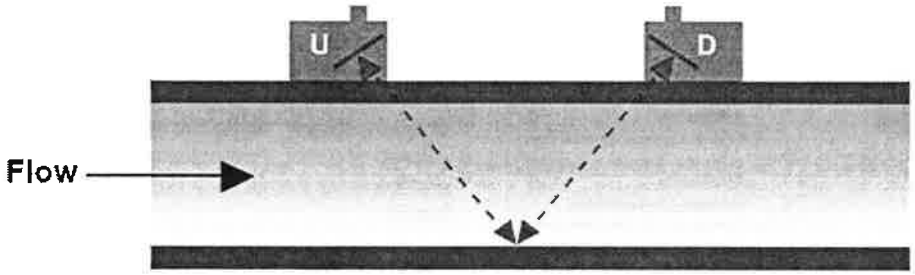
- The measuring principle of ultrasonic flow meter is based on ultrasound transmission through a liquid. Transit time measuring technique utilizes two transducers, which function as both ultrasonic transmitters and receivers. The ultrasound waves traveling in the direction of the flow of the liquid require less time than when traveling in the opposite direction, therefore the difference in time taken by the ultrasonic waves to travel over the same distance but in opposite directions is an indication of the velocity of the liquid.



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POWER MEASUREMENT

Power Measurement

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Class 1



Class 1

Power Meter

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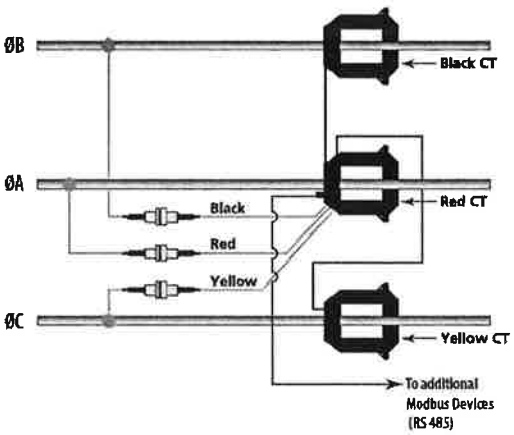
- Use integrated meter (Power meter and CT by same supplier) to measure Volt, Amps & PF
- True RMS power





Integrated Power Meter Specifications

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SPECIFICATIONS

Input Primary Voltage	208 to 480VAC RMS[††]
Number of Phases Monitored	One to Three
Frequency	50/60Hz
Primary Current	Up to 2400 amps cont. per phase[††]
Internal Isolation	2000VAC RMS
Insulation Class	600VAC RMS[†††]
Temperature Range	0° to 60°C (32°F to 140°F), 50°C (122°F) for 2400A
Humidity Range	0 - 95% non-condensing
Systems Accuracy	±1% of reading from 10% to 100% of the rated current of the CTs...accomplished by matching the CTs with a meter and calibrating them as a system
Output Physical Characteristics	RS-485, 2 wire + shield
Baud Rate	9600, 8N1 format
Protocol	Modbus RTU**(*)

Data output:
kWh, Consumption
kW, Real Power
kVAR, Reactive power
kVA, Apparent power
Power factor
Average Real power
Minimum Real power
Maximum Real power
Voltage, line to line
Voltage, line to neutral[†]
Amps, Average current
kW, Real power ØA[†]
kW, Real power ØB[†]
kW, Real power ØC[†]

[†] *Based on derived neutral voltage.*

Errors in Power Measurement

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- Power meters that are not integrated with CT may be prone to error due to wrong settings.
- For example, an energy audit revealed that power measurement error existed in BAS for 15 years because of a wrong CT ratio setting.



Class 1

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MEASUREMENT UNCERTAINTY AND HEAT BALANCE

Measurement uncertainty

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- Root Sum Square Method

Power meter uncertainty	+/- %	1.0%
Current transformer uncertainty	+/- %	1.0%
Total power measurement uncertainty	+/- %	1.4%

Flow meter uncertainty	+/- %	1.0%
Site installation error allowance	+/- %	1.0%
Total flow measuremnt uncertainity	+/- %	2.0%

Temperature sensor end-to-end uncertainty	+/- C	0.05
DT temperature measurement uncertainty	+/- C	0.071
CHW design DT	C	5.5
CHW DT measurement uncertainity	+/- %	1.3%

Total KW/RT measurement uncertainity	+/- %	2.8%

Heat Balance Validation Check

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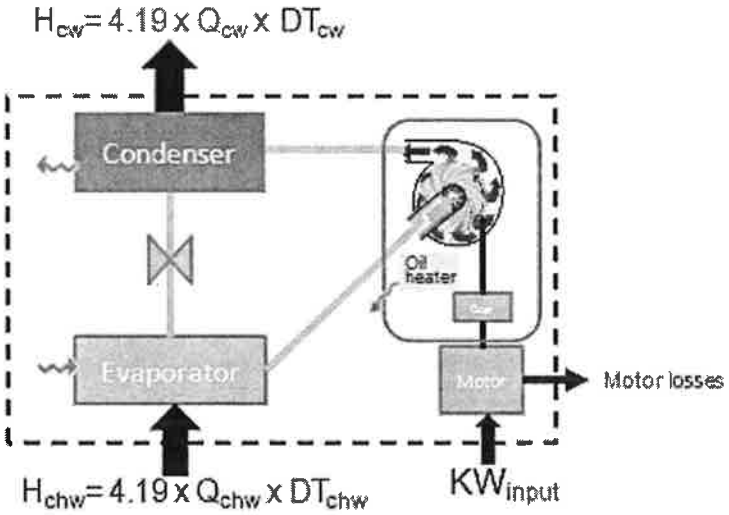

Energy in = Energy out

$(4.19 \times Q_{chw} \times DT_{chw}) + (KW_{input} - \text{Motor losses}) = 4.19 \times Q_{cw} \times DT_{cw}$

Heat Balance Validation Check

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$$\frac{\text{Heat input} - \text{Heat output}}{\text{Heat output}} <= 5\%$$

80% of data point must fall with +/-5% for 1 week operating period .

For more information

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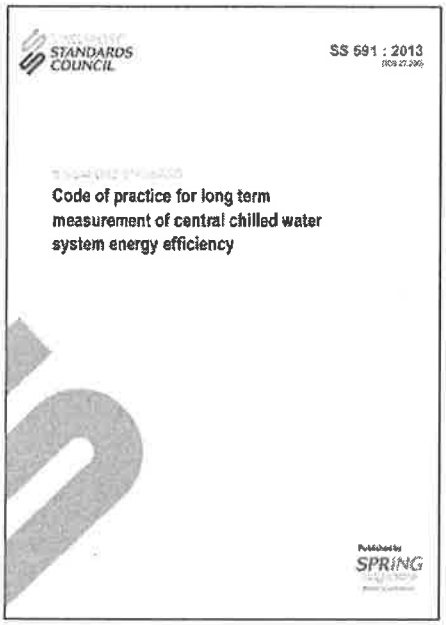

Chiller Plant Measurement & Verification Course (3.5 days)

THANK YOU

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Chiller, Pumps and Cooling Tower Optimization

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AGENDA

1. Overview of chiller plant optimization
2. Chiller Optimization
3. CHWP Optimization
4. CWP Optimization
5. Cooling Tower Optimization

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OVERVIEW OF CHILLER PLANT OPTIMIZATION

Chiller Plant Efficiency

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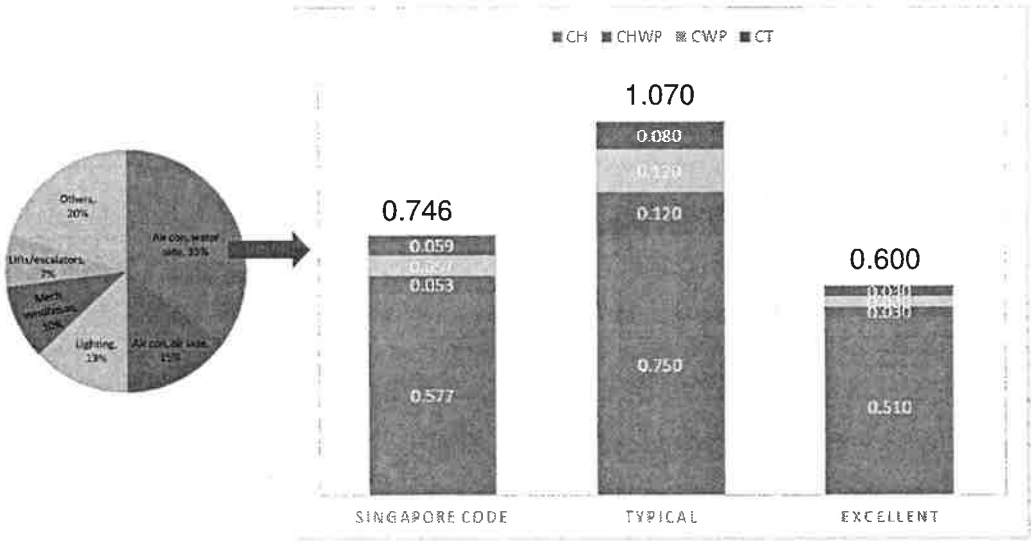
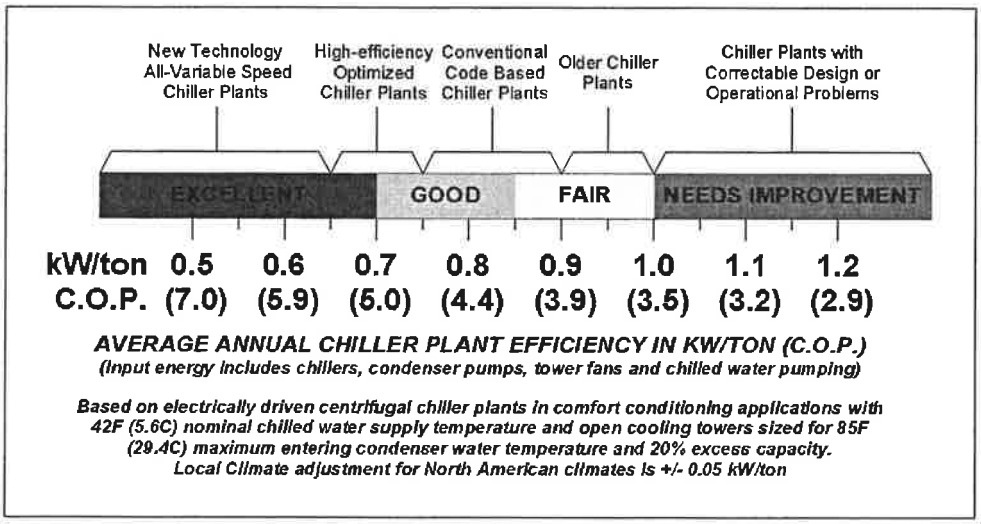


Chiller Plant Efficiency Benchmark

By Thomas Hartman Ashrae Journal



AVERAGE ANNUAL CHILLER PLANT EFFICIENCY IN KW/TON (C.O.P.)

(Input energy includes chillers, condenser pumps, tower fans and chilled water pumping)

Based on electrically driven centrifugal chiller plants in comfort conditioning applications with 42F (5.6C) nominal chilled water supply temperature and open cooling towers sized for 85F (29.4C) maximum entering condenser water temperature and 20% excess capacity. Local Climate adjustment for North American climates is +/- 0.05 kW/ton

CHILLER OPTIMIZATION

Chiller plant configuration

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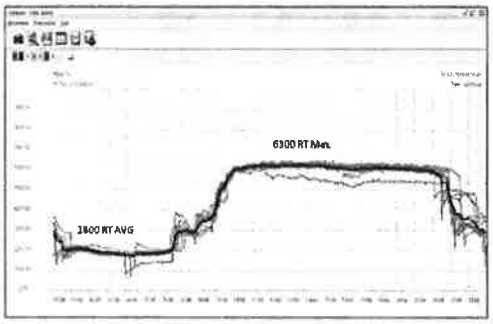
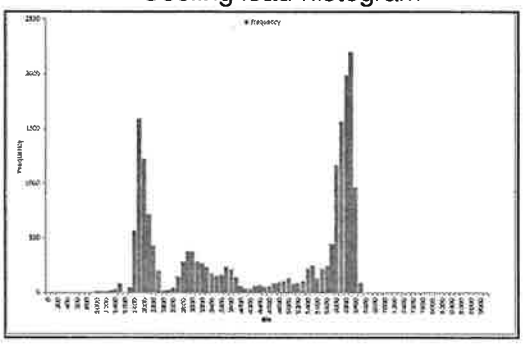
Get the chiller plant configuration optimized by plotting
- Minimum 1 week cooling load profile
- Cooling load histogram

1 week cooling load profile



Cooling load histogram



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Chiller plant configuration

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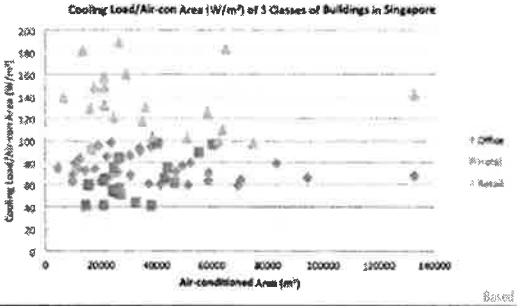
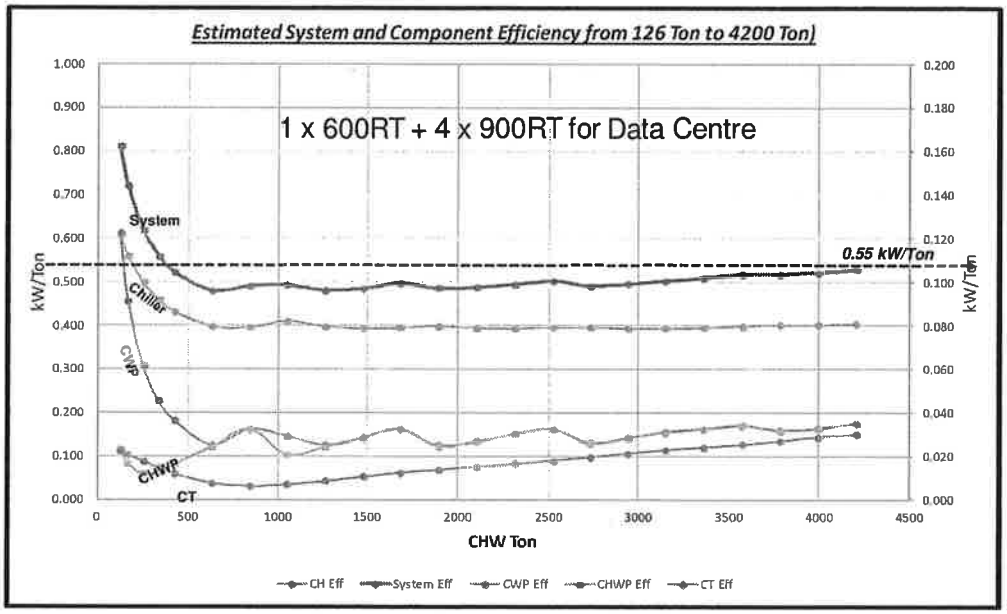
- Cooling load calculations for new buildings are for worst case scenarios.
- Bench mark design cooling load calculations with similar type of buildings to provide "sanity" check whether there are potential oversizing.
- While chiller plant may be <u>designed for maximum anticipated load</u>, it must also be <u>design to be efficient at minimum anticipated cooling load too</u>.

	Measured Cooling Load/Air-con Area (W/m²)	
	Range	Average
Office (35 projects)	60 – 98	76
Hotel (24 projects)	47 – 95	64
Retail (21 projects)	93 – 190	135



Based on data from BCA and NEA

Chiller plant configuration



Chiller plant configuration

Data Centre example

1 x 600RT + 4 x 900RT configuration

Energy efficient over wide range

Load (RT)	Operation stage	Chiller loading range range		Chiller efficiency			
				600RT KW/RT range		900RT KW/RT range	
4200	1 x 600 + 4 x 900	100%	86%	0.412	0.418	0.400	0.395
3600	4 x 900	100%	75%	-	-	0.400	0.397
2700	3 x 900	100%	67%	-	-	0.400	0.403
1800	2 x 900	100%	50%	-	-	0.400	0.427
900	1 x 900	100%	67%	-	-	0.400	0.403
600	1 x 600	100%	50%	0.412	0.450	-	-

Chiller plant configuration

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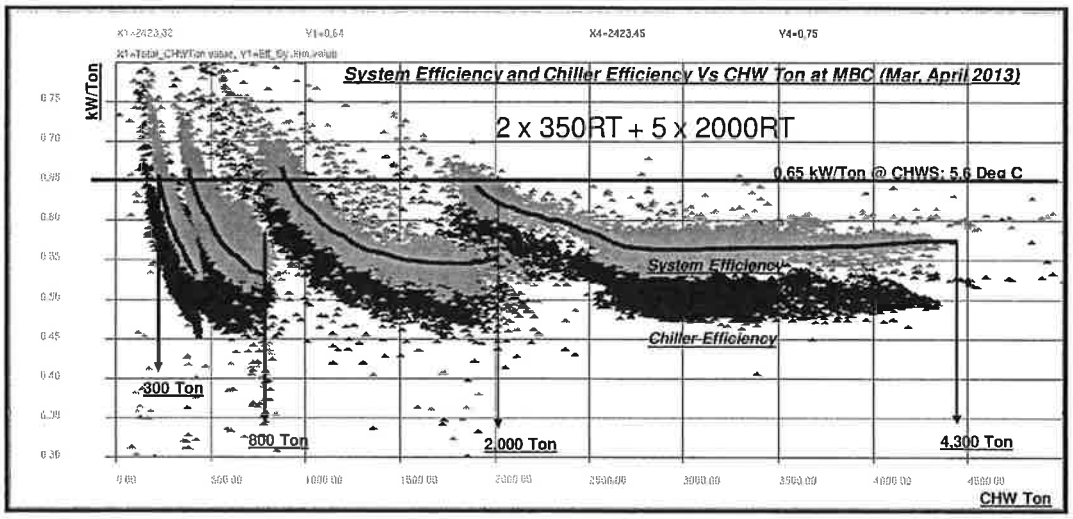
Accurate sensing and Optimization Control can optimize the system efficiency not only at 1 point but almost 90% of the load range.



W/C Chiller part load performance

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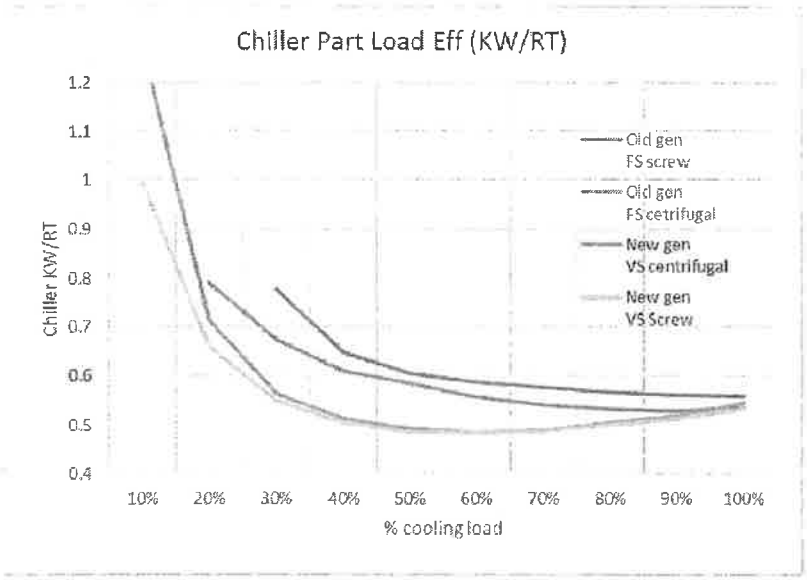

<u>Old gen fixed speed chillers (R134a)</u>
Peak eff: ~ 0.530 to 0.54 KW/RT
Peak eff point: 80-90% load

<u>New gen variable speed chillers (R134a)</u>
Peak eff: ~ 0.49 to 0.50 KW/RT
Peak eff point: 50-60% load

Performance rated at
CHW: 6.70C at fixed 2.4gpm/RT
CW: 29.4C at fixed 3.0 gpm/RT

Chiller performance are indicative and do not represent any particular brand.

Microsoft Excel Worksheet

Chilled water supply temperature

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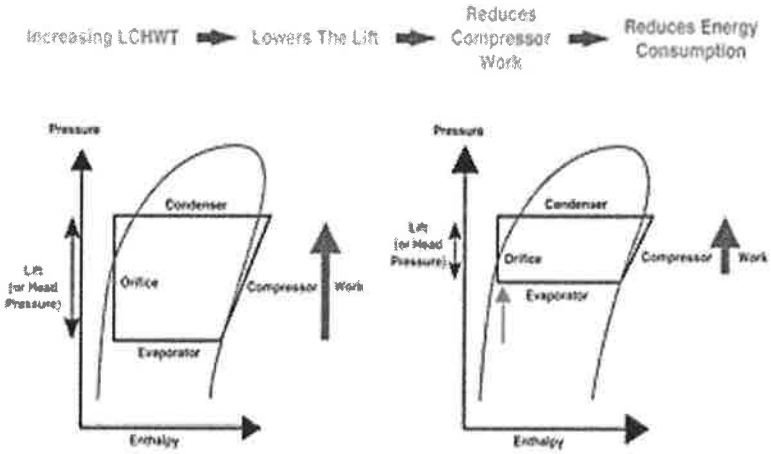

Chilled water supply temperature

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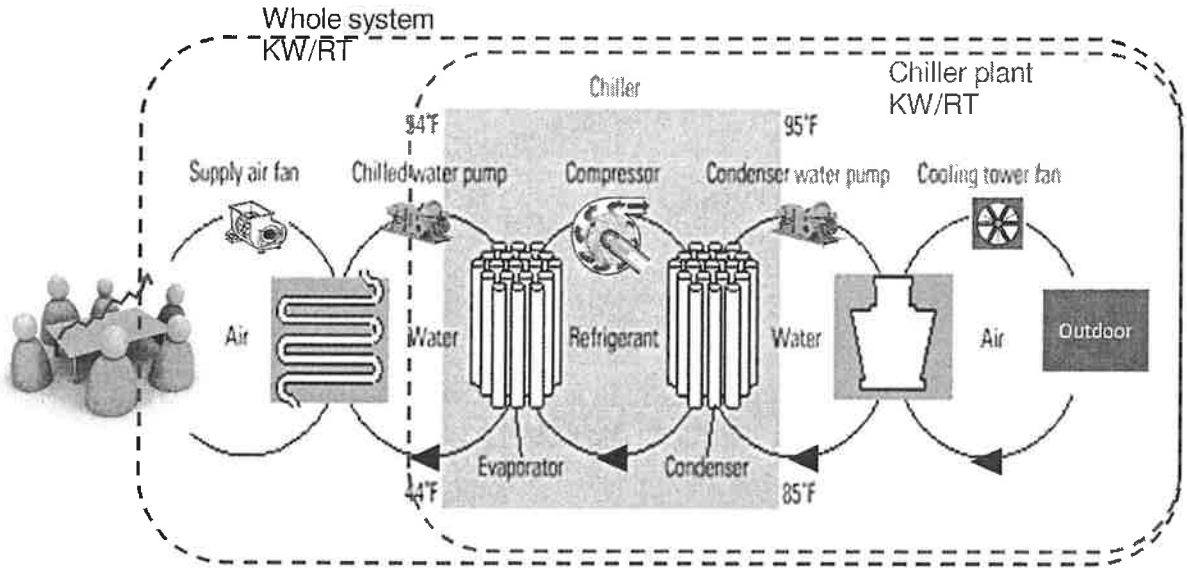


Consider whole system energy efficiency and cost

CHW temp (supply / return)	Option #1 6.7 / 12.2C		Option #2 8.0 / 15.0C	
Description	Eff (KW/RT)	Cost (S$)	Eff (KW/RT)	Cost (S$)
1 x 600RT Chiller	0.530	305,000	0.516 ↓	305,000 —
1 x CHW pump	0.042	9,200	0.023 ↓	8,500 ↓
20 x 30RT AHUs	0.227	200,000	0.236 ↑	220,000 ↑
Total	**0.799**	**514,200**	**0.775** ↓	**533,500** ↑
Annual energy cost	301,616		292,747	
Simple payback	Base		**2.2**	

Based on 55 hrs/ week and $0.22/kwh

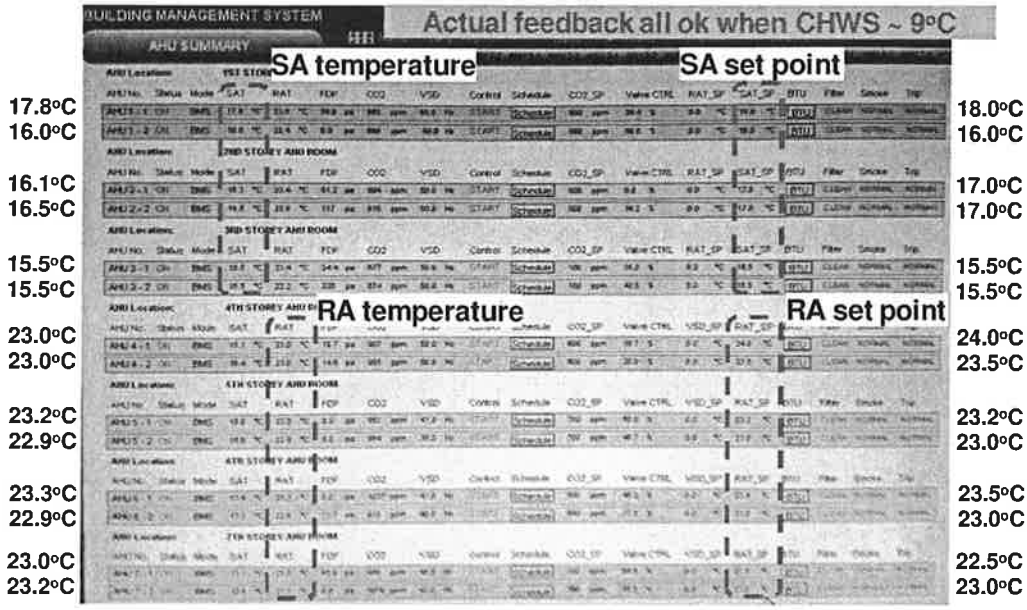
15

CHWST 1°C → ↑2~3% in chiller efficiency

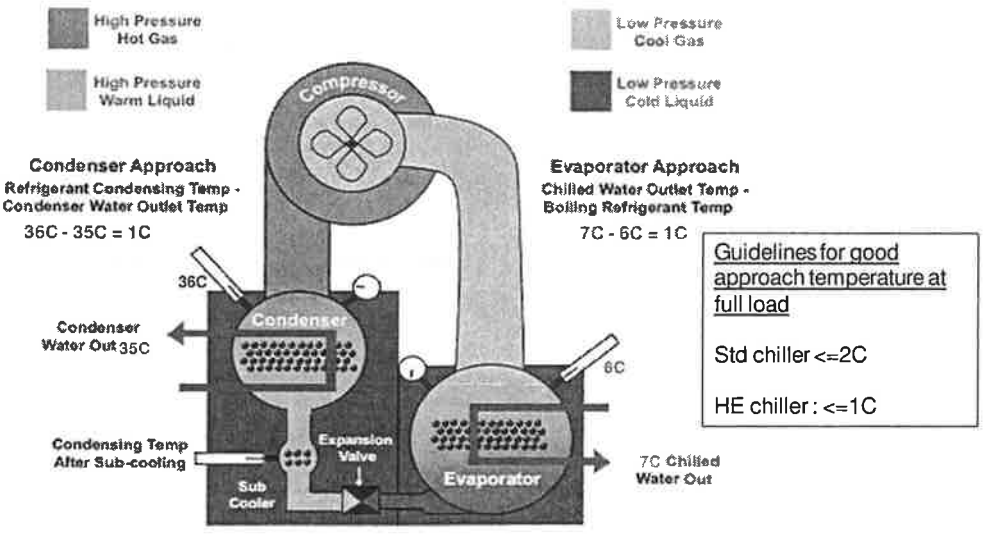
Chilled water supply temperature

- Potential to reset CHW temperature upwards when the cooling coils are oversized.
- Especially so at off-peak periods.
- Consider whole system efficiency. Eg Higher CHW temp will save chiller energy but may increase CHWP and VAV AHU fan energy.
- Consider humidity when raising CHW temperature.

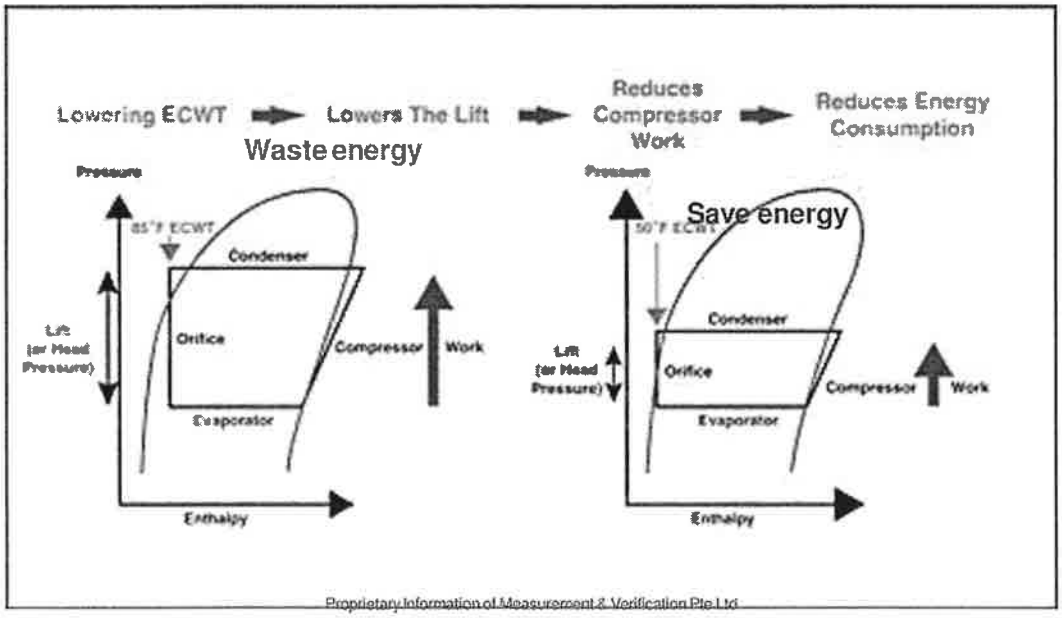
Chilled water supply temperature



Hx Efficiency and Approach Temperature



Compressor Lift (Condenser Side)



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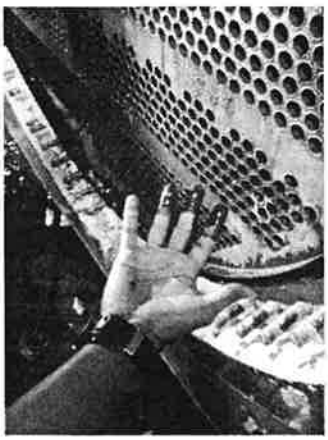
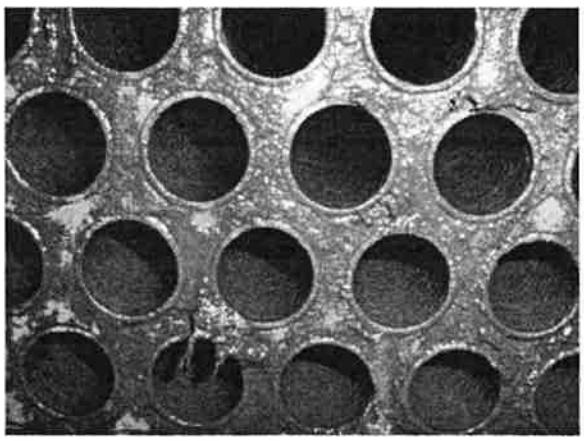
Dirty Evaporator





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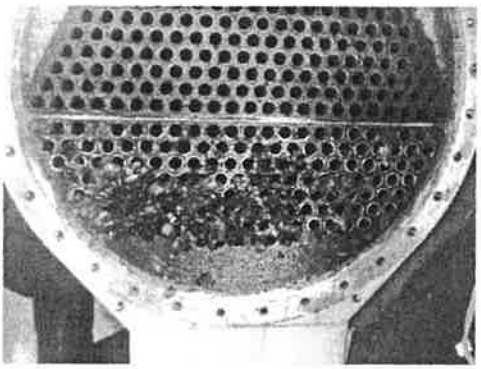
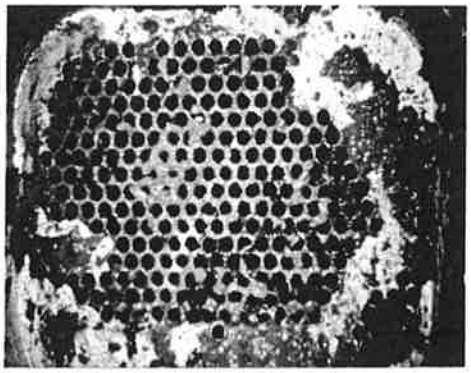
Dirty Condenser



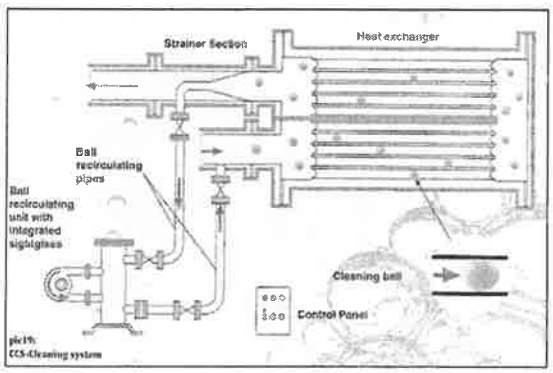


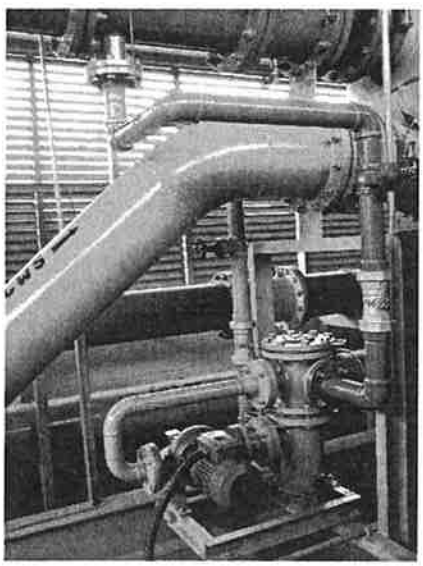
Auto tube cleaners

Your teeth if you only brush once a year



Auto tube cleaners for chillers



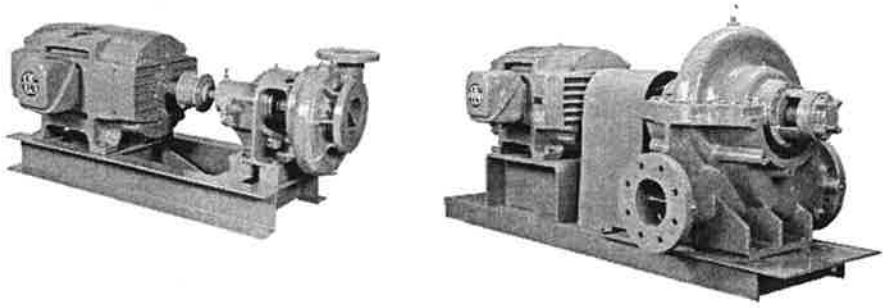


Hx Efficiency and Approach Temperature

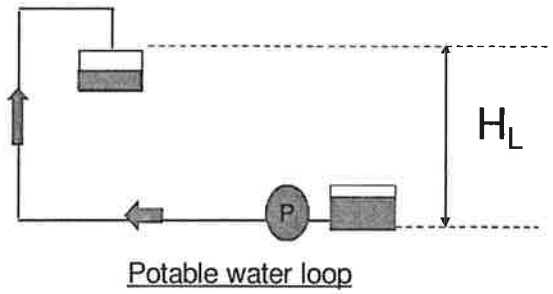
- Use EMS to monitor and trend approach temperatures
  - Via high level interface with chiller
  - Evaporator approach (< 1C)
  - Condenser approach (< 1C)

- Use EMS to monitor and trend condenser water quality
  - Via analog output from water treatment system
  - pH (Control limits: 7-9)
  - Total Dissolved Solids TDS (< 2500 µS/cm)

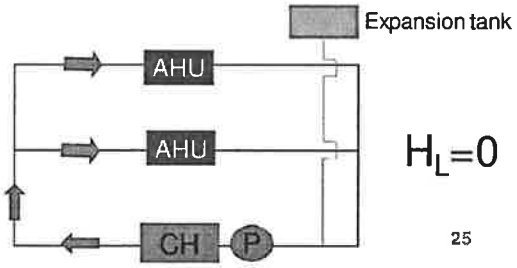
CHWP OPTIMIZATION



Right size pump head



Potable water loop



CHW water loop

25

$H_p = H_L + H_E + H_F$

H_p: Pump head

H_L: Static lift

H_E: Equipment friction loss at design flow

H_F: Pipe and fitting friction loss at design flow

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Right size pump head

- Design "fat & straight pipe and small pump" instead of "spaghetti pipe and big pump".
- Size pipe to be between pressure drop of 1 to 2 ft / 100ft
- Design for high DT cooling coil to get lower flow per KWc (gpm/RT)
- Select low pressure drop evaporators : < 10' WG (36 kpa)
- Select low pressure drop cooling coils: < 12' WG (36kap)
- Select low pressure drop control valves: < 10' WG (30kpa)

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Select HE pump and motors

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Small size pumps (700 to 1200 gpm)
End suction pump
Pump Efficiency >82%
Motor Efficiency> 92%

Medium size pump (1200 to 2000 gpm)
End suction or Split casing pump
Efficiency > 84%
Motor Efficiency > 93%

Large size pump (>2000 gpm)
Split casing
Efficiency > 85%
Motor Efficiency > 95%

Case Comparison

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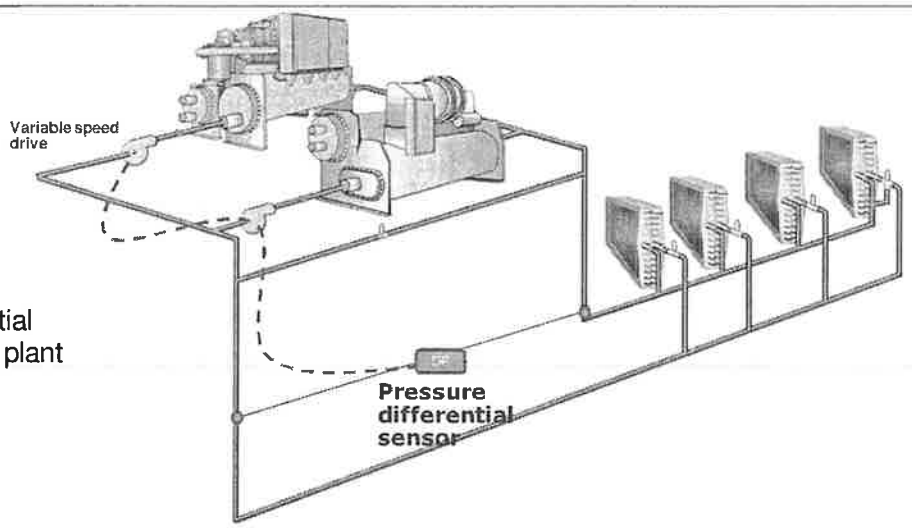
Description	Units	Conventional design	Low pressure design	Remarks
Flow rate	Gpm	2,400	2,400	
Chiller evaporator pressure drop	Ft wg	25	12	
AHU coil pressure drop	Ft wg	25	10	
Control valve pressure drop	Ft wg	25	10	
Pipe and fittings	Ft wg	150	30	Spaghetti vs fat & straight
Resulting pump head	Ft wg	225	62	
Pump eff	%	75%	85%	
motor eff	%	90%	95%	

Microsoft Excel Worksheet

Case Comparison

Description	Units	Conventional design	Low pressure design	Savings
Power Input	KW	150.7	34.7	116.0
Operating hours / year	Hrs	8,760	8,760	-
Annual Energy	KWH	1,320,194	304,094	1,016,100
Elect tariff	$/kwh	0.25	0.25	-
Annual operating cost	S$	330,048	76,024	254,025
CO2 emission	kg	660,097	152,047	508,050

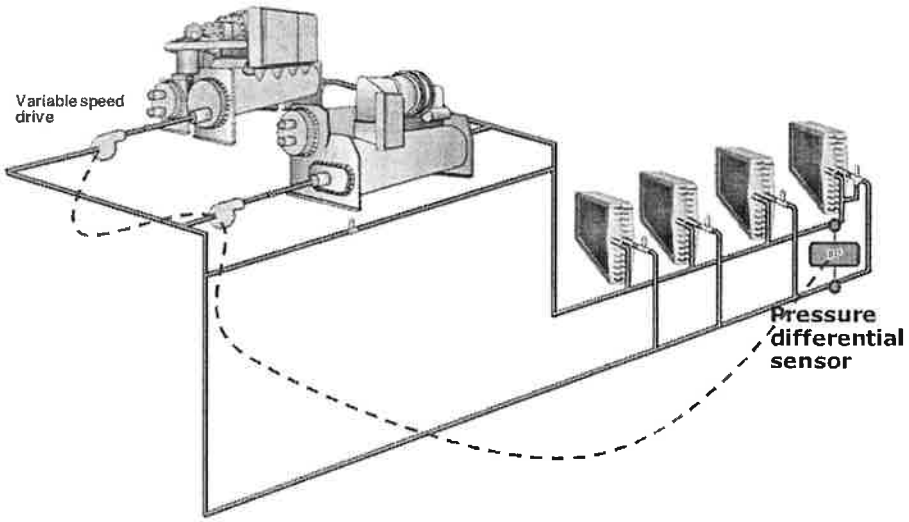
Pump speed control



Pressure differential sensor located at plant room

The distribution pump is typically equipped with a variable-speed drive that is controlled to maintain a certain pressure difference between the supply- and return-water piping. In response to a reduced cooling load, the two-way valve modulates closed, restricting the flow of water through the coil. This causes an increase in system differential pressure, which can be measured and used to signal a reduction in the speed of the distribution pump. In many CHW plants, the differential pressure sensor is located in the chiller plant room out of convenience.

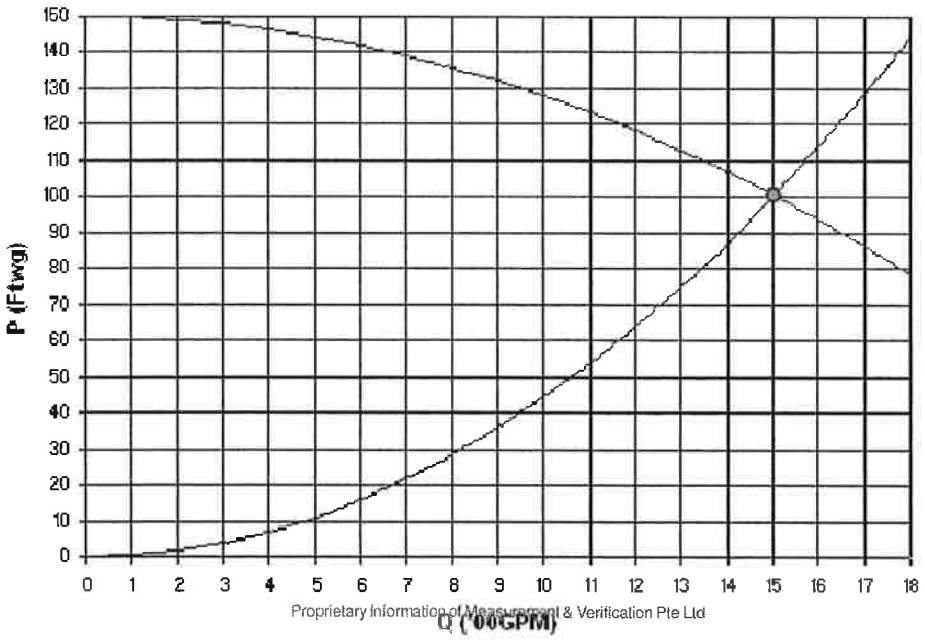
Pump speed control



Pressure differential sensor located at remote AHU

The differential pressure senor could also be place at a remote AHU location. However this entails more wiring.

CHWP speed control using DP sensor Full load



CHWP speed control
Part load

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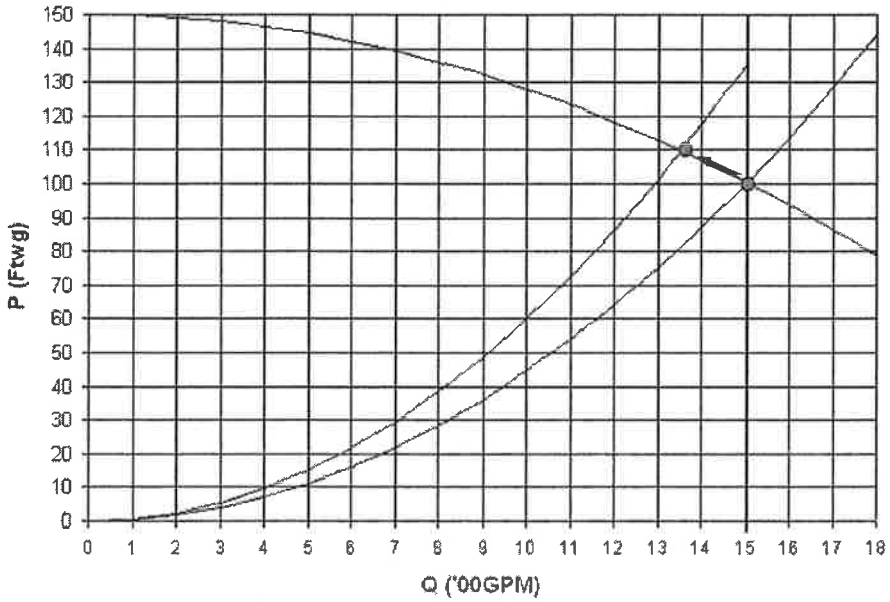
Operating point will ride up the pump curve



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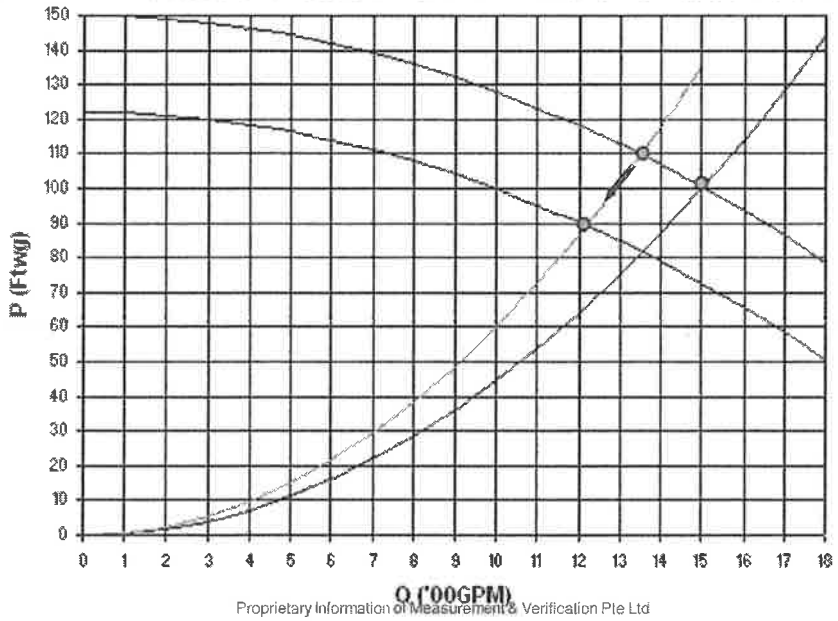
CHWP speed control
Part load

Measurement & Verification Pte Ltd

The DP sensor detects the raised DP & will command VSD to slow down



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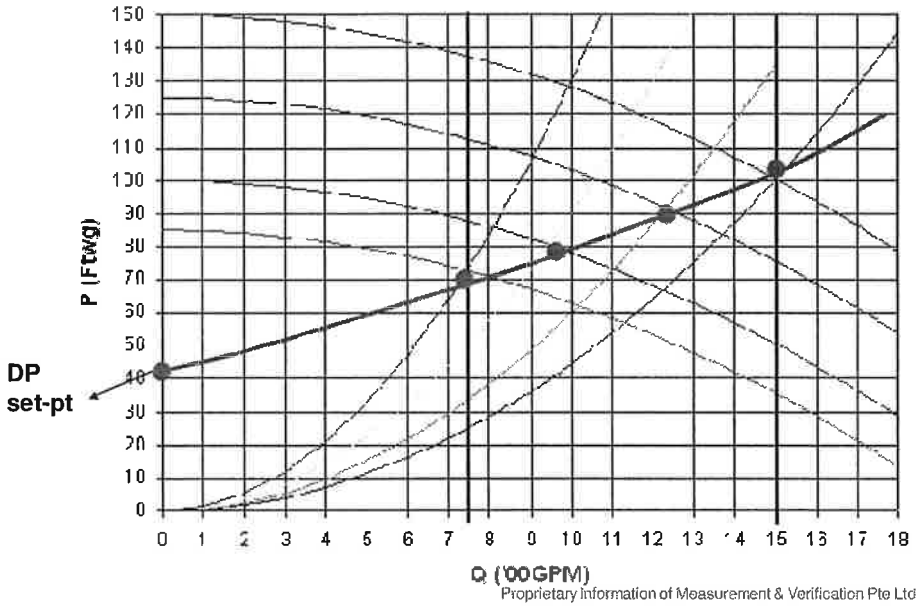
CHWP speed control
Part load



The system unloading curve is represented by the red line which ends at the DP set-point.

To optimize the pumping energy at part load, the DP sensor should be placed at the most remote AHU. If the DP sensor is placed near the pumps, the DP sensor set-point will have to be higher. This will reduce the unloading potential of the pumps.

CHWP speed control
Dynamic set-point

The building automation system shall monitor the valve position of all valves and determine the system's critical zone valve (CZ), which is the valve that is the widest open.

Critical zone valve position	
>95% open	Reset pump DP pressure set-point upwards
	Do nothing
<85% open	Reset pump DP pressure set-point downwards

Beware: Just one wrong valve position or feedback can lead to wrong CHWP speed control

CHWP speed control

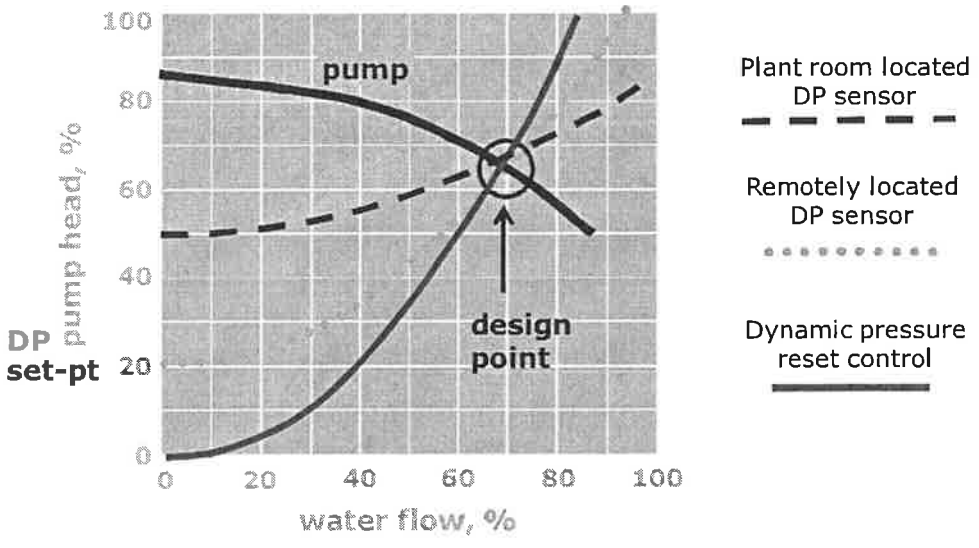
Dynamic set-point

The DP set-point can by dynamically set based on other parameters such as:

1. # of chillers operating
2. Cooling load
3. Peak period and off peak period

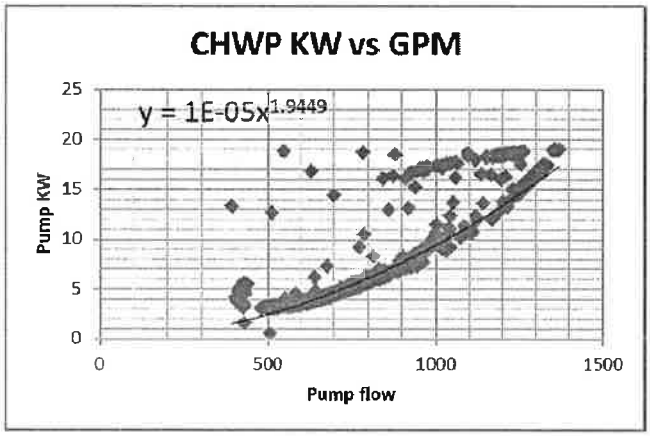
The set-point has to be optimized and fine tuned for each project to attain the sweet spot.

CHWP speed control



CHWP Part Load Efficiency

Actual measurement where pump speed is controlled by DP at riser.



Pump affinity law only applies when system curve does not change as flow varies.

In most CHW systems, the system curve becomes steeper during part load as modulating valves throttle to reduce cooling.

<u>If pump speed is controlled by DP,</u>

KW $\propto$ (flow) $^{1.5 \text{ to } 2.3}$

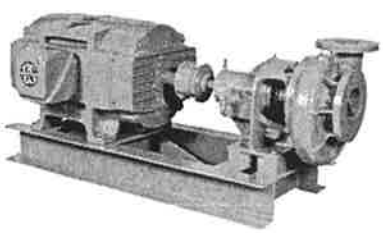
<u>If pump speed is controlled by dynamic reset,</u>

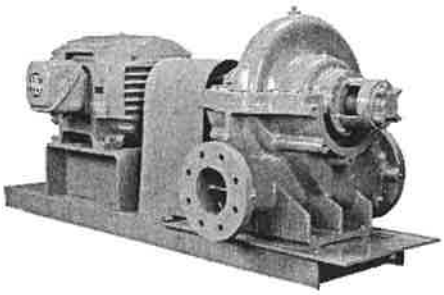
KW $\propto$ (flow) $^{2.5 \text{ to } 3}$

CHWP speed control

- CHWP speed turn-down ratio is limited by Chiller Min Allowable Flow.
- Typical min allowable flow is 50 to 75% range.
- Chiller efficiency is typically unaffected when flow is above the minimum so VPF is always recommended.
- Chiller with lower minim allowable flow is preferred as it can help save pump power.
- EMS must be programmed to modulate open the by-pass pipe control valve to keep the flow through each chiller above the minimum limit.

CWP OPTIMIZATION

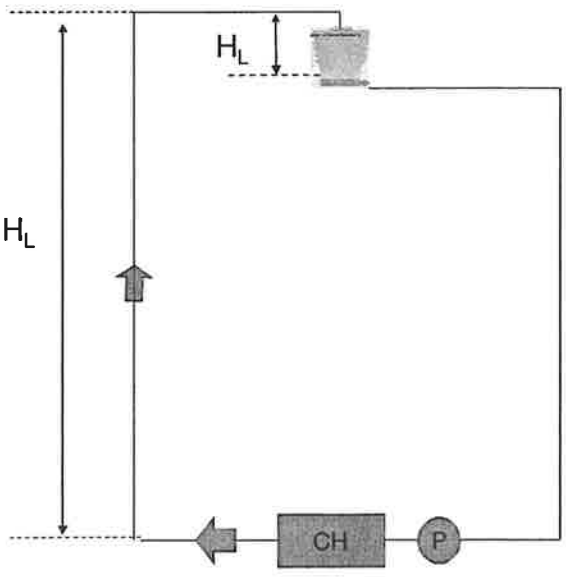




Right size pump head

$H_p = H_L + H_E + H_F$

H_p: Pump head
H_L: Static lift
H_E: Equipment friction loss at design flow
H_F: Pipe and fitting friction loss at design flow



Common mistake to assume HL is = height of building because CW loop is open

Right size pump head

- Design "fat & straight pipe and small pump" instead of "spaghetti pipe and big pump".
- Size pipe to be between pressure drop of 1 to 2 ft / 100ft
- Select low pressure drop condenser: < 12' WG (36 kpa)
- Select low pressure drop cooling tower: < 10' WG (30kpa)

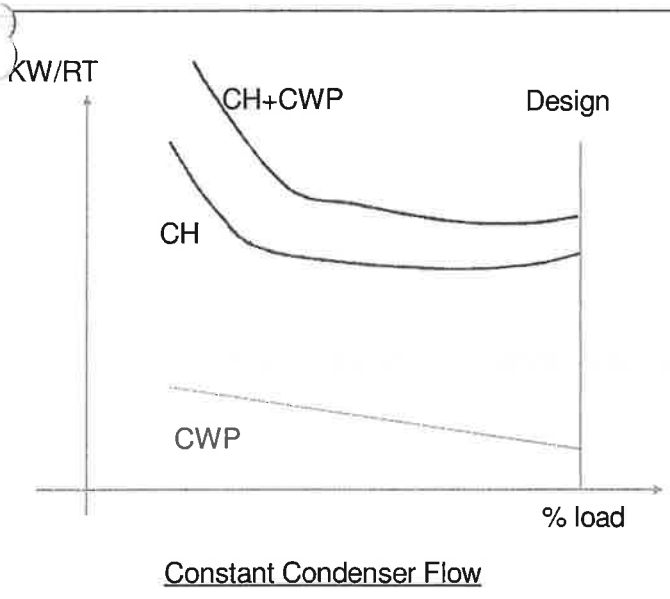
CWP Part Load Efficiency

- There are no control valves CW system, so the system curve does not change at part load.
- Therefore, for variable condenser flow system, the pump affinity laws applies:
  - Head $\propto$ $(\text{flow})^2$
  - Power $\propto$ $(\text{flow})^3$

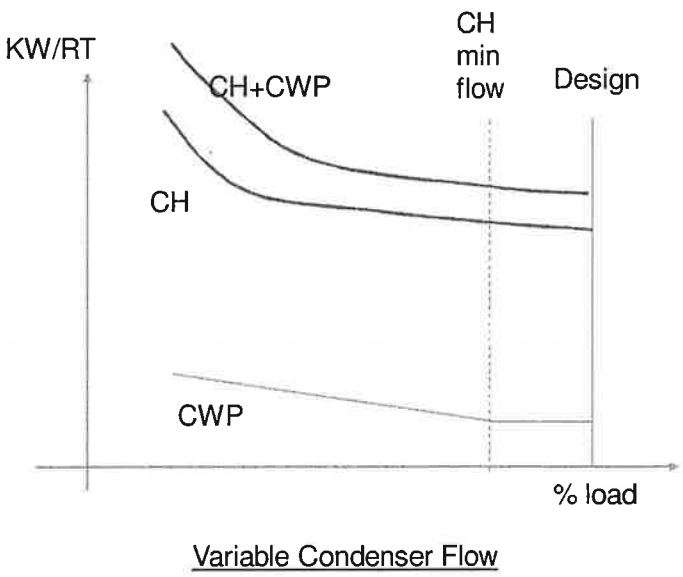
CWP speed control

- Chiller efficiency is sensitive to reduction in condenser water flow. Need to consider the net benefit when applying variable condenser water flow.
- Most projects use CWP VSD as only "adjustable only" ie adjust VSD speed to achieve design flow only.
- Variable flow control method can be gpm/RT.
- The set point needs to be fine tuned at site. Depends on chiller characteristics and pump size.
- Flow turn-down is limited by Chiller Allowable Min Flow.
- Typically minimum allowable is only about 80% of design flow.

CWP speed control



Constant Condenser Flow



Variable Condenser Flow

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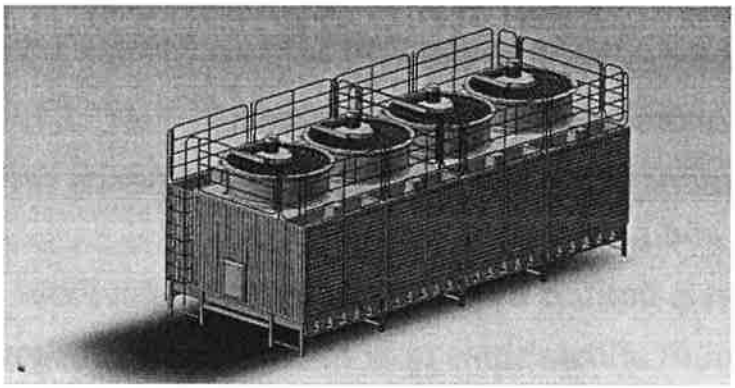
COOLING TOWER OPTIMIZATION



Cooling tower layout

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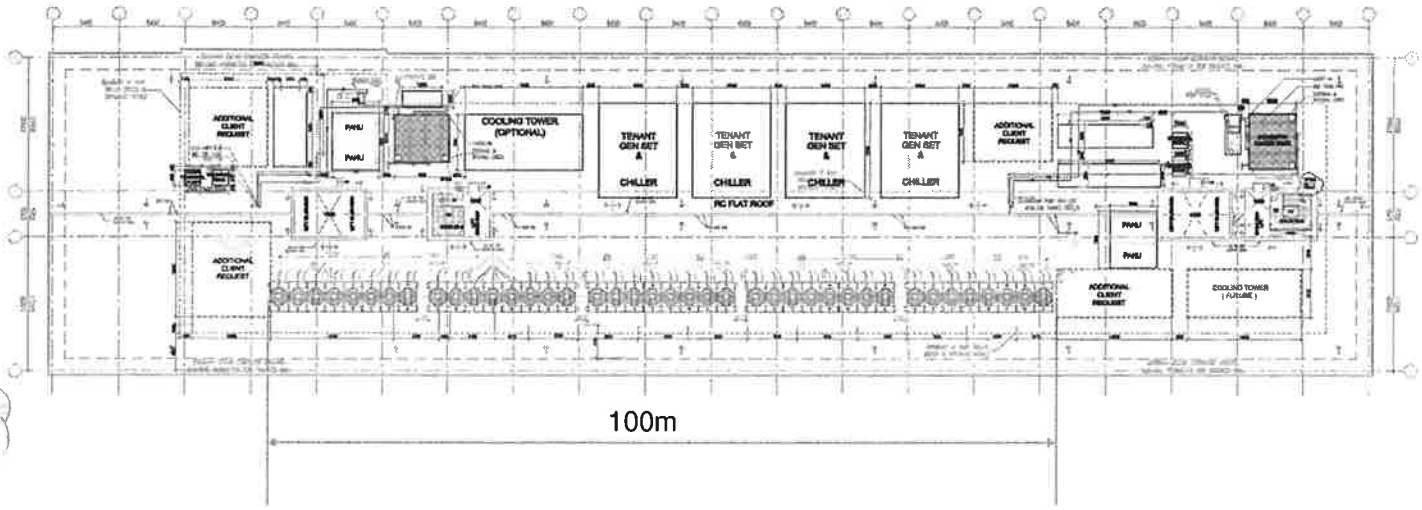
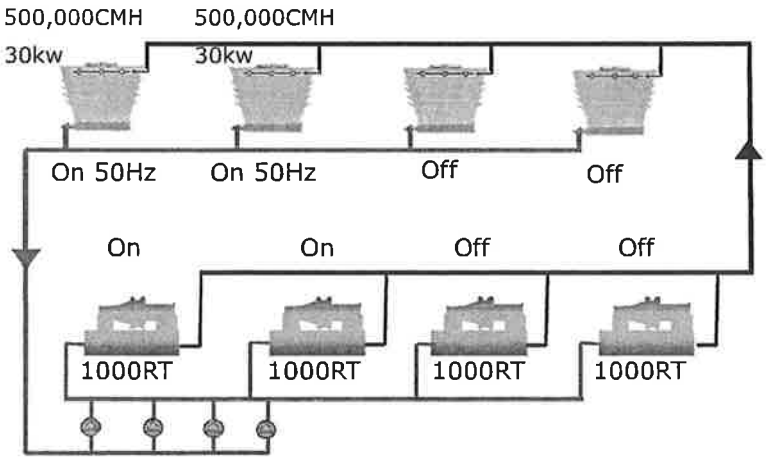
- High efficiency chiller plant is not possible if CT heat rejection path is congested.
- This message has to "hammered" into owners and architects who are used to cramming and camouflaging cooling towers.
- The ideal CT layout is in a straight row with at least "1H" of clearance on the air intake side.
  - Where 1H is the height of the cooling tower infill.

Cooling tower layout

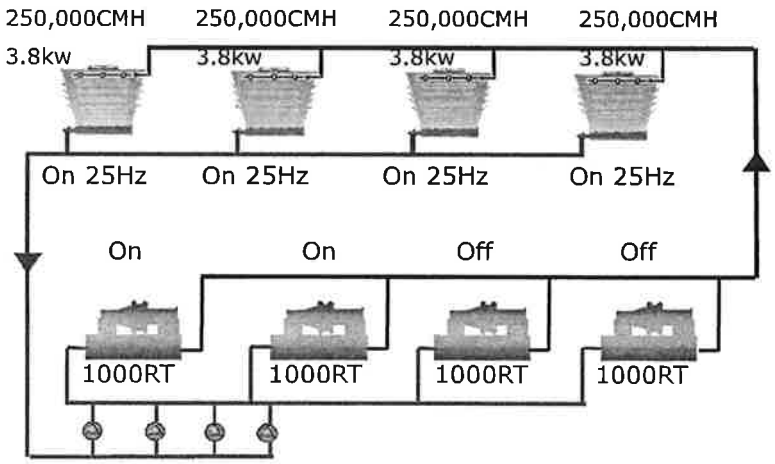


Conventional Cooling Tower Operation



Total CMH=1,000,000
Total KWi = 60

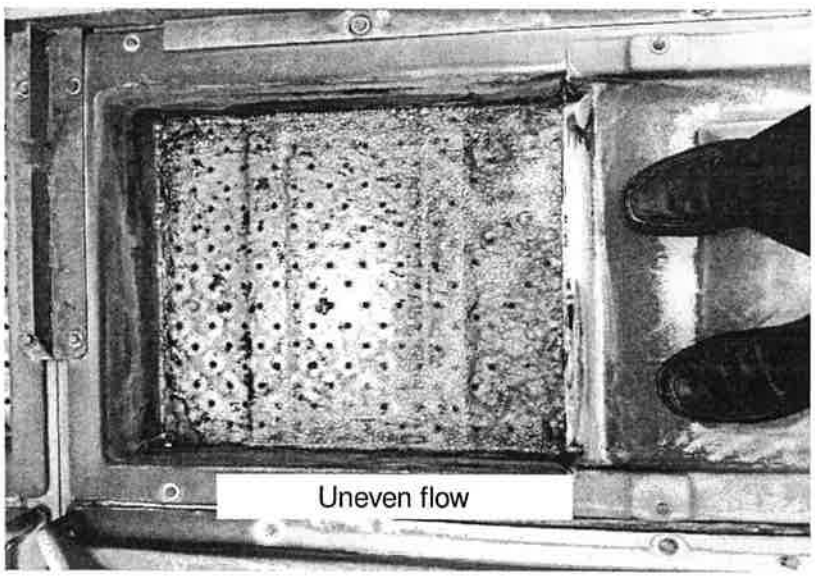
Cooling Tower Operation Optimization



Individual CT KW $= \left(\frac{250{,}000}{500{,}000}\right)^3 \times 30KW = 3.8KW$

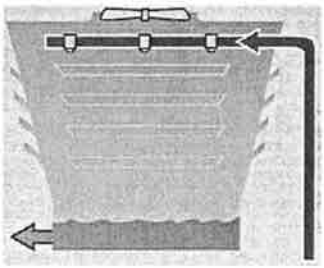
Total CMH=1,000,000
Total KWi = 15.2

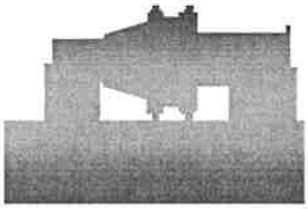
Cooling Tower Operation



Beware of uneven flow into cooling towers infills when there are many more CTs in operation for lesser chillers

Operating Dependencies

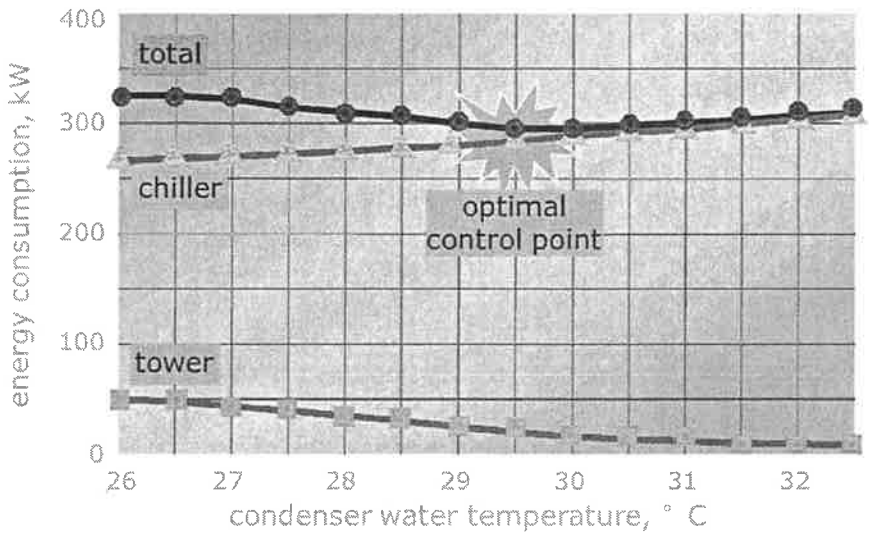




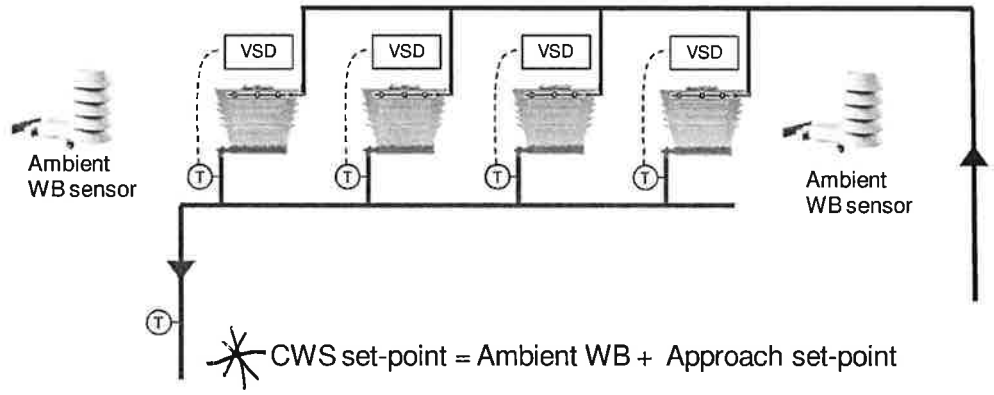
- Wet bulb
- Condenser water temperature
- Load
- Tower design

- Load
- Condenser water temperature
- Chiller design

Chiller–Tower Interaction



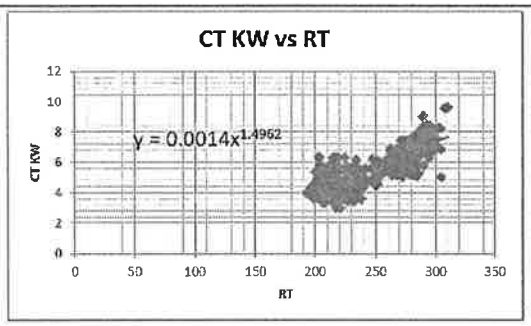
Cooling Tower Fan Speed Control

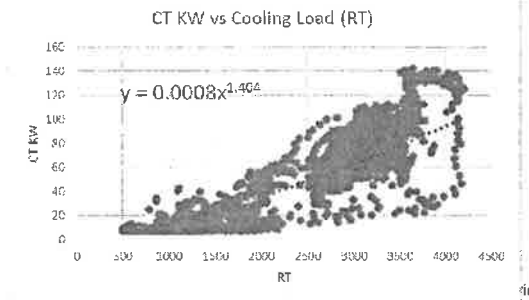


- Where ambient WB is favorable, CWS temperature set-point is lowered to reduce chiller energy.
- Approach set-point is fine-tune at site for best overall CT and chiller energy net efficiency.
- Approach set-point can be variable based on # of CTs, cooling load or on/off peak period.
- Good practice to have multiple sensors to enable continuous consistency checks.

Two WB sensor tend to (comparison) data.

CT Part Load Efficiency





The cooling tower fan head and fan power follows closely to the affinity law as airflow is reduced.

However, the CW evaporation effect is affected at part load because less air crosses the water.

Furthermore, when more cooling towers are operated in parallel, the CW evaporation effect is affected as the flow down the infill is less turbulent.

Heat rejection is also affected by WB. Therefore, the relationship between CT KW and RT varies widely. Often CT part load performance is not available from suppliers.

KW $\propto$ (RT) $^{1.0 \text{ to } 2.0}$

Optimization of chiller plant

There is no off-the-shelf solution to optimization of a chiller plant.

Optimal efficiency requires each plant to be fine tuned:

- Experienced engineer is essential
- Experienced EMS programmer is essential
- Accurate instruments installed properly is essential
- User friendly EMS is essential
- Powerful graphical trending is essential
- Remote monitoring is essential

THANK YOU



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Airside Optimization

Steven Kang
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Agenda

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1. Airside system optimization & IAQ
2. Comparison of AHU and FCU
3. Optimization of AHU system
4. Optimization of FCU
5. CO2 demand ventilation

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Airside System Optimization & IEQ

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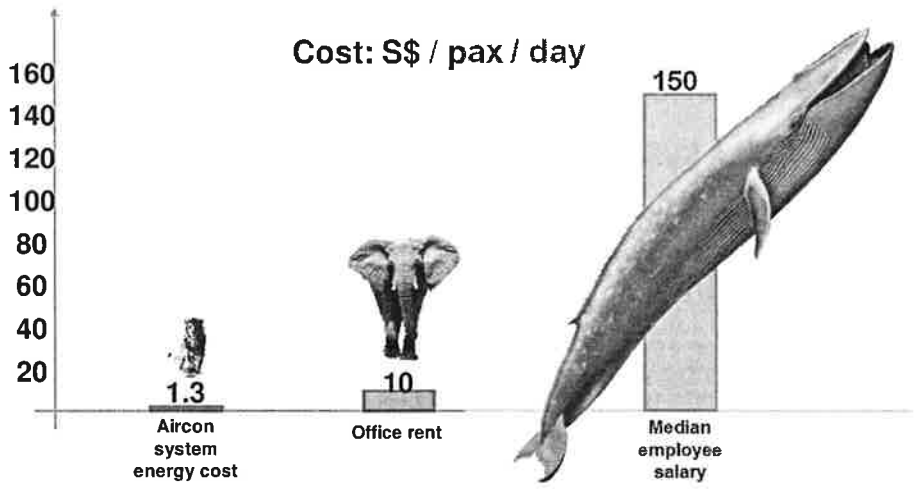
Where does it really count for a building?



Make sure your building IEQ is conducive for high performance.

Some energy cost saving strategies at the expense of IEQ

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- Raise room temperature
- Reduce airflow
- Use low rating filters
- Reduce outside air
- Minimize maintenance services
- Squeeze all M&E equipment above ceiling







Goals for Good Air Quality (IAQ)

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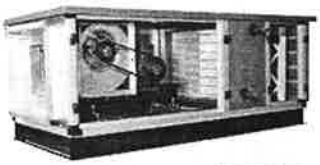
Goals for good airside efficiency



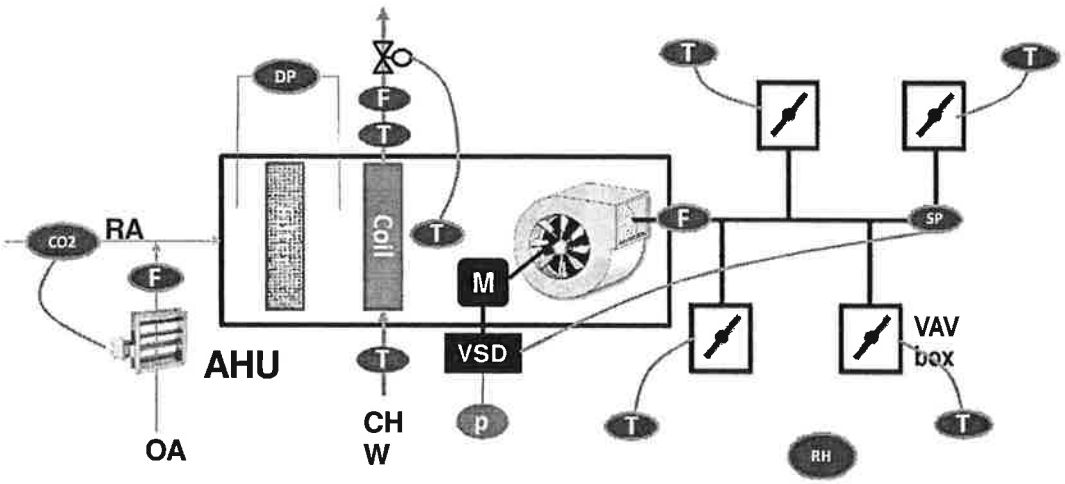


$$\frac{W}{CMH} = \frac{\text{Total Pressure}}{3600 \times \eta_{fan} \times \eta_{motor}}$$

Types of air system	Green Mark (W/CMH)	Green Mark (KW/RT)	Excellent class (W/CMH)	Excellent class (KW/RT)
Constant Volume AHU Motor ≥ 4KW	0.42	0.29	0.20	0.14
Variable Volume AHU Motor ≥ 4KW	0.58	0.39	0.20	0.14
AHU/FCU Motor < 4KW	0.17	0.12	0.10	0.07

KW/RT = W/CMH * 680 CMH/RT /1000

Accurate measurement is the essential to "drive towards our goals"



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Comparison of AHU and FCU

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AHU versus FCU

Factors	Sub-factors	AHU	FCU
Energy efficiency	Fan ESP	High due to long ducts (ESP: 300 to 500pa)	Low due to short ducts (ESP: 50 to 100a)
	Fan efficiency	High (60 to 75%)	Low: (40 to 50%)
	Motor eff	High (85 to 93%)	Low: (35 to 50%)
	Overall efficiency	**0.20 to 0.70 W/CMH**	0.15 to 0.30 W/CMH
Cost	Initial cost	Generally high as custom built & require lots of ducts	Generally lower as standard product
	Life span	More durable (10 to 15 years lifespan)	Less durable (7 to 10 years lifespan)
Comfort	Temp	Generally better due to better temp, RH control	Generally lower
		VAV needed for zone temp control	Zone temp control is std
	IAQ	Better IAQ due to better filters. UV possible.	Flat filter
	Noise	Quieter as AHU confined and duct can attenuate noise	More prone to noise problem
Space requirement	Room	Generally need AHU room	Ceiling mounted
	Ceiling	Larger ducts, so more ceiling depth required	Smaller duct, so less ceiling depth required
Maintenance	-	Easier since less qty and floor mounted	More challenging due to large qty and ceiling mounted
Cooling coil	-	Can be customized for different temperatures. High DT &/or low DP possible.	Std conditions only
Fan	-	Can be customized for different CMH, static and fan types	Usually std FC fans only
Filters	-	Can be customized for various filtration efficiency	Usually std low efficiency filters only
Applications	-	High grade offices Large open areas such as function rooms Factories	Low grade offices Hotel room Retail shops Class rooms

Proprietary Information of Measurement & Verification Pte Ltd

Microsoft Excel Worksheet

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Optimization of AHU systems

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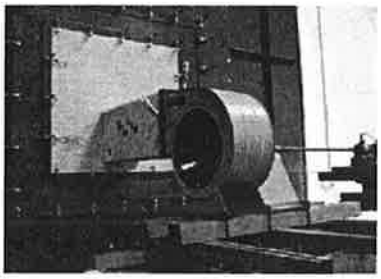
Optimization of AHUs

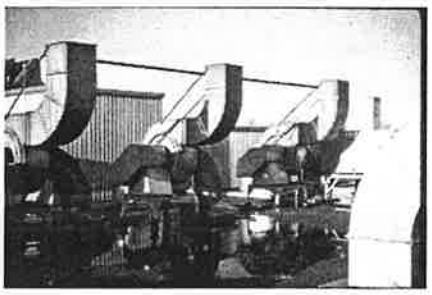
1. Minimize fan system effect losses
2. "Fat short ducts with gradual bends" instead of "long skinny ducts with sharp bends".
3. Select high efficiency fans and motors.
4. Variable Air Volume (VAV) system using VSD instead of Constant Air Volume (CAV) system.
5. At the least, Adjustable Air Volume system using VSD.
6. Select low pressure drop filters
7. Low face velocity when space permits
8. Select high efficiency fans
9. Select high efficiency motors
10. High CHW DT, especially for PAHU.
11. Heat recovery for PAHUs

Fan System Effect Losses

- A factor used to correct for system induced installation effects.
- The difference in performance between a fan tested in the laboratory and one tested in a real installation.



AMCA Rate Fan Condition



Real Installation

Minimize Fan System Effect Losses

Rules of thumbs to minimize fan system effect losses

- Minimum of 2.5 Duct Diameters of straight duct on Outlet
- Minimum 3 to 5 Duct Diameters of straight duct on Inlet
- Avoid inlet swirl or turbulences

Minimize Fan System Effect Losses

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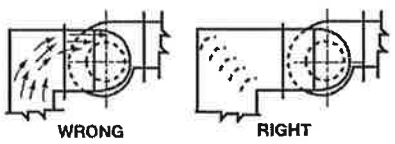
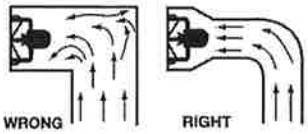
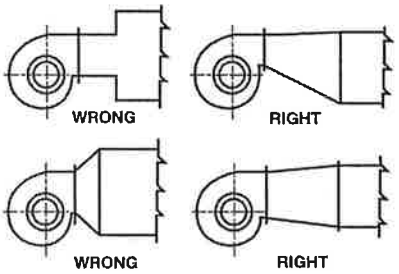
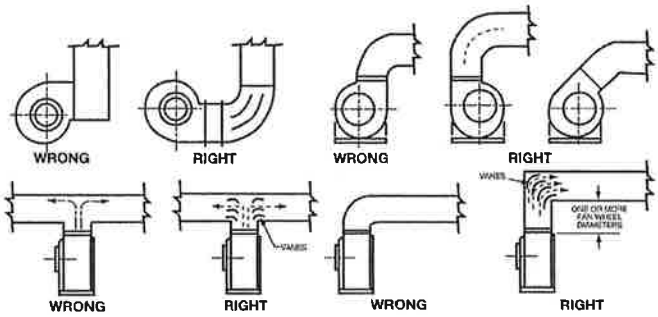
Inlet





Outlet





Example of inefficiencies

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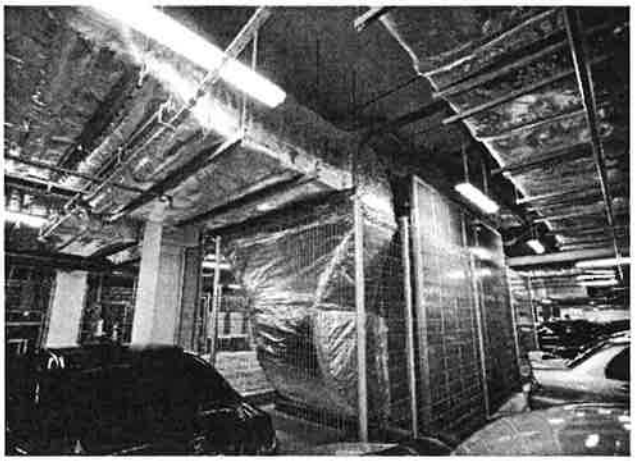

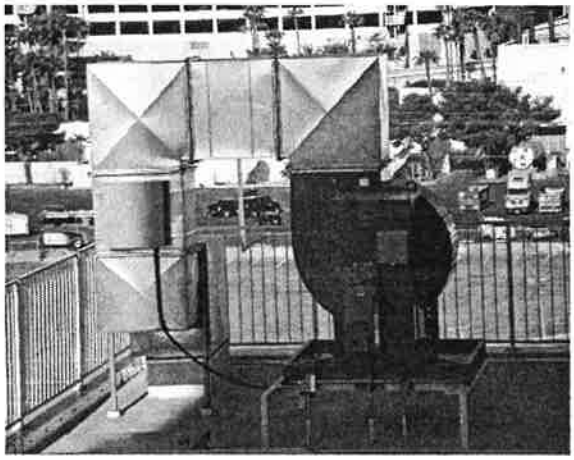

Example of inefficiencies

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Patented

duct

$$\frac{W}{CMH} = \frac{\text{Total Pressure}}{3600 \times \eta_{fan} \times \eta_{motor}}$$



Rocket launcher



Traffic lights needed

Example of inefficiencies

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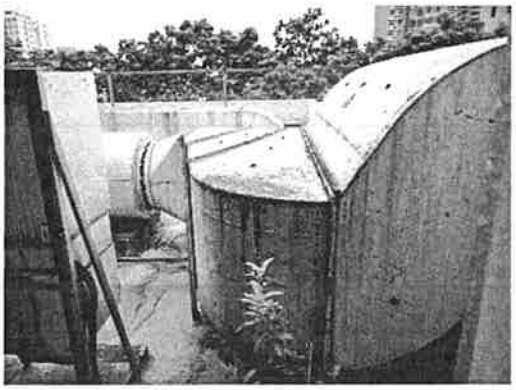

Example of inefficiencies

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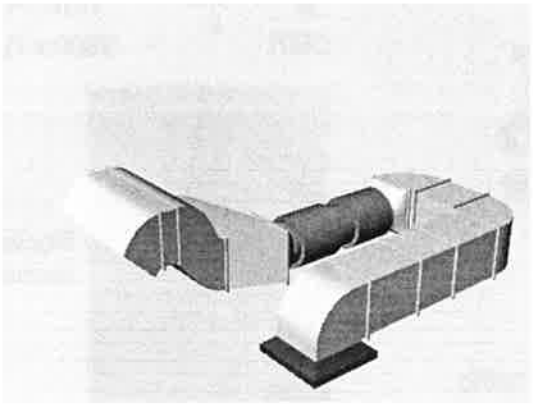

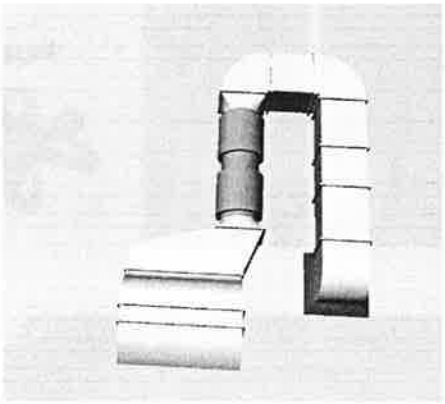

Ideas to improve

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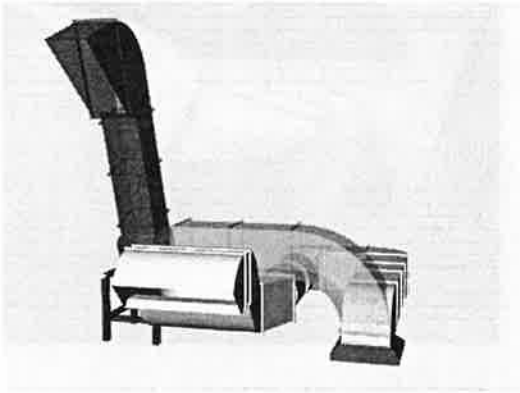
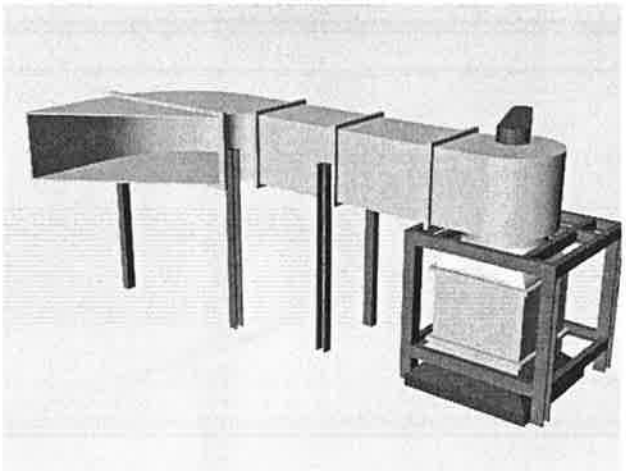


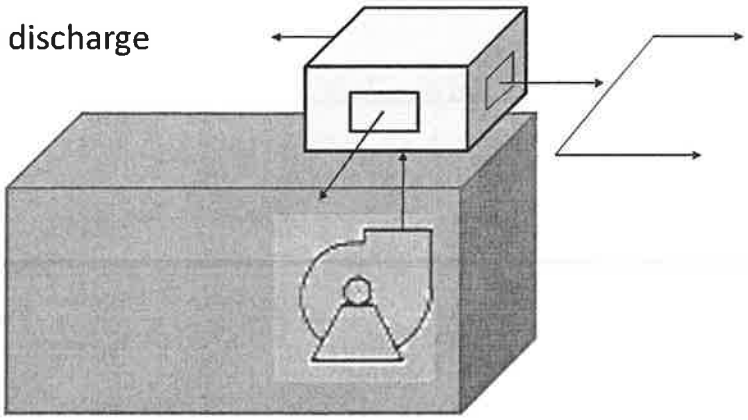


Ideas to improve



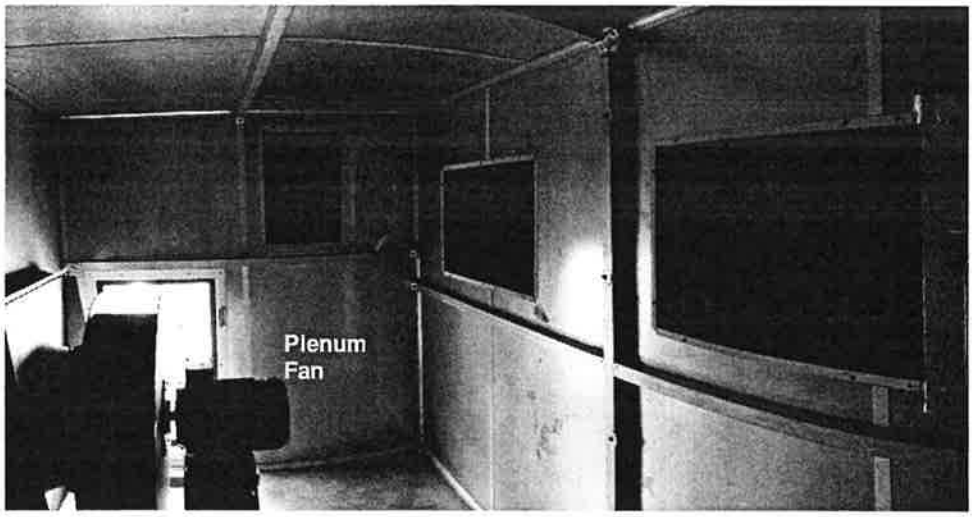
Example of inefficiencies

AHU with multiple discharge



Ideas to improve

Measurement & Verification Pte Ltd



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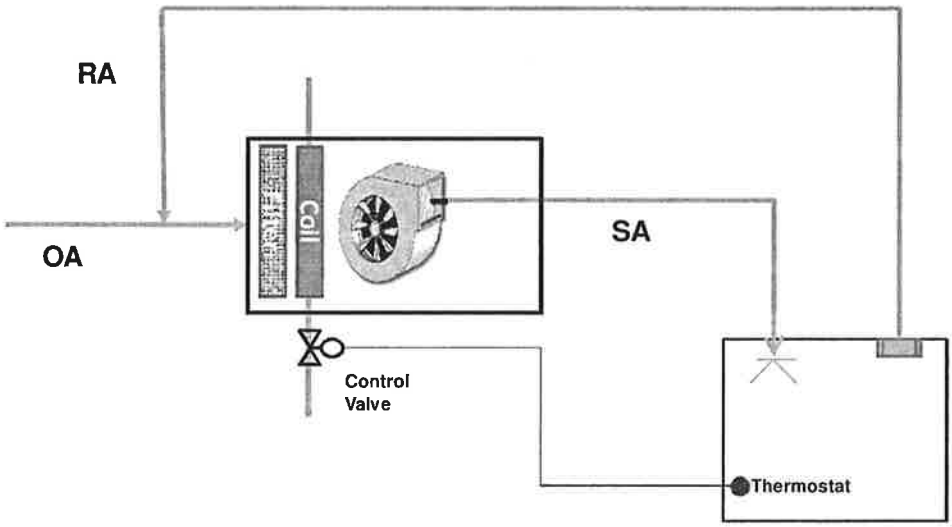
CONSTANT Vs. VARIABLE AIR VOLUME SYSTEM

Constant Air Volume (CAV) System



CAV Full Load

With these design conditions ...

space sensible heat gain = 11.72Kw

space dry-bulb temperature = 23.9C

supply air dry-bulb temperature = 12.8C

SI units

$$\text{Supply Airflow CMH} = \frac{2960 \times \text{Sensible Heat Gain (KW)}}{(\text{Space DB (C)} - \text{Supply DB (C)})}$$

$$\text{Supply Airflow CMH} = \frac{2,960 \times 11.72\text{KW}}{(23.9^{\circ}\text{C} - 12.8^{\circ}\text{C})} = 3,125 \text{ CMH}$$

CAV Part Load

With these part load conditions ...

space sensible heat gain = 5.86KW

space dry-bulb temperature = 23.9

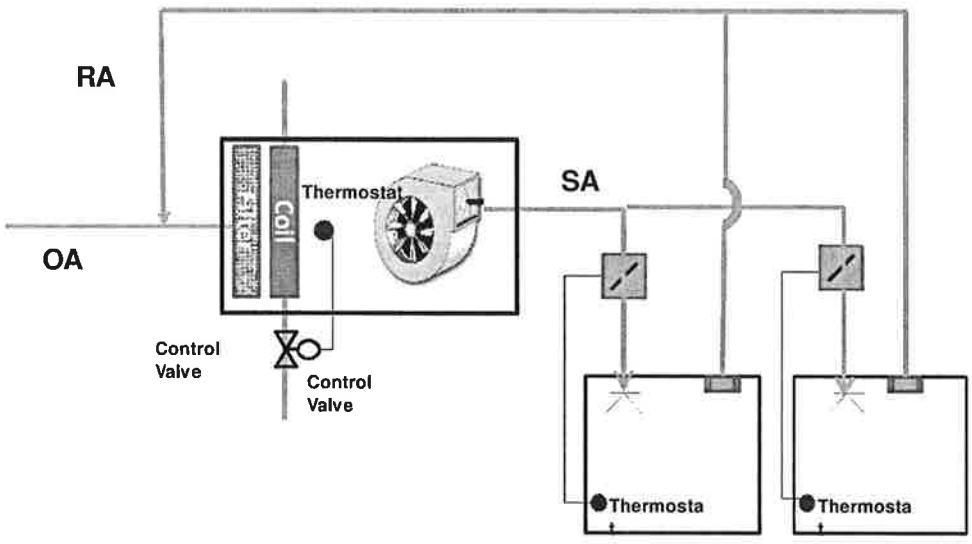
supply airflow = 3,125 CMH

SI units

$$\text{Supply DB (C)} = \text{Space DB (C)} - \frac{2{,}960 \times \text{Sensible Heat Gain (KW)}}{\text{Supply Airflow (CMH)}}$$

$$\text{Supply DB (C)} = 23.9^{\circ}\text{C} - \frac{2{,}960 \times 5.86\text{KW}}{3{,}125\ \text{CMH}} = 18.3^{\circ}\text{C}$$

Variable Air Volume (VAV) System



VAV Full Load

With these design conditions ...

space sensible heat gain = 11.72Kw

space dry-bulb temperature = 23.9C

supply air dry-bulb temperature = 12.8C

SI units Supply Airflow CMH $= \frac{2{,}960 \times 11.72\text{KW}}{(23.9^\circ\text{C} - 12.8^\circ\text{C})} =$ 3,125 CMH

VAV Part Load

With these part load conditions ...

space sensible heat gain = 5.86KW

space dry-bulb temperature = 23.9C

supply air dry-bulb temperature = 12.8C

SI units Supply Airflow CMH $= \frac{2{,}960 \times 5.86\text{KW}}{(23.9^\circ\text{C} - 12.8^\circ\text{C})} =$ 1,563 CMH

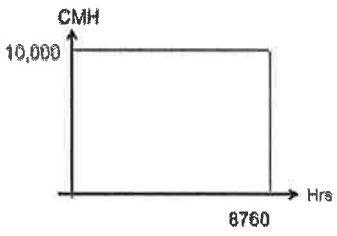
VAV Part-Load Energy Savings

- Reduced fan energy
- Reduced refrigeration energy



Constant Air Volume System

KWH = CMH x W/CMH x Hrs
= 10,000 x 0.47 x 8760/1000
= 41,172 KWH



Fan operating cost = 41,172KWH *0.24/KWH
= S$9,881

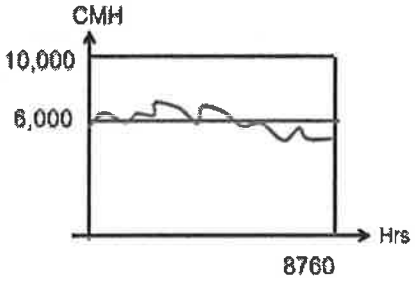
Fan C02 emission = 41,172KWH * 0.59 kg CO2/KWH
=24,291kg CO2 emission

Variable Air Volume System

KWH = CMH x W/CMH x Hrs
= 6,000 x [$(0.60)^2$ x 0.47] x 8760/1000
= 8,893 KWH

Fan operating cost = 8,893KWH *0.24/KWH
= S$2,134

Fan C02 emission = 8,893KWH * 0.59 kg CO2/KWH
=5,247kg CO2 emission



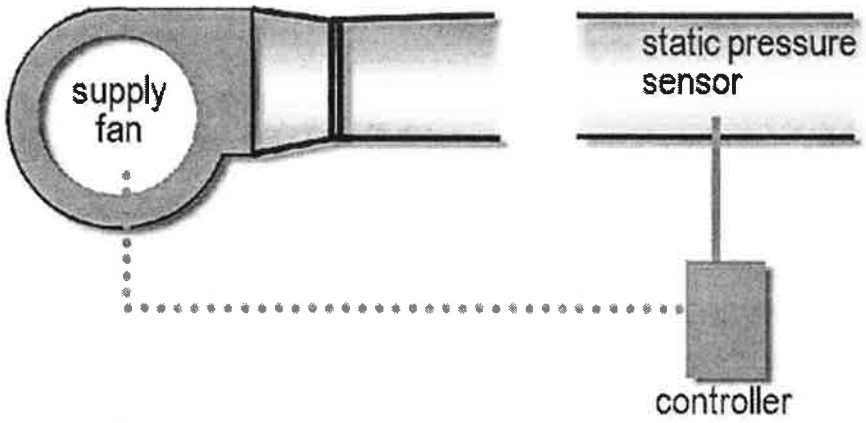
$W \propto Q^3$

$W/Q \propto Q^2$

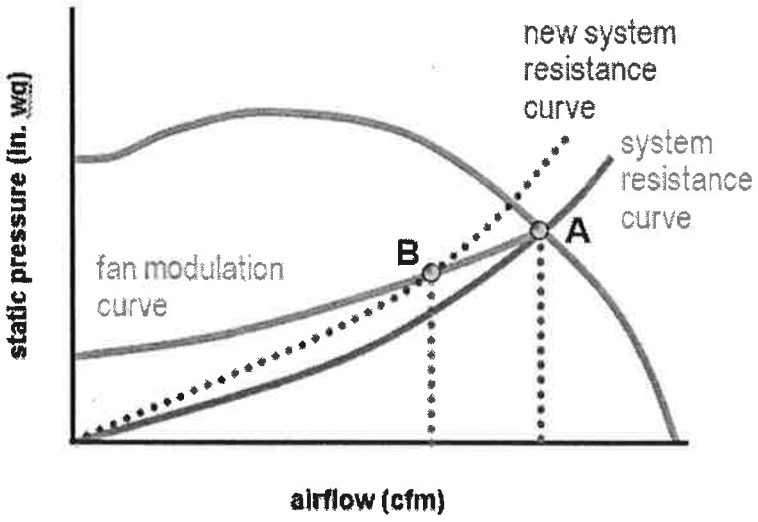
Impact of CAV Vs. VAV system

- Operating cost savings/year= S$7,747
- C02 emission reduction = 19,044kg
- A car CO2 emission/year ~ 5,000kg
- Emission reduction is equivalent to about removing 4 cars

Fan Control Loop

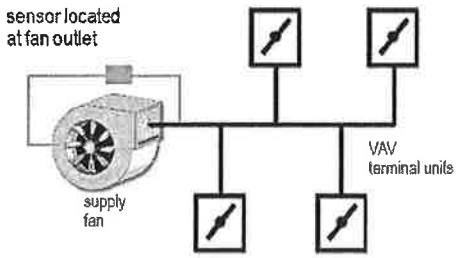


VAV Fan Modulation

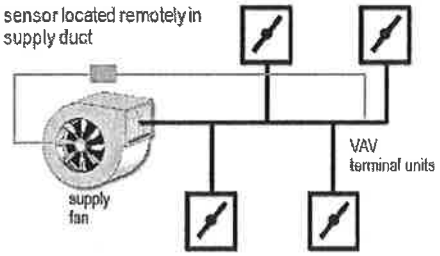


Static Pressure Control

1.Fan Outlet Static Pressure Control

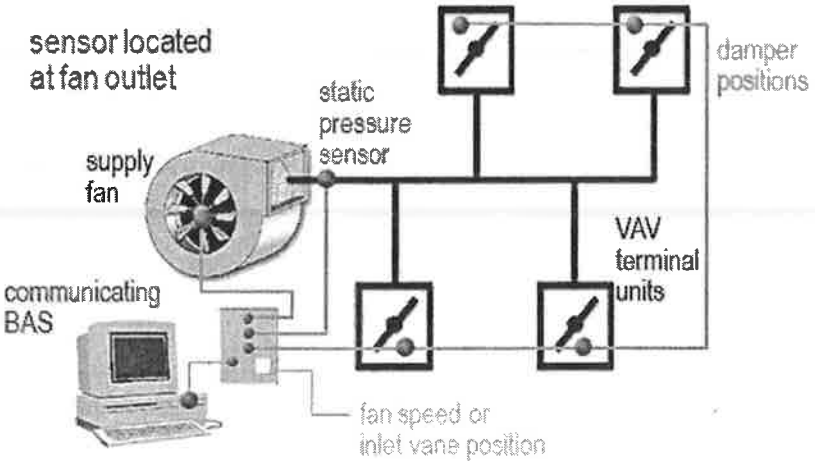


2.Supply Duct Static Pressure Control



Static Pressure Control

3.Optimized Static Pressure Control

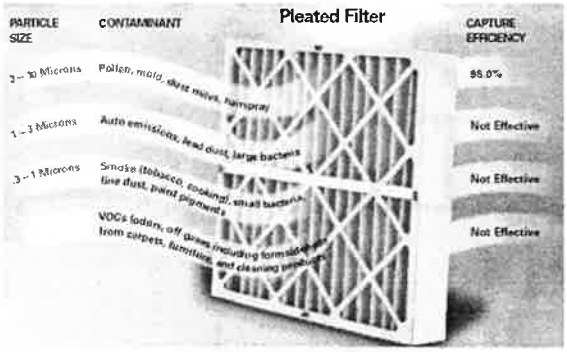


Part-Load Comparison

Control Method	Airflow (Cfm)	Fan SP (In. Wg.)	Brake Hp	% Full-load Power	Relation
Full Load	24,000	2.70	22	---	
Part Load …					
Fan Outlet	18,000	2.13	13	60%	HP α $(cfm)^{1.8}$
Supply Duct	18,000	1.85	12	55%	HP α $(cfm)^{2.1}$
Optimized	18,000	1.52	9.5	43%	HP α $(cfm)^{2.9}$

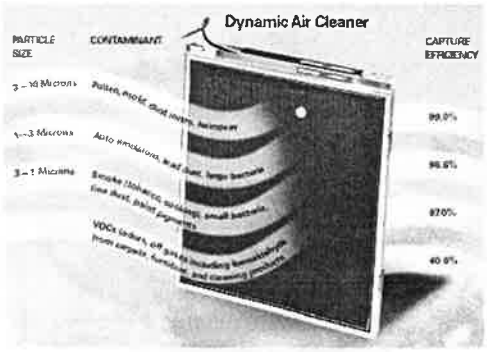
Ideas to improve: Electronic Air Filters

Traditional Passive Filters



Mid-life Pressure Drop: 100 – 200pa

Electronic Air Cleaners



Mid-life Pressure Drop: 30 – 70pa

40

Electronic Air Cleaners

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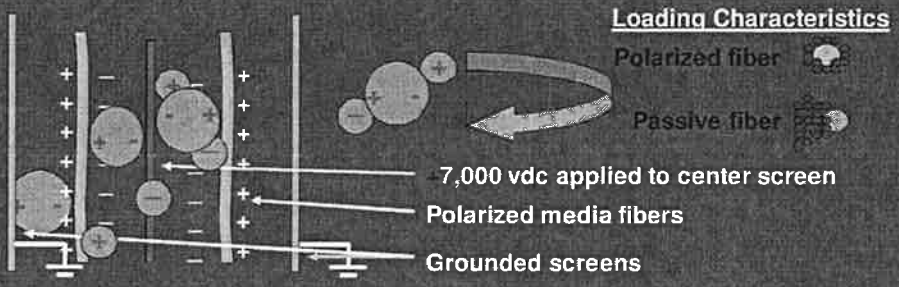
Passive mechanisms:
- because there is a media, there is passive collection.

Polarization and electrostatic attraction:
- polarized fibers collect polarized and charged particles.
- charged center screen collects polarized and charged particles.

Agglomeration:
- natural process (Brownian Motion) greatly accelerated by the field inside the air cleaner. Polarized particles attract each other and charged particles to form clusters that are more easily captured.



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Ideas to improve: Intelligent Control Valve

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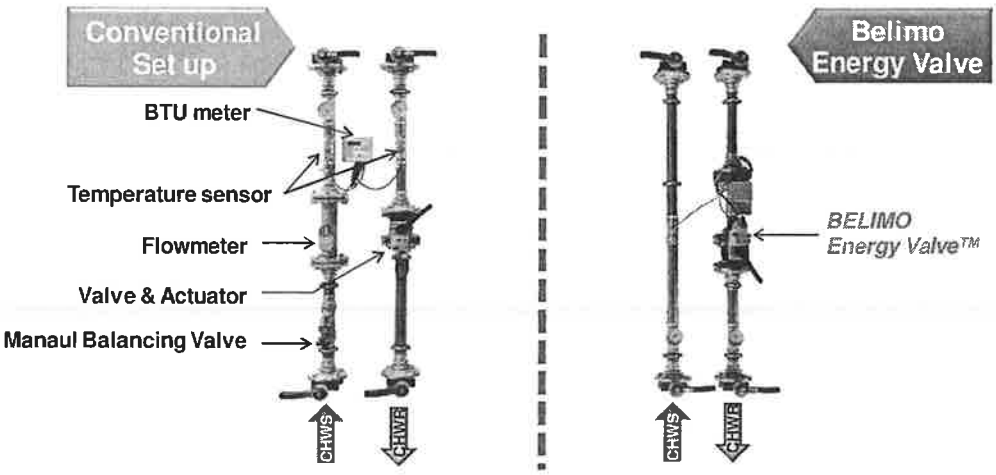

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Ideas to improve: Intelligent Control Valve

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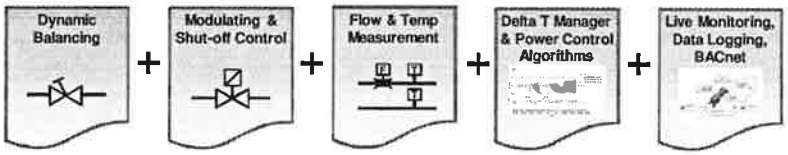
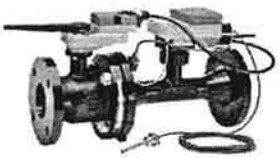
Belimo Energy Valve - Five Functions in One Valve





DN15...50



DN65...150

Key element for improveme nt

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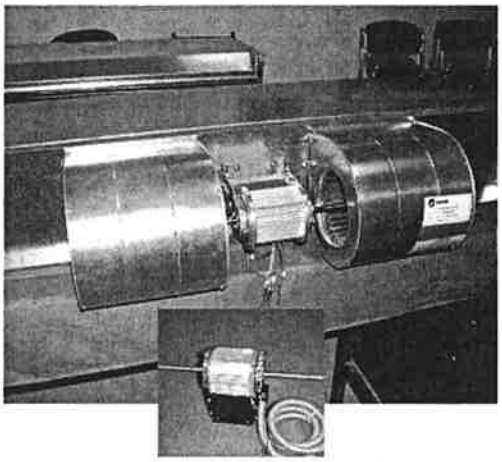
Optimization of FCU

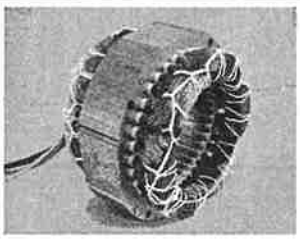
VAV FCU with DC motor



Integrated signal driver to the DC (brushless) fan motor

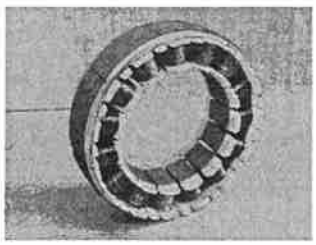
- *Substantial energy saving*
  - **Almost double efficiency than PSC motor**
- *Stepless control*
  - **Non-stage variable speed**
- *Quiet*
  - **Low electro-magnetic noise**
- *Comfortable*
  - **Constant temperature**
- *PI control*
  - **Fast response to the load changes**

VAV Fan Coil Unit -- Motor Coil



Traditional AC Motor

- **Less reliable**
- **Lower energy efficient (35-50% FL eff)**
- **Lower 1st cost**



DC brushless Motor

- **More reliable**
- **Higher efficiency (65-70% FL eff)**
- **Higher 1st cost (approx 2 x vs ac motor)**

Energy Efficiency comparison

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Table 1. Simple payback for replacing three-speed permanent split-capacitor AC (PSC-AC) induction motors with brushless DC (BLDC) motors in fan–coils[a,b]

Nominal airflow, cfm	First cost (est), $ USD		Motor efficiency, %		Energy input, watts			Annual energy cost savings, $ USD	Simple payback, yr
	PSC-AC (std)	BLDC (opt)	PSC-AC	BLDC	PSC-AC	BLDC	Energy saved		
200	—	+70	35	65	55.5	30.0	25.5	16.29	8.6
300			35	70	80.2	39.0	41.2	26.35	5.4
400			35	70	96.4	49.1	47.3	30.29	4.6
600			50	70	115.2	79.1	36.1	23.12	6.0
800			50	70	124.3	93.2	31.2	19.95	7.0

[a] This comparison isn't exactly "apples to apples": The PSC-AC motors varied in size, with the smallest fan–coil using a 1/20 hp motor; but in each case, the replacement was a 1/3 hp BLDC motor.

[b] Saving calculations are based on annual operation of 4,000 hours and an energy cost of $0.08/kWh.

Extracted from Trane *Engineers Newsletter* volume 33–4 (Lit. no. ADM-APN013-EN)

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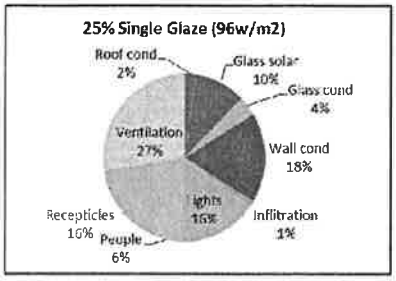
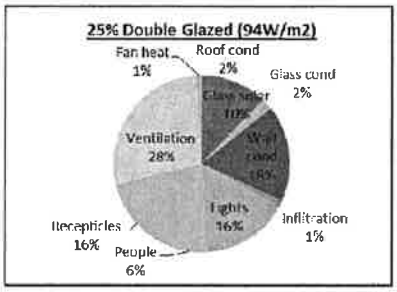
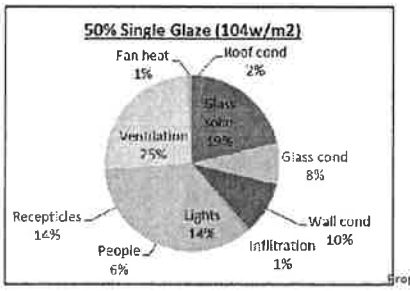
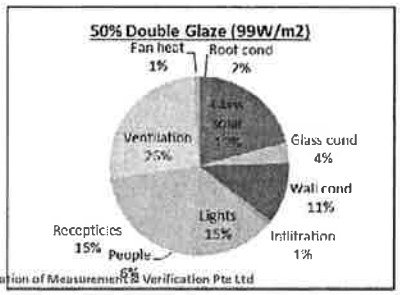
CO2 demand Ventilation

Cooling Load Component Breakdown









Ventilation load is a very large % of total building cooling load due to Singapore's hot & humid climate

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Green Mark Rating

Non residential New Building, V4.0

Part I-G

(g) Sensors or similar automatic control devices are used to regulate outdoor air flow rate to maintain the concentration of carbon dioxide in accordance with Table 1 – Recommended IAQ Parameters of SS 554.

Carbon dioxide acceptable range: ≤ 700 ppm above outdoor.

BCA GREEN MARK

1 point

Is CO2 a Contaminant?

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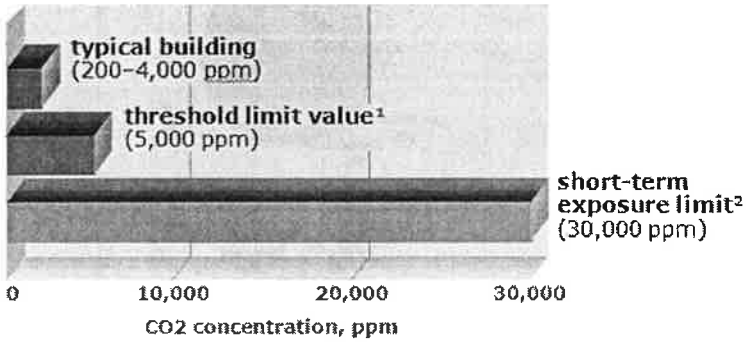
- Not at the concentrations commonly found in buildings



Notes:
[1]Time-weighted average over five 8-hour workdays
[2]Fifteen-minute exposure

Sources:
- Occupational Safety and Health Administration (OSHA)
- American Conference of Governmental Industrial Hygienists (ACGIH)

CO2 Consumption & Production

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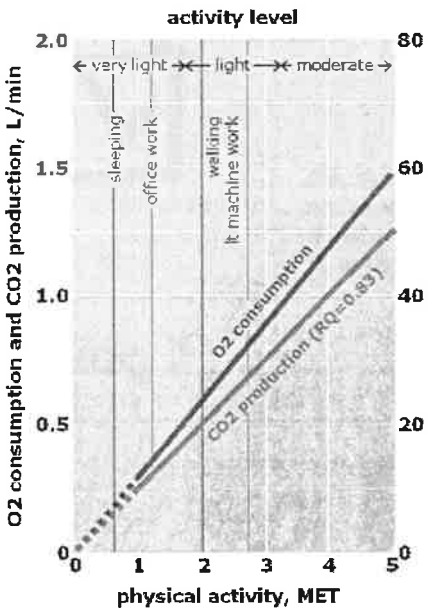
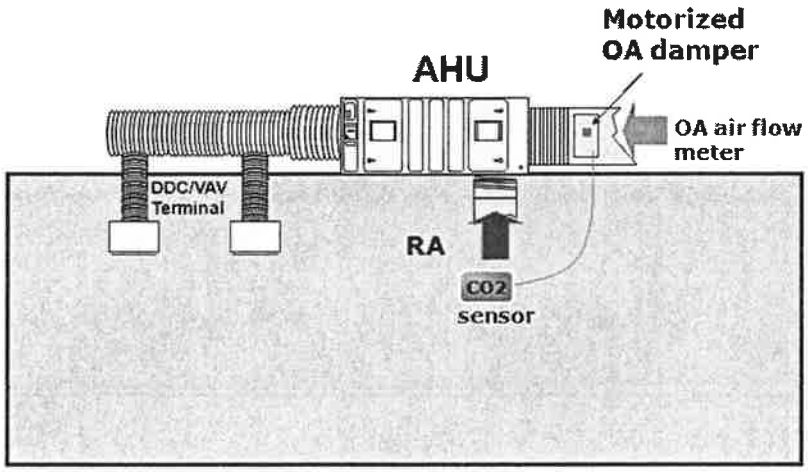


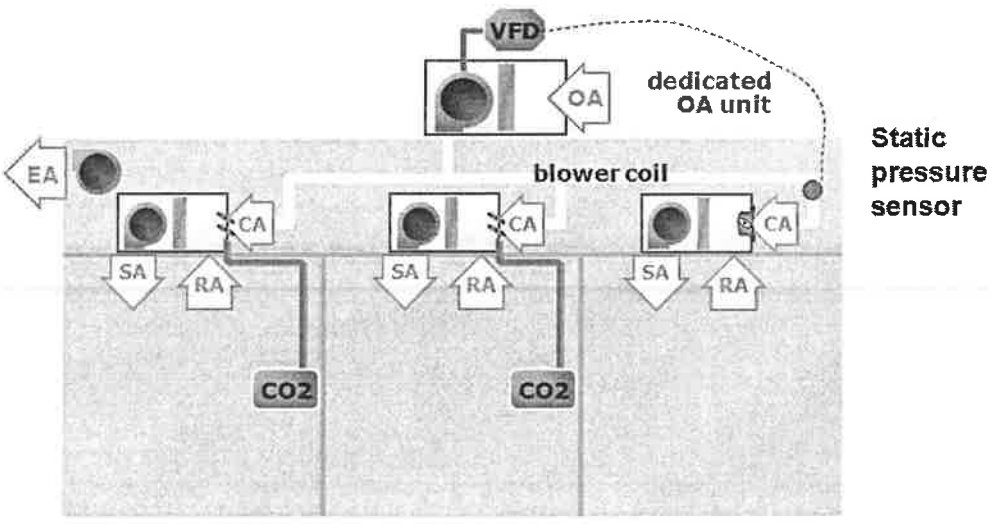
CO2 production is related to person's level of activity …

So CO2 can be used as a "tracer gas" for bioeffluents

Single Zone CO2 Controlled Ventilation



Multi zone CO2 Controlled Ventilation with PAHU



THANK YOU

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