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Volume 6, chapter 7

Improving Energy and Water Consumption in Existing and New Buildings

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1.0 PURPOSE

1. The purpose of this document is to provide the Entity the techniques that can be used to lower consumption of electricity and water in existing buildings, and ways to improve design of the Heating, Ventilation, and Air Conditioning (HVAC) system and other building services systems. Technical explanations are also included for the reader to understand why the recommendations are being made and how each technique affects power and water consumption. The document also includes guidance when techniques are applicable, and important points to consider when techniques are implemented.
2. The document is prepared as supporting document to EPM-KEM-GL-000001, Mechanical Design Guidelines; EPM-KEE-GL-000001, Electrical Design Guidelines; EPM-KEE-GL-000004, ELV Engineering Guideline; and EPM-KEA-GL-000001, Architectural Design Guidelines.

2.0 SCOPE

1. Power saving techniques covered by this document includes the following;
   - HVAC design that should be improved, changed, or superseded due to technological advancement.
   - Improvements in existing buildings field devices, controls, and system “Sequence of Operation” to reduce power consumption.
   - Selecting the proper HVAC equipment and system ancillaries, including their proper placement in the system distribution.
   - Improvement in architectural design to reduce HVAC and lighting loads.
   - Providing efficient mechanical and electrical equipment.
   - Commissioning activities that affects power and water consumption which needs strict implementation during construction phase.
   - Operation and Maintenance (O&M) personnel responsibilities to maintain efficient power consumption of equipment and systems.
   - Integrated building design between architectural, mechanical, and electrical discipline.
   - Other power consumption reduction techniques.
2. Water savings technique covered by this document includes;
   - Innovations in plumbing fixtures to reduce water consumption.
   - Ways to reduce water consumption in irrigation system.
   - Providing water efficient equipment.
   - Use of TSE (Treated Sewer Effluent) in lieu of Domestic Water in various application.
   - Recovery of condensates and other water consumption reduction techniques.
   - Other water consumption reduction techniques.
3. The requirement of the Commissioning Authority (CxA) is not discussed in this document, although the entity takes an important role in providing a comprehensive and energy efficient Building Services System design. CxA service is mandatory by National Fire Protection Association (NFPA), American Society of Heating, Refrigerating, and Air-conditioning Engineers (ASHRAE), International Energy Conservation Code (IECC), BS Standards, Chartered Institution of Building Services Engineers (CIBSE), Leadership in Energy and Environment Design (LEED), and International Building Code (IBC) to develop the Owner Project Requirements (OPR) with the client before the design stage. Refer to document EPM-KT0-GL-000003, Project Testing and Commissioning Guideline for complete scope of the CxA in the Commissioning Process.
4.

3.0 DEFINITIONS

Abbreviations	Description
A/C	Air Conditioning
ADP	Apparatus Dew Point
A/C	Air Conditioning
ACH	Air Change Hour
AFD	Adjustable Frequency Drive
AHU	Air Handling Unit
AMCA	Air Movement and Control Association
ASHRAE	American Society of Heating, Refrigerating, and Air-conditioning Engineers
ASD	Adjustable Speed Drive
BEP	Best Efficiency Point
BMS	Building Management System
CBECS	Commercial Building Energy Consumption Survey
CDT/HDT	Cold Deck Temperature/ Hot Deck Temperature
CHW	Chilled Water
CIBSE	Chartered Institution of Building Services Engineers
COP	Coefficient of Performance
CT	Cooling Tower
CWST	Chilled Water Supply Temperature
CWT	Condenser Water Temperature
CxA	Commissioning Authority
DCV	Demand Control Ventilation
DHT	Daylight Harvesting Technique
DOAS	Dedicated Outdoor Air System
DPS/T	Differential Pressure Sensor/Transmitter
DP	Differential Pressure
DRV	Double Regulating Valve
DSP	Duct Static Pressure
Dx Unit	Direct Expansion Unit
ECB	Energy Cost Budget
EER	Energy Efficiency Ratio
ERW	Energy Recovery Wheel
ESP	External Static Pressure
ECM	Electronically Commuted Motors
ETS	Energy Transfer Station
FCU	Fan Coil Unit
FLS	Fire and Life Safety System
FSC	Fixed Speed Chiller
FSD	Fixed Speed Drive
HRC	Heat Recovery Chiller
HVAC	Heating, Ventilation, and Air Conditioning
IPLV / NPLV	Integrated Part Load Value / Non-Integrated Part Load Value
IAQ	Indoor Air Quality
IBC	International Building Code
IECC	International Energy Conservation Code
LED	Light Emitting Diode
LEED	Leadership in Energy and Environment Design
LNG	Liquefied Natural Gas
LPF	Liter per Flush
LPG	Liquefied Petroleum Gas
LSG Ratio	Light to Solar Gain Ratio
LDAC	Liquid Desiccant Air Conditioning
MAT	Mixed Air Temperature
MIL	Magnetic Induction Lighting

Abbreviations	Description
NFPA	National Fire Protection Association
OA-AHU	Outdoor Air Handling Unit
O&M	Operation and Maintenance
OPR	Owners Project Requirement
PID	Proportional Integral Derivative
P/S	Primary/Secondary
PICV	Pressure Independent Control Valve
PRC	Performance Rate Criteria
RH	Relative Humidity
RHG	Relative Heat Gain
SC	Shading Coefficient
SHGC	Solar Heat Gain Coefficient
THD	Total Harmonic Distortion
TSE	Treated Sewer Effluent
US CBECS	United States Commercial Building Energy Consumption Survey
UV	Ultra Violet
VAV	Variable Air Volume
VFD	Variable Frequency Drive
VOC	Volatile Organic Compound
VSC	Variable Speed Chiller
VSD	Variable Speed Driver
VRV	Variable Refrigerant Volume
VT	Visible Transmittance
WWTP	Waste Water Treatment Plant

4.0 RESPONSIBILITIES

This document will be owned and maintained by EXPRO Engineering. Entities are responsible to provide this guideline to A/E's responsible for the design of the Building Services System.

5.0 INTRODUCTION

HVAC consumes about 60% to 70% of the total building power consumption in the Middle East due to large solar heat load penetrating the building façade and fenestration; and cooling load needed to treat or cool outdoor air for occupant ventilation purpose. Cooling load for outdoor air is compounded by high relative humidity especially in those regions close to the sea. Since the HVAC is the prime consumer of building electrical power, the main concentration of this document focuses on techniques in reducing the required power either by energy recovery, selecting the proper HVAC system, using advance field devices, providing proper algorithm and Sequence of Operation, increasing the building thermal mass, utilizing HVAC parameters for increasing equipment efficiency, and proper placement of system ancillaries.

As important as power reduction, water wastage is also the main focus of the document. Techniques to reduce water wastage is the secondary focus of the document to reduce operating cost of Waste Water Treatment Plants, thereby reducing cost of delivering clean water to consumers. Reducing water wastage also results in reduction of power for pumps to deliver water to occupants or consumers.

Techniques discuss in this document are those not commonly used in projects due to a poor understanding of designers and engineers of current available technological advancement, "unfamiliarity" in problems associated with HVAC systems and controls, repeated poor site practices, and other practices during Testing and Commissioning (T&C) which leads to inferior system performance and increased power consumption. Energy Standards such as ASHRAE 90.1 which requires minimum compliance is discussed in the later part of the document. This document does not intend to concentrate in the numerical rating and efficiency data since it changes over time, but concentrate on design techniques which play the most important part in energy and water optimization. The highest Energy Efficiency Ratio (EER) for direct expansion Air Handling Unit (AHU) in the market today (which is 11.7) will not be true tomorrow due to technological advancement fueled by tight competition in the market.

6.0 POWER SAVING TECHNIQUES

6.1 Use of Pressure Independent Control Valve (PICV) for Hydronic Recirculating System

PICV (Pressure Independent Control Valve) is a two-way control valve which combines function of a control valve, a differential pressure regulating valve, and an auto-balancing valve. Differential pressure regulation is required for a recirculating hydronic circuit since it greatly affects the operation of the system resulting in power and efficiency degradation. Consider the hydraulic gradient of a primary-secondary chilled water system secondary pumping distribution using a conventional two-way control valve (with no differential pressure regulator), a manual balancing valve, and the DPS/T (Differential Pressure Sensor/Transmitter) installed in the following locations;

1. ***DPS/T in the Index Point- see Fig. 1***

   Sensor location measures the pressure drop from the supply side to the return side where it is tapped. The pressure drop includes drop in the piping and accessories like control valve, strainer, isolation valve, balancing valve, and cooling coil. This pressure drop measured by the DPS/T is sent to the Building Management System (BMS) controller and then to the Variable Frequency Drive (VFD) controller of the chilled water pump. The data represent the lowest pressure setting where the VFD controller can reduce the pump speed based on the required flowrate.

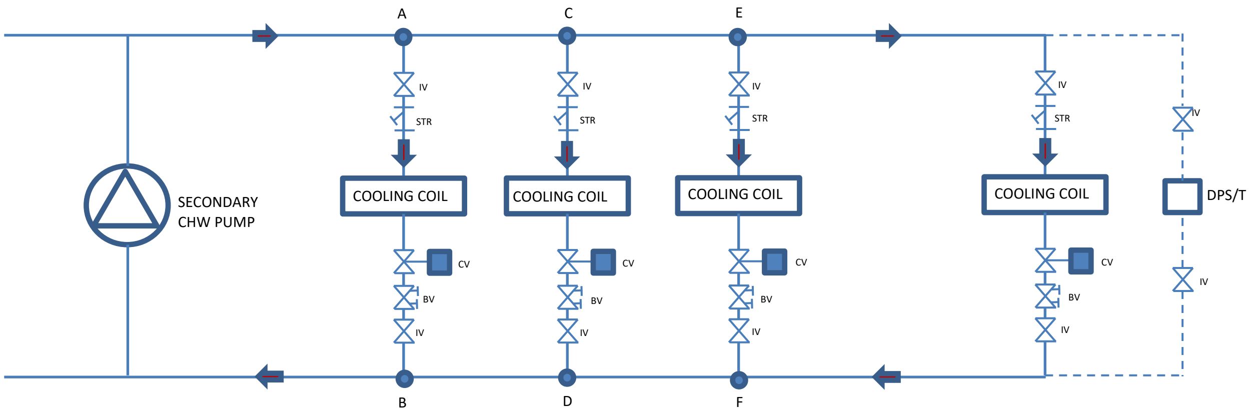

Fig. 1 - Secondary Loop of P/S CHW Distribution System with DPS/T in the Index

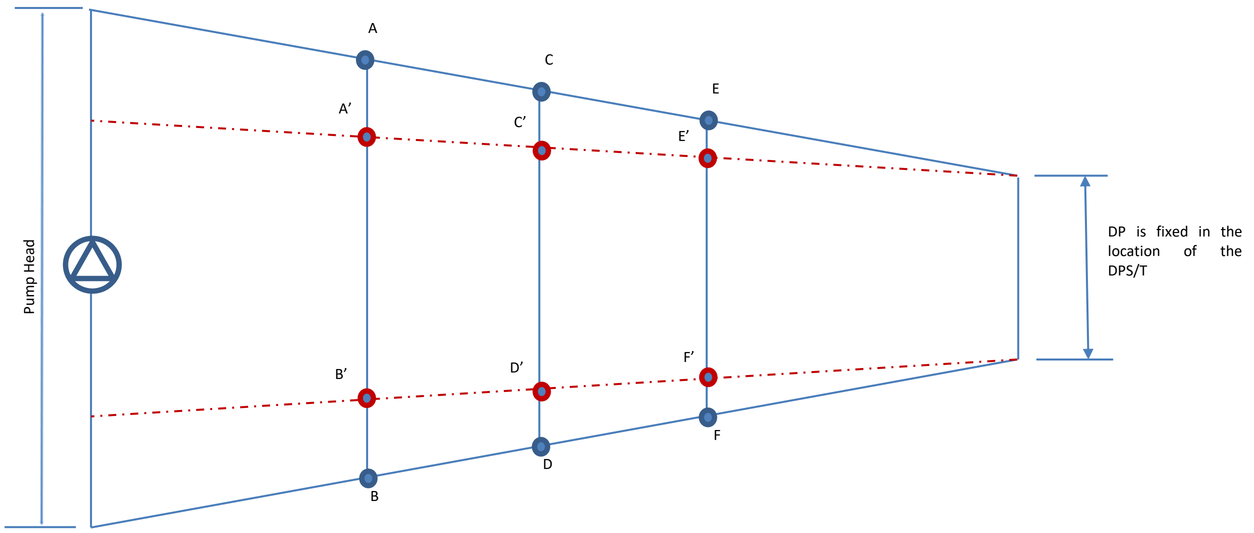

Fig. 2 - Hydraulic Gradient of the Secondary Loop above

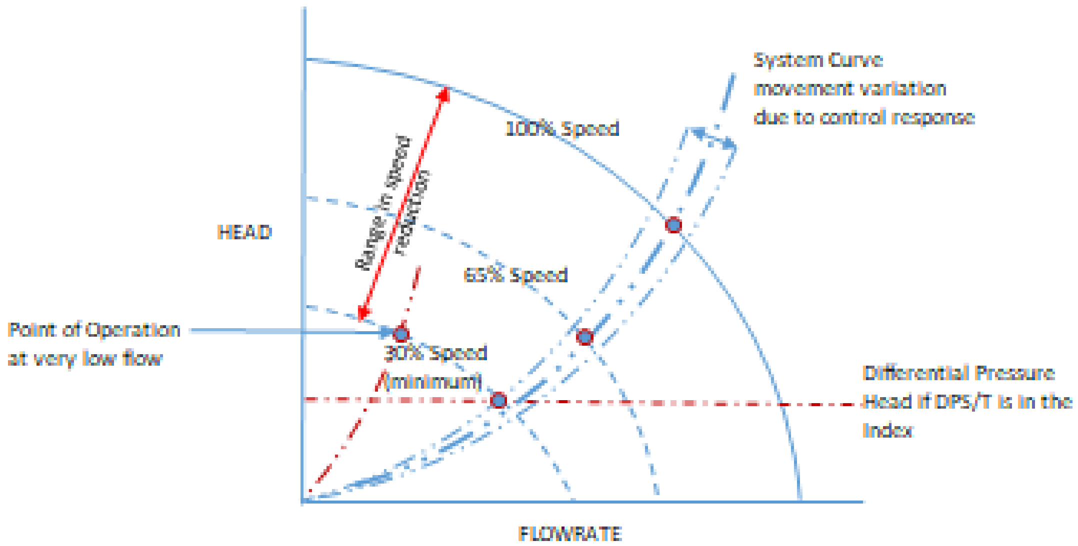

Fig. 3 - CHW Pump Performance when DPS/T is in the Index

Reference to Fig. 2, the hydraulic gradient represent the Secondary Piping Distribution and the Secondary Chilled Water (CHW) pump head. The pump discharge represent the highest available head and the slope indicates pressure drop due to piping friction as water travels to the index circuit and back to the pump. The DPS/T is located at the Index Point of the circuit and usually set at 0.8 to 1 bar pressure head. Line A-B, C-D, E-F represent each terminal circuit when the manual balancing valve is set at the balance position after the proportional balancing. Line A'-B', C'-D', E'-F' represent each terminal when the CHW Pump reduces its speed due to reduction of cooling load. Point A-A', B-B', C-C', D-D', E-E', F-F' represent the reduction of pressure differential in each terminal branch during pump speed reduction which result in lower water flowrate. It can be noticed that the reduction in differential pressure is higher as the terminal circuit is closer to the pump so that the CHW starvation is greater in branches closer to the pump. It is evident that every time the pump reduce its speed, the result is water starvation in all the terminal circuit when the DPS/T is at the index. It is for this reason that terminal branches control valve close to the pump must be at partially closed position during proportional balancing. Although partially closing the control valve during balancing will help in resolving reduced pressure differential during reduction in pump speed (since there will be allowance for the valve to open), problem about under flow cannot be resolved totally since most of the conventional control valve are only effective from 70% to 100% of the closed position for flow control. It is not at all possible to determine the right position of closing the valve during proportional balancing to accurately compensate for reduced pressure differential during actual operation of the system. Site practice is that control valves close to the pump (about 2/3 of the circuit from the pump) are set at partially close position during proportional balancing. In principle, conventional control valve in the index point (and all that closer to it) must be full or nearly at full open position so that the pressure drop is at the lowest but this condition will result in low Valve Authority. For example, at the design flow, a 20mm. diameter ball valve or butterfly control valve has a pressure drop of 70 pascals at fully open position while the rest of the circuit pressure loss is 40,000 pa (25,000 pa for the coil and 15,000 pa for the rest of piping accessories). The valve authority (VA) will be almost 0 which means there is no means of flow control until the valve closes resulting in at least 0.25 VA or all control valve be replaced by high pressure drop type (low Cv or Kv value). The Index setting is therefore increase to 0.8bar (or sometimes up to 1 bar) resulting in higher pumping power. This is the main advantage of PICV which minimum pressure loss is only 0.016 bar (16,000 pa) to attain the required flow, therefore DPS/T setting can be as low as 0.6 bar resulting in power savings.

Reference to Fig.3, which shows the performance of the pump based on lower setting of DPS/T since it is located in the index circuit. It is favorable in terms of power consumption since the pump speed can be reduced up to the minimum speed (usually 25 to 30% of the maximum speed), and the system curve stays in-line with the pump power curve (line of highest efficiency at any point of speed reduction) but unideal since this set-up will create chilled water starvation in all the terminal branch, which results in under cooling.

As a summary, locating the DPS/T in the index has the following disadvantage;

a. It creates underflow or starving in all terminal branches and cannot satisfy cooling demand during partial loading.

Locating the DPS/T in the index has the following advantage;

a. Efficient control operation since range of speed change is maximized which results in power savings.
b. Efficient pump performance since system curve stay close to power curve.

2. ***DPS/T located across the CHW Pump – see Fig.4***

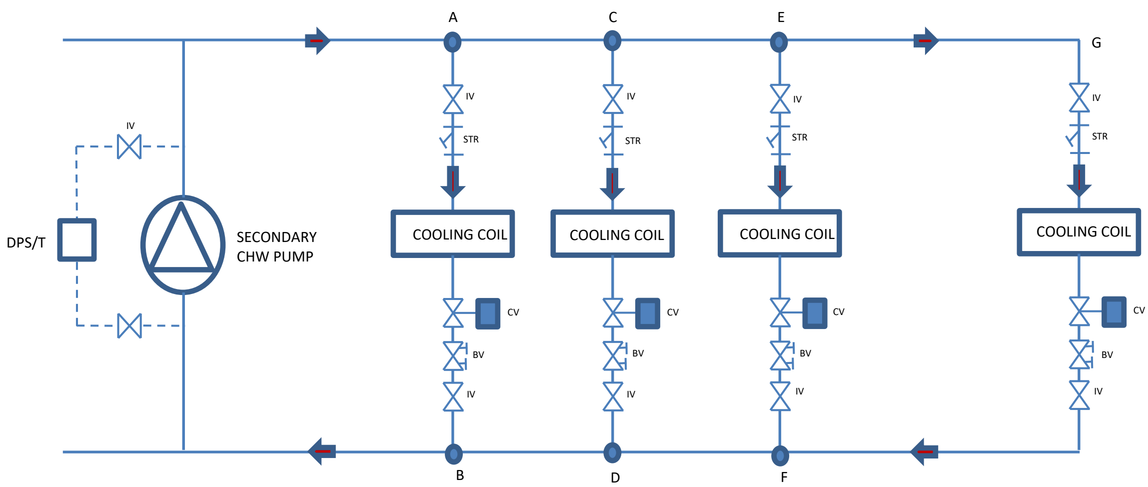

Fig. 4 - Secondary Loop of P/S CHW Distribution System with DPS/T across the pump

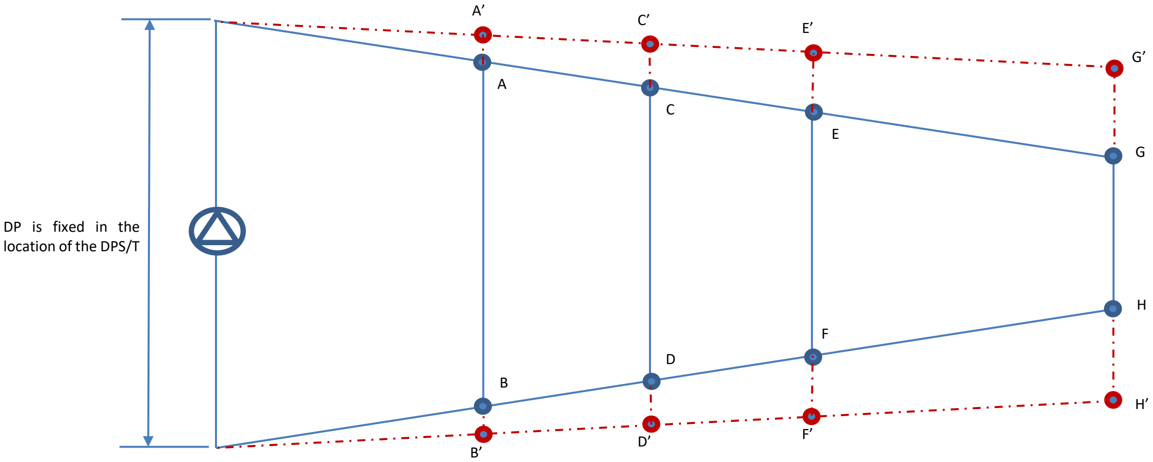

Fig. 5 - Hydraulic Gradient when DPS/T is located across the pump

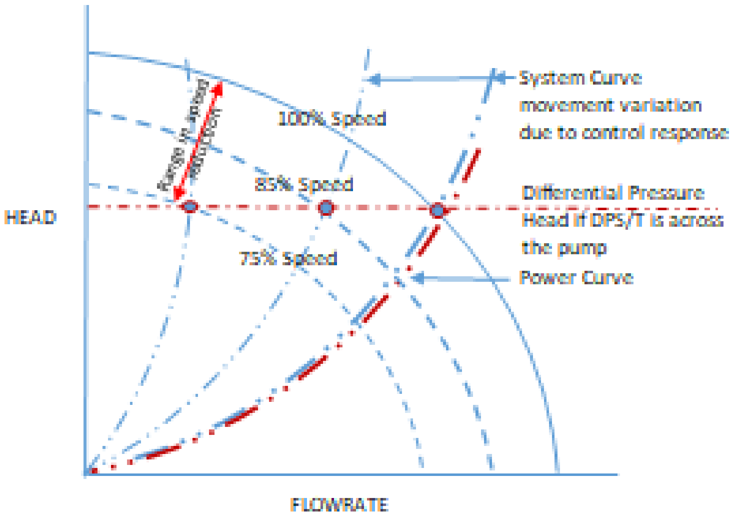

Fig. 6 - CHW Pump Performance when DPS/T is located across the pump

Reference to Fig.5 when the DPS/T is located across the CHW Pump, the setting is equivalent to the pressure loss from the discharge of the pump going to the index point and back to the suction of the pump. In other words, the setting of the DPS/T is equal to the head of the pump. Since the setting is fixed across the pump, any change of speed due to load demand reduction will create excessive pressure differential across each terminal circuit. The control valve in each terminal can handle the increase in differential pressure provided that it is selected for the correct differential pressure and valve authority, which is always not the case resulting in overflow.

Reference to Fig.6, the DPS/T is fixed and the setting is equal to the head of the pump. The pump System Curve shifts to the right of the BEP (Best Efficiency Point) and away from the Power Curve. Speed reduction is limited to that point that can intersect to the set pressure (differential pressure head). At this set-up, speed reduction is limited only to appx. 75% of the total speed and there is a tremendous increase of radial thrust which can damage the impeller since the operation of the pump is away from the BEP. Efficiency of the pump is also worst at lower cooling demand.

As a summary, locating the DPS/T across the pump has the following disadvantage;

a. Overflow specially for all branches close to the pump due to excessive pressure differential.

b. High impeller thrust load, vibration, and noise especially in very low cooling demand which results in premature damage of the pump impeller and bearings.

c. Inefficient operation since system curve operates far to the left of the Best Efficiency Point and consumes excessive electrical power per delivered capacity.

d. Opportunity for power savings is very poor since range of speed reduction is only narrow. Lower speed is about 75% to 80% of the maximum speed.

e. Increase in initial capital expenses due to higher differential pressure requirements of control valves.

As a summary, locating the DPS/T across the pump has the following advantage;

a. Solves the problem with underflow and satisfy or exceeds cooling demand for all branches in the circuit.

3. ***DPS/T located midway of the circuit – see Fig. 7***

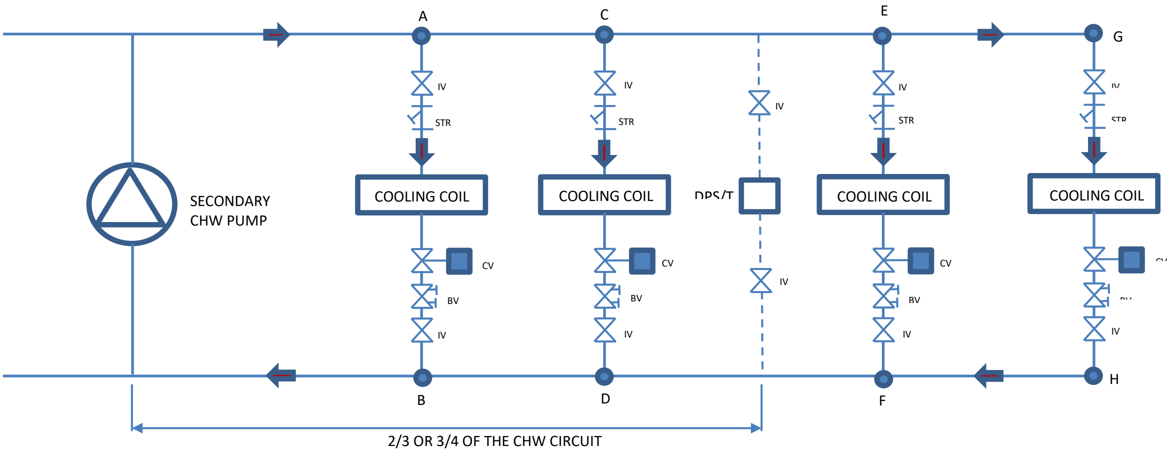

Fig. 7 - Secondary Loop of P/S CHW Distribution System with DPS/T midway the circuit

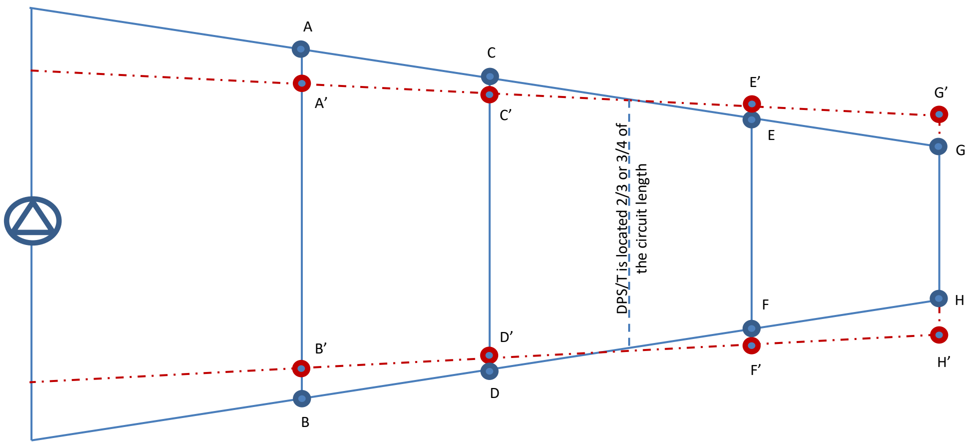

Fig. 8 - Hydraulic Gradient when DPS/T is located midway of the circuit

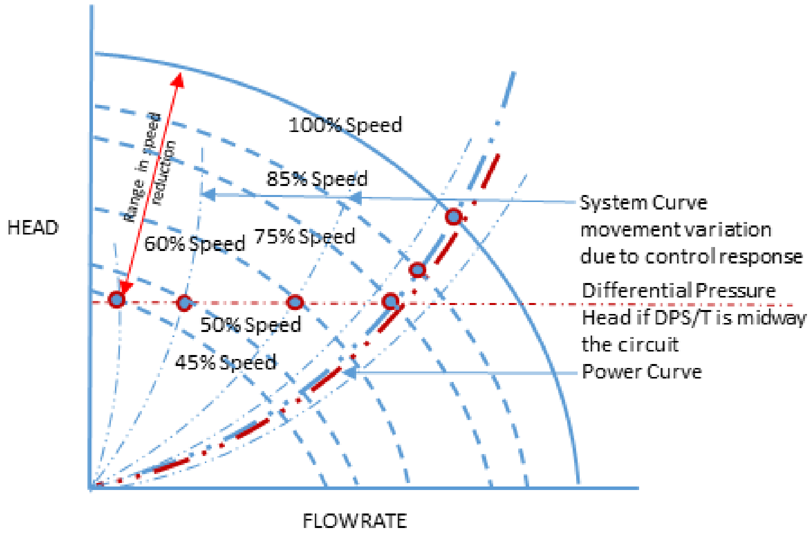

Fig. 9 - CHW Pump Performance when DPS/T is located across the pump

Locating the DPS/T midway of the circuit is a compromise between the other two previous techniques and it inherit the advantages and disadvantages of both systems. As seen in Fig.8, upstream branches from the location of the DPS/T will suffer from underflow or starving which results in undercooling. Reference to Fig. 9, at certain cooling demand (about 75% of the speed), the system curve is maintained at BEP (Best Efficiency Point). At the pressure head equal to the setting of the DPS/T, the system curve move away from the BEP and Power Curve resulting in inefficient operation (decreased efficiency) and increase in “thrust load” which can damage the impeller.

As a summary, locating the DPS/T midway the circuit has the following disadvantage;

a. High impeller thrust load, vibration, and noise especially in low cooling demand which results in premature damage of the pump impeller and bearings. Thrust Load is lower compare to the DPS/T located across the CHW Pump.

b. Inefficient operation in low cooling load (below 75% of the speed) since system curve operates far to the left of the Best Efficiency Point and consumes excessive electrical power.

c. Opportunity for power savings is very poor since range of speed reduction is narrower than DPS/T located in the Index Point. Lower speed is about 45% to 50% of the maximum speed.

As the reader can see, there is no ideal system using the conventional two (2) way control valve and the problem with decreasing and increasing differential pressure must be resolved so that the DPS/T can be located in the Index Point to maximize system efficiency and power savings.

PICV (Pressure Independent Control Valve) was originally created to resolve the issues as stated above since the valve is capable to regulate variation in differential pressure due to changes in pump speed (due to cooling demand change). It is also design to automatically balance the water flow to simplify balancing. Providing the PICV also resolves a common problem encountered in chilled water system which is unknown to many HVAC Engineers resulting in tremendous waste of energy. It was only “serendipity” or unexpected that the Low Delta was also resolved using PICV.

What is Low Delta T?

Low Delta T is a phenomena in chilled water system or any recirculating hydronic system where the design differential temperature is not attained. Returning water temperature to the chillers is lower than the design temperature. There are ASHRAE and other technical articles that were previously released to explain the cause of this phenomena. Most explanation relates the phenomena to improperly flushed or uncleaned system, wrong coil and other appurtenances selection, undersize piping and equipment, and wrong valve authority; but there are no sufficient proof. The writer of this document, being involved extensively in Testing and Commissioning of many projects experienced problems with Low Delta T and although the above stated reasons for cause of

Low Delta T are complied, the Low Delta T Syndrome still exist. All projects using conventional two-way valve will suffer the same problem and no one really knows the reason for Low Delta T.

Low Delta T created by three (3) way control valve is not discussed in this book since this type of valve should not exist in modern world especially for chilled water application. Chilled water that is by-pass by the 3-way valve to the AHU during partial cooling demand is mixed with the AHU out-going water resulting in lower return water temperature to the chiller. The effect regarding chiller performance is the same as explained below.

What problems are created by Low Delta T?

a. Since Delta T is the reference for determining the required water flow from a given cooling or heating load, and the flow is the reference for sizing the distribution piping and cooling/heating coils, Low Delta T will require increased pumping capacity to match the cooling/heating demand. Increased pumping capacity will increase system resistance therefore aggravate power consumption further. HVAC Designers who are well aware of the problem will resort in increasing distribution pipe size and coil size to compensate for this phenomena, thus increasing the Project Capital Expenses.

b. Delta T is the reference for all chiller capacity control since it is the representation of the building cooling load in terms of chilled water return temperature. Obtaining the correct delta T is highly important for the chiller to reach its full capacity. Low Delta T will force chillers to run at partial load even there is sufficient load to run at full capacity. In real life condition, a project chiller will reach only up to 70% (higher for water cooled condenser) of its full capacity due to Low Delta T during high cooling load demand and lower (even as low as 40%) during low load condition. Most of the time, chillers will run in the range of 50% to 60% of its full capacity. HVAC Designers who is very well aware of this phenomena, are force to increase the calculated cooling load by 25% to 30% to satisfy the cooling demand, thus increasing the Project Capital Expenses since chillers are one of the most expensive equipment in the building. Power consumption is further aggravated by the fact that fixed speed chillers are very inefficient in part load, unlike Variable Speed Driver (VSD) Chillers which are efficient during part load (from below the full capacity up to 60% as some chiller manufacturer have reported).

It is reported that from 20% up to 30% of the building energy is wasted due to the above reasons related to Low Delta T. PICV eliminates low Delta T resulting in tremendous power savings.

What other problems are resolved by PICV?

a. Conventional control valves in correlation to the branch terminal characteristic always results in lower Valve Authority. Lower Valve Authority means that the valve is unable to control or regulate the flow within its full stroke. Valve seat movement from fully open up to partially close (say 60%) has no or little effect on the flow so as to the coil heat exchange. PICV, a truly equal percentage valve with Valve Authority close or equal to unity, when correlated to the hydraulic performance of the coil, valves, and piping produces a true linear characteristic in terms of stroke, flow, and heat exchange. This means that every movement or stroke of the valve has an effect in water flow as well as the coil heat exchange providing accurate and efficient space temperature control.

b. Conventional control valves has no ability for pressure differential regulation. Each time a valve throttles the flow of a terminal branch, the reduced flow from the supply side will be diverted to the nearest branches of lowest resistance, which in result will throttle to maintain the required flow. This phenomena will continue resulting in restless movement of each valve actuator and compounded by the change of pump speed due to change in cooling demand. This phenomena is the suspected reason for Low Delta T since when the PICV has arrested the changing differential pressure, the Low Delta T was also resolved.

   PICV has an internal pressure regulator, the adjustment due to changing flow from other branches and system demands are not any more reliant in the valve actuator. The valve actuator only moves in response to the variation in cooling demand of the room it serves. The life of actuators and BMS controllers are expanded since its life highly depends on cycles of actuator response.

c. Using PICV, the hydronic system now follows the law of heat exchange and temperature monitoring can now be used against flow monitoring for chiller loading and unloading. This simplify the BMS programming and reduce flow meters required for the chilled water installation.

There is no reason why PICV should not be used in any project, either existing or being constructed. Cost of PICV is about 1.5 times the cost of conventional control valves (reference to the author's previous project using Flowcon PICV) but is offset by the cost due to elimination of DRV, reduction in chiller quantity, smaller CHW pumps, lower cost for T&C balancing, and lower power consumption. Designers needs to be aware of the advantages using PICV since conventional 2-way valves are still prevalently used in building chilled water system design.

If PICV will be utilized for existing and operating buildings with fixed speed chillers, reduction of power consumption from 20% to 30% can be attained and there will be surplus in the capacity of chillers that can be used for future expansion (refer to Fig.10 below from Siemens case study).

In addition, PICVs have a pre-setting function that provides even finer temperature control accuracy, further eliminating temperature fluctuations and discomfort. As a result, occupants are less likely to raise or lower temperature settings, adding to the overall energy savings that the valves generate. PICVs also allow for advanced pump control strategies that reduce energy use even more. In total, PICVs can generate energy savings of up to 30%. They can be used in almost any heating and cooling application to provide year-round comfort for building occupants.

This paper discusses the energy savings methods in detail and includes a case study example that quantifies them.

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Fig. 10 – PICV case study from Siemens

Information as stated above with regards to the correction of Delta T is based on actual experience of the author reference to his last three (3) projects in Oman (Asian Beach Games and Muttawar Tower) and in KSA (King Faisal Specialist Hospital in Riyadh).

6.2 Multiple DPS/T in Prospective Indexes and Providing the Optimum Setting

As explained in the previous section, locating the DPS/T in the Index Point provides opportunity for power savings since variation in pump speed range is maximized. Common setting for the DPS/T ranges from 0.7 bar to 1.0 bar (for non-PICV application) depending mainly in the pressure drop across the cooling coil (fewer row coil and wider Delta T has less pressure drop). The lower the differential pressure setting of the DPS/T, the higher the opportunity for power savings especially for larger systems. Refer to section 6.1 regarding power savings with the use of PICV when the DPS/T is in the Index Point.

For large recirculating system with multiple sub-main branches, locating one (1) DPS/T in an index point cannot represent the load for other sub-main branches which leads to excessive pressure absorbed by the control valves when the branch in the initial index point is at partial flow (or partial load). The initial index point was calculated based on assumption that all branches are at the design flowrate which is not the case during the actual operation, thus multiple DPS/T must be installed in end branches that can possibly become a new index point when all other branches are at partial load. Control valves in the index point must always be at fully open position for maximizing power consumption. The BMS must be program in a way to determine the maximum and minimum differential pressure in each index branch to ensure that overflow and underflow are avoided as well as to locate the actual index point during part load.

6.3 Primary (P) against Primary-Secondary (P/S) Chilled Water System

In chilled water distribution system, there are two (2) prevalent design that can be seen nowadays. Most of the existing system are Primary/Secondary System, and Primary System is now gaining popularity against the P/S for first cost and energy consumption reasons. From Figure 11 and 12 below, Primary System and Primary/Secondary System are illustrated;

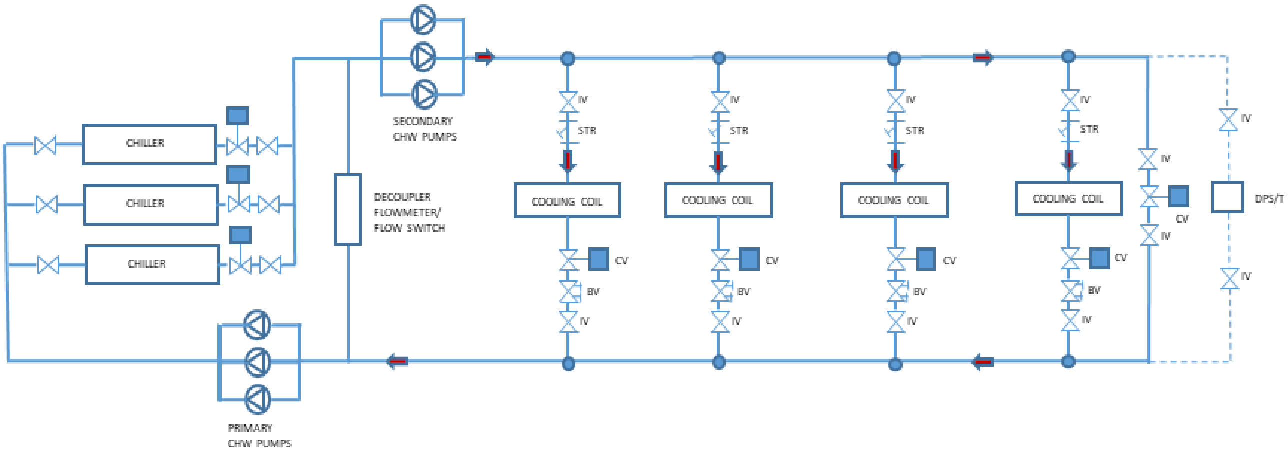


Fig. 11 - Primary-Secondary Chilled Water System

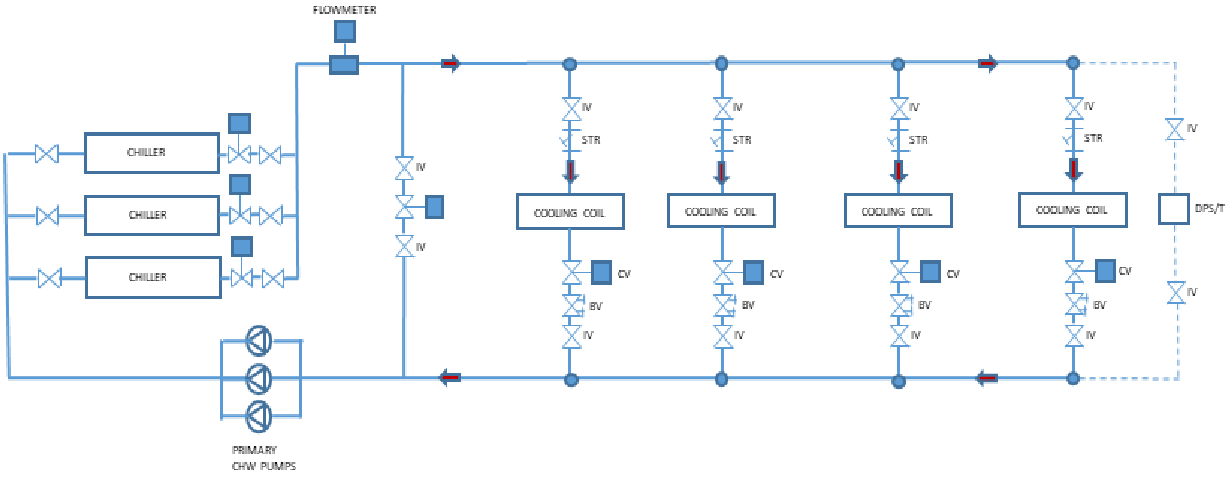


Fig. 12 - Primary Chilled Water System

Using the Primary Chilled Water System against Primary/Secondary System reduce building power consumption by 3%-8% since it only requires primary pumps to operate. Initial cost of investment is also lower compare to P/S System. P/S only gains popularity against the Primary System in the past since chiller controllers are not that sophisticated and fast that chiller manufacturers do not agree to use Primary Chilled Water System. Moreover about 10 years ago, control valve differential pressure were only designed for maximum 2.0 bar pressure due to limitation in design and materials. Nowadays, chiller controllers are very sophisticated and fast responding that manufacturer's recommends to implement Primary System for the reason of energy and cost savings. Control valves with 4.0 bar differential pressure are readily available in the market and owners can even request special design which can reach up to 8 bar differential pressure and higher. Today, there is no real reason for HVAC designers to incline with Primary/Secondary System. Sad to say, almost all chilled water system are still

designed for P/S System instead of Primary System because of misinformation of the reasons why P/S is used against Primary System.

See extract from David Brooks, HVAC Expert of facilitiesnet, Building and Operating Management & Facilities Maintenance Division regarding power savings.

> • Potential to save pumping energy. This is arguably the greatest benefit to variable only pumping. Daily operational costs can be minimized by moving only the necessary amount of water required for the chiller or boiler load demand. Savings typically fall in the single digits, between 3 percent and 8 percent of the daily energy operating budget, compared to constant primary/variable secondary pumping systems.

It is also an interesting fact that chilled water pumps are oversized from previous designs because of the problem with Low Delta T. If PICV will be implemented in existing and operating buildings (with modern chillers capable of Primary System operation) and the Delta T is corrected, lower flow will be required which in turn reduce the head required by the pumps. It is therefore possible that the secondary pumps of existing projects can handle the required flow and head when PICV will be used, so that the primary pumps can be removed. Savings from 25% to 35% of the building power consumption can be attained. Chiller capacity will be excessive that it can be used for project expansion or other projects.

6.4 Increasing the Building Thermal Mass

Increasing the mass of the building by providing thicker external walls and massive roof slabs reduce the required cooling for the building due to heat storage and thermal time lagging. Consider a thin wall (5 inches thick wall) against a double wall (10 inches thick wall with sandwich insulation) made of same material so that the U-value (thermal coefficient) will be the same. The thin wall has a thermal time lag of 4 hrs. and the thick wall has 10 hrs. thermal time lag based on 46°C external temperature and 24°C indoor temperature. Thermal time lagging is the time for peak solar heat to reach the internal surface of the wall affecting the room thermal load. The higher the thermal difference between the outer surfaces to the internal surface decreases the thermal time lag.

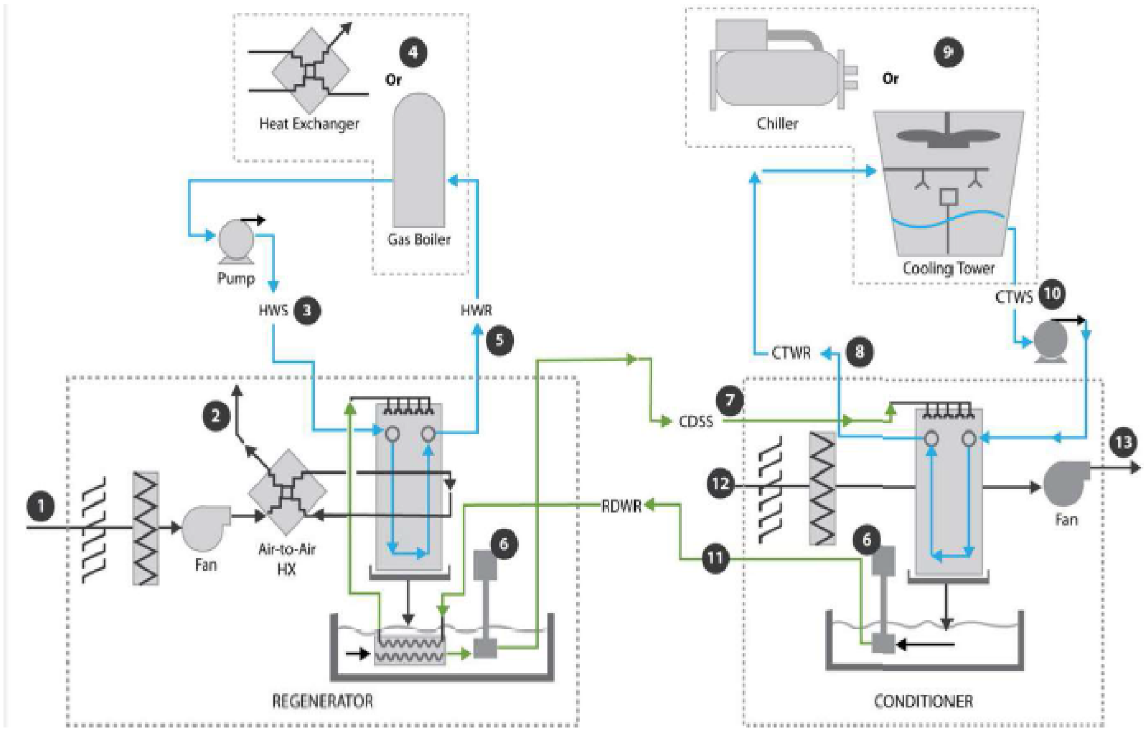
An east wall received peak solar strike at 10am and using thin wall as describe above, the majority of the load will reach the internal surface by 2pm and thus increase the cooling load of the room. Some of the total heat which strikes the external wall and absorbed will be traveling back to the external surface and convected back to the ambient air by the time the sun shifts to the west side of the building, thus providing shade and cooler surface to the east wall. If the east wall is a thick wall, then the room will be affected by appx. 8pm. At 7pm, the sun is already set so that there is a significant decrease in ambient temperature that most of the heat absorbed by the external wall travels back to the external surface and convected back to the ambient air. Only a portion of the total heat will reach the internal surface, thus increasing the room cooling load marginally if the building is intended to use for 24 hrs. If the function of the building is an office building, the Air Conditioning (A/C) is shut down by 6pm so that the heat load from the external wall reaching the room has minimal effect in HVAC load and power consumption.

Reduction in power consumption of HVAC system depends on the function of the building as well as the orientation of the building walls to the south, west, and east. An office building and other functions that operates 10 hrs. a day has greater energy savings compare to a 24-hrs a day operating building. A rectangular building whose longer side faces the north has lesser power consumption than if the shorter wall faces the north.

6.5 Liquid Desiccant Air Conditioning (LDAC) System

LDAC System uses the natural tendency of salt solution to absorb moisture in the air and rejecting (or desorbing) the moisture by heating the solution. To increase the latent cooling capacity of the conditioner, the salt solution is sprayed in a chamber (a heat sink) to increase air to salt solution contact surface. Heat released exothermally by the solution is carried away by evaporative cooling system or Direct Expansion System. Air Psychrometry depends on the concentration of salt solution in contact with supply air and the flow of salt solution. To increase the rejection of absorbed moisture in the regenerator, the solution is again sprayed in a chamber inside the regenerator to increase heated-air to salt-solution contact. Scavenge air is heated by means of boiler, solar heater, or other means of waste heat. LDAC is intended to be used for high latent load application (such as high %RH outdoor air treatment) or application that requires low %RH. As it can be seen, LDAC comprises of too many parts and accessories compare to Dx System so that the equipment first cost and maintenance is higher. Savings in power consumption can reach up to 40% compare to Dx System when solar heating in the regenerator and evaporative cooling for the conditioner is utilized, but will increase the Capital Expense even more. Savings in first cost can be utilized using waste heat from other source instead of solar heating. Using Dx System to retrieve heat dissipated from the conditioner will further reduce power savings by half. Main source of power savings for LDAC is that it eliminates the heat added to the Dx System by compression.

See Fig. 13 below for the complete component of LDAC System.



LDAC components and fluid flow diagram.

Image adapted by NREL from image by Lowenstein et al., 2006

LDAC Components and Fluid Flows

Conditioner Side		Regenerator Side	
1	Outdoor air for desiccant regeneration (scavenging air)	8	Conditioner water return to cooling source (CTWR)
2	Hot, humid exhaust air exhausted to ambient	9	Cooling source for chilled water
3	Hot water supply to regenerator (HWS)	10	Chilled water supply to conditioner (CTWS)
4	Thermal source for hot water	11	Weak desiccant return to regenerator (RDWR)
5	Hot water return to thermal source (HWR)	12	Humid outdoor air for ventilation
6	Desiccant sump pump	13	Dry outdoor air for space ventilation (processed air)
7	Strong desiccant supply to conditioner (CDSS)	–	–

Fig. 13 - The Liquid Desiccant System

6.6 Properly Segregating Circuits – Reducing Power Consumption

The power consumption of a system with straight circuit will consume 30% to 40% more power compare if the same system is divided equally into two. For example, a building fresh air AHU is located at the roof which serves the 16-storey building using a single circuit as shown in Figure 14;

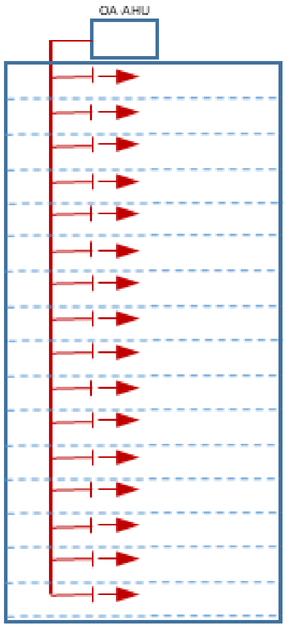


Fig. 14 OA-AHU with Single Circuit

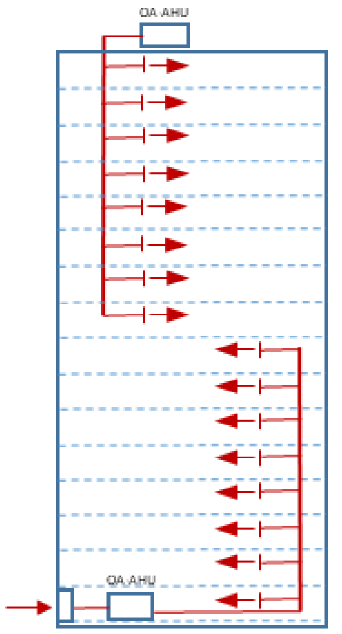


Fig. 15 OA-AHU with Single Circuit

If the total fresh air required is 10,000 L/s and the ESP (External Static Pressure) is 500 pascals, the power consumption for a 70% efficient fan-motor assembly will be 13.7 Kw (Internal Static loss is considered as 300 pascal). If the system is split into two as shown in Figure 15 so that the fresh air requirement is 5,000 L/s each and the ESP is 250 Pa, the power consumption for each AHU will be 4.3 Kw considering same fan-motor efficiency and internal static loss. The total power consumption for the two units is 8.6 Kw only with minor increase in Capital Expenses.

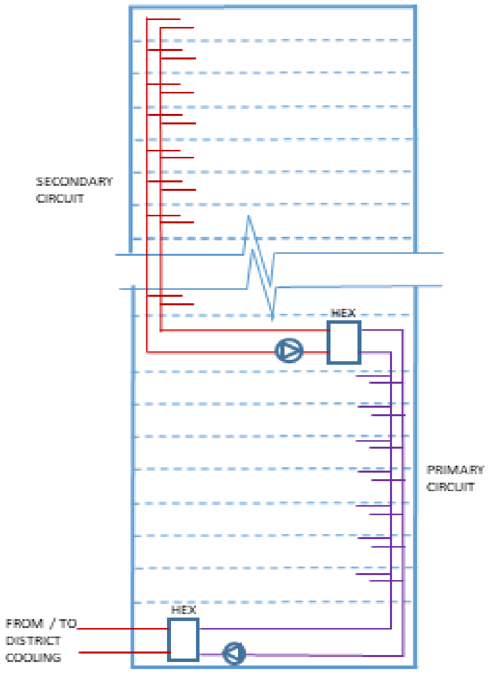


Fig. 16 Common Design for ETS in District Cooling

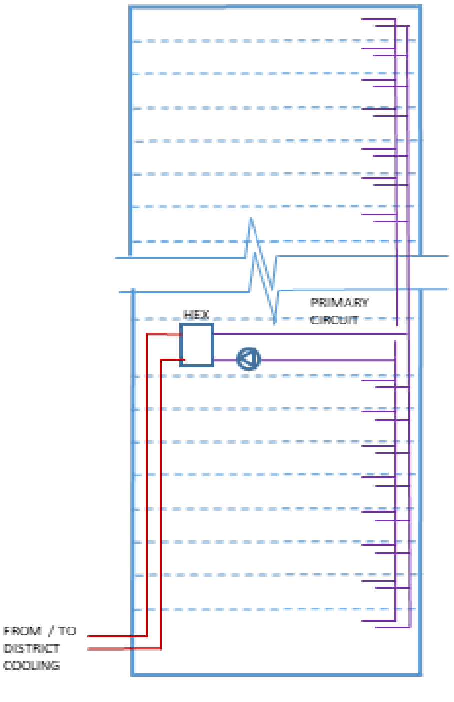


Fig. 17 ETS Midway of the Building

Now consider a common design for District Cooling System with the Energy Transfer Station (ETS) located in the Basement compare when the ETS is located mid-level of a 40-storey building as shown in Fig. 16. If the total flowrate of chilled water to the building is 400 L/s and the pump head for the primary circuit is 4 bar, the power required to run the pump for the primary circuit is 229 Kw considering 70% combined efficiency for pump and motor. If the secondary circuit requires half of the building load so that the secondary pump required flowrate is 200 L/s and 4 bar pump head, the power required is 115 Kw considering same pump-motor efficiency. The power required for both pumps will be 344 Kw. If the ETS is located midlevel of the building, the pump head will be calculated in one circuit only so that it will require same head as 4 bar, the power requirement is only 229 Kw using same pump-motor efficiency. There will be no additional power in the Central Station of the District Cooling if the building is not located in the Index Point (or the farthest branch). The static pressure build-up in Fig.17 is about 6.5 bar plus the total pump head of the pump (say 6 bar) so that the discharge pressure will be about 12.5 bar. The District Cooling equipment and CHW Distribution must be rated at 16 bar minimum.

Consider a single up-feed booster pumping system as shown in Figure 19 for a 20-storey building, which is the normal design applied for almost all buildings. If the water requirement is 50 L/s and the pump head required for the 20-storey building is 10.5 bar (7 bar for static head, 2 bar for friction loss, and 1.5 for residual pressure for plumbing fixture), the power requirement is 75 Kw (for 70% combined pump-motor efficiency). If the system is divided into two (2) zones as shown in Fig.20, the bottom circuit will require same flowrate which is 50 L/s. The head required in the bottom circuit is 6 bar (3.5 bar for static head, 1 bar for friction loss, and 1.5 for residual pressure). The power requirement for the bottom circuit is 43 Kw for the same pump-motor efficiency. The top zone will require 25 L/s and 6 bar pressure head which will result in 21.4 Kw, for a total 64.4 Kw. The 10.6 Kw savings is the power requirement of the motor but in actual operation, as in the case study made by Grundfos, Series Connected Booster System with Intermediate Tank will consume nearly half of the energy compared to "Single Booster Up-feed System" and "Transfer Pumping and Roof Tank System", and with the lowest Life-Cycle Cost. The main reason for the big energy savings is from the fact that the static head required to be overcome by the pump during minimum load (especially in the case of very less load) is less in Series Connected System than Single Up-feed Boosted System. For example, during the day (or late night) which is an off-peak, the flow requirement in the building is 1 L/s at the upper floor. Reference to the graph, the efficiency falls from 70% to 40% due to the shifting of the System Curve to the left of BEP (Best Efficiency Point) at 75% speed. The power required is 2.2 Kw for a Single Up-feed Boosted System (considering 8.8 bar head since Static and Residual Pressure is constant and 0.3 bar friction considered due to reduced flow). For a Series Connected- with Tank System, the power required for the same required flowrate is 1.1 Kw only (5.3 bar head only since Static Head is reduced by half and pump-motor efficiency will improve for smaller pump in lower load).

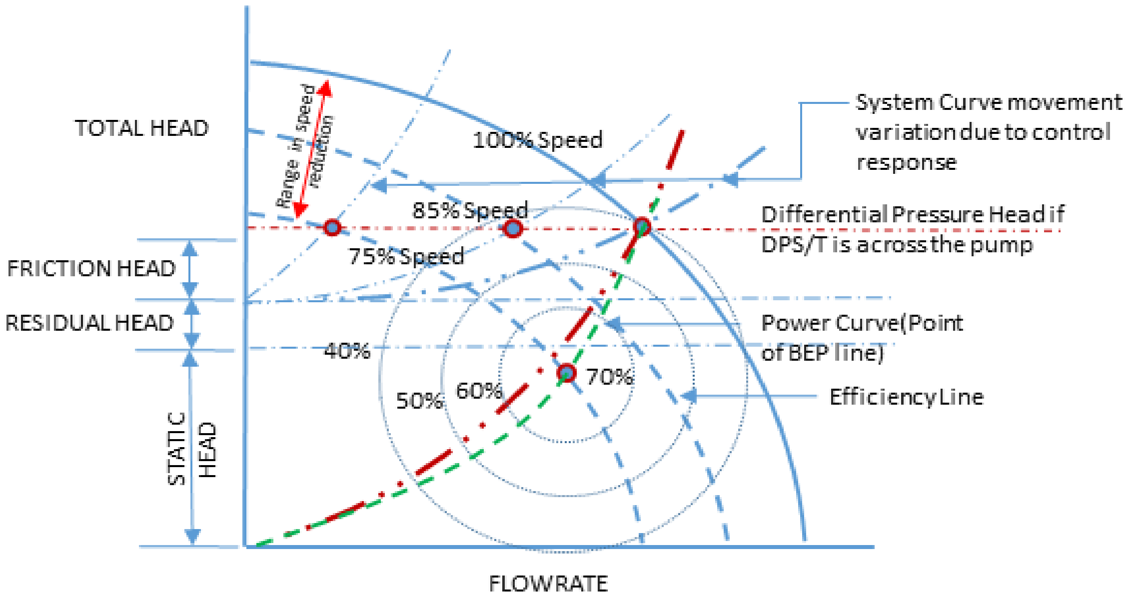


Fig. 18 - Operational Curve of Up-feed Domestic Water Boosting System

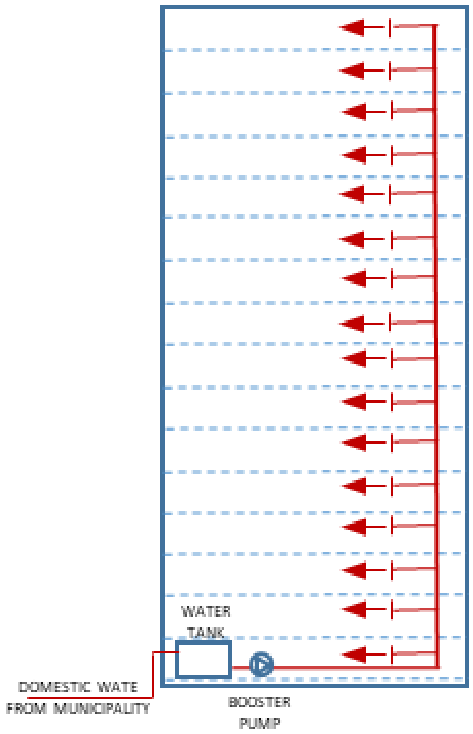


Fig. 19 - Single Up-feed Boosted Pumping System

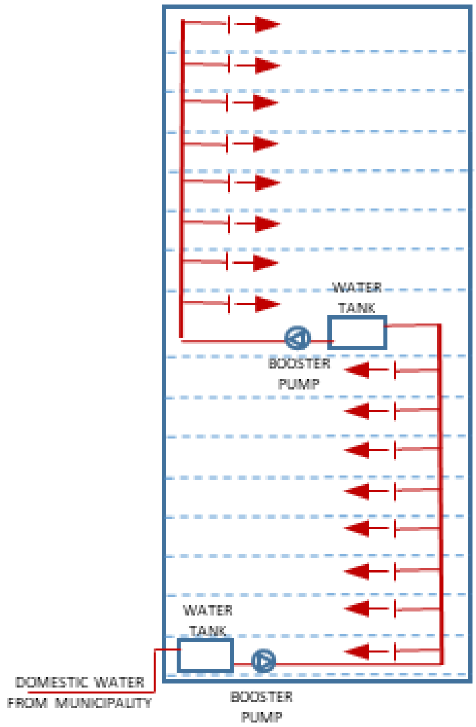


Fig. 20 - Series Connected System with Intermediate Break Tank

Since Domestic Water System is most of the time operating in partial load, therefore the energy consumed overtime is very less compare to single Up-feed Boosted System. Increase in Capital Expenses is minimal for such arrangement.

Note that Transfer Pumping System with Roof Tank and Down-feed Boosted System has the same power consumption as Single Up-feed System as per case study made by Grundfos, therefore this document does not intend to provide detailed discussion for such system. For Life Cycle Cost and power consumption details, refer to Figure 21 below for Grundfos case study extract. (Note that lost revenue is the business lost income over time due to space taken by additional tanks and pumps).

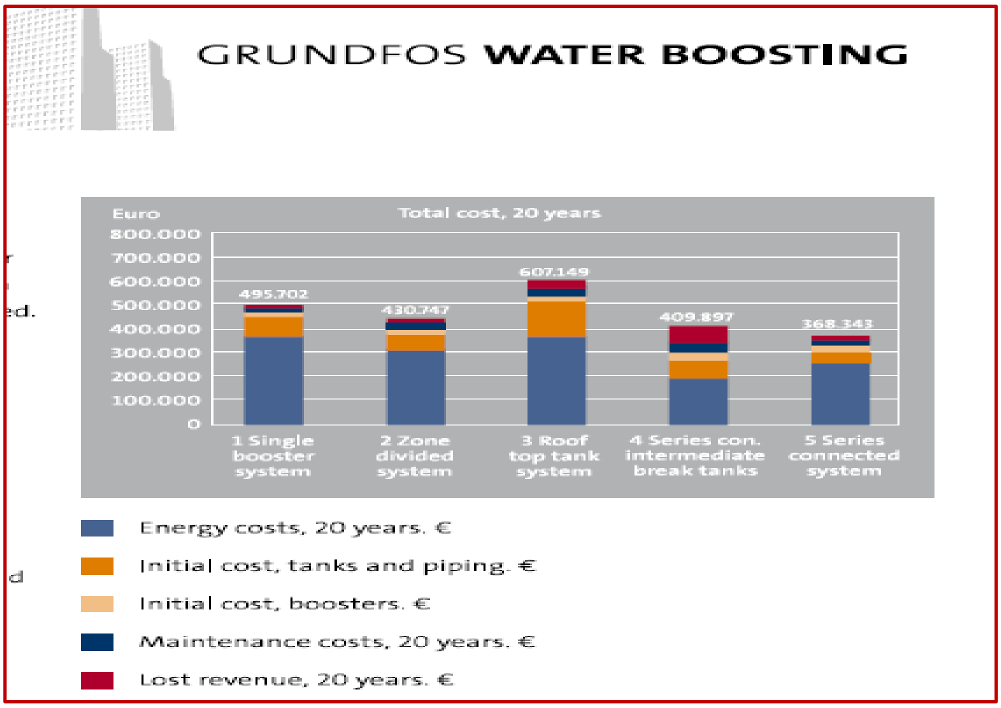


Fig. 21- Grundfos Life Cycle Cost Case Study for all type of Domestic Water System Boosting Design

For application having dual season, air conditioning and the air distribution system must zone properly. Rooms attached to façade must be zoned separately to the internal rooms to avoid cooling followed by excessive heating during summer night and winter, which is a tremendous wasting of energy (especially for healthcare and other clean room application which require minimum air changes and small range of %RH fluctuation; and buildings with low thermal mass- see section 6.4 for further explanation).

6.7 Shared-Loading against Single Equipment Loading during Part Load for VFD (Variable Frequency Drive) Fans/Pumps

A single fan or pump running at part load consumes more power compare to two (2) equipment sharing the same load. For example, a primary pump composed of 2-duty and 1-back up and the cooling load required during part load can be handle by single pump. If the flowrate is 100 L/s and the head is 4 bar, the power required by single pump is 57 Kw (for 70% combined pump-motor efficiency) when it runs at full speed. If the same load is shared by the two (2) duty pumps (considering that the DPS/T is in the index then the speed range reduction can be maximized down to 25%-30% of full speed and the efficiency is maintain and maximized throughout) then each of the two (2) pumps will run at 50% of its maximum speed since the affinity law states that the speed varies directly with the flow. The power consumed by each pump will be 7.125 Kw since the affinity law states that the power varies directly as the cube of rotating speed. Then the power consumption of the two (2) pumps in reduced speed is only 14.2 Kw.

Almost all projects uses single equipment loading during part load resulting in excessive power consumption. This technique can easily be implemented in any existing projects without additional cost since this will only require change in VFD controller programming. Note that the technique is not applicable for booster pumping due to fixed static head and residual pressure, and is applicable (but not limited) to the following applications;

1. Multiple Variable Speed Cooling Towers
2. Primary and Secondary Variable Speed Pumping System for Chilled Water
3. Variable Speed Condenser Pumping System
4. Fan Arrays used for Variable Speed Air Handling Units

6.8 Air-to-air Energy Recovery Equipment

Air-to-air energy recovery equipment are used for the following reasons;

1. To recover energy (sensible and latent heat) from the exhaust air and transfer the energy to the incoming fresh air, thus reducing the energy require to treat or condition the incoming air. The following are the commonly used energy recovery equipment suitable for this purpose:
    a. Energy (or Heat) Recovery Wheel – recover both sensible and latent or sensible heat only. Has the highest effectiveness available in the market (based on current data) and is the most common type of energy recovery equipment. Equipment requires sufficient headroom due to height restrain and has issues with minor cross contamination.

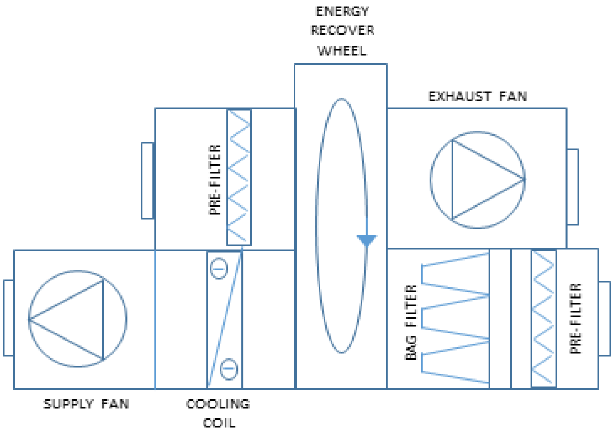


Fig.22 OA-AHU with Energy Recovery Wheel

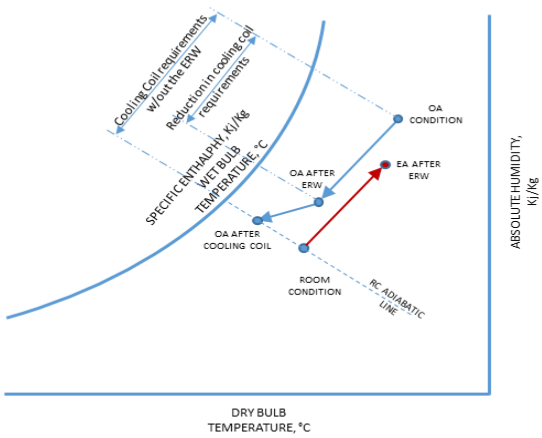


Fig.23 Psychrometry showing reduction in Cooling Coil Requirements during summer

Figure 22 shows a typical OA-AHU with Energy Recovery Wheel. Fig. 23 shows the Psychrometric Process indicating the reduction of Cooling Coil requirements due to ERW during summer season. ERW can offer up to 85% recovery of the total energy from the exhaust air by thermal heat and mass transfer. Outside air conditioning needs to stop at the room adiabatic condition since this condition does not add or deduct heat from the room. The room cooling load is taken care by the FCU (Fan Coil Unit) or other indoor cooling unit.

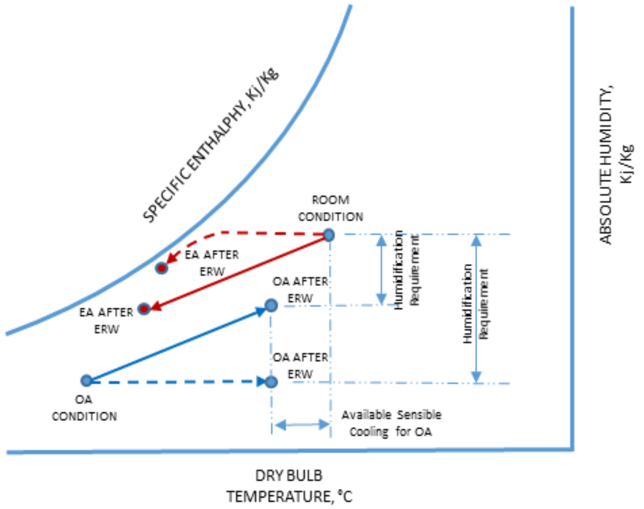


Fig.24 Psychrometry of OA-AHU with Energy Recovery Wheel During Winter (above 0°C) - High Total Effectiveness

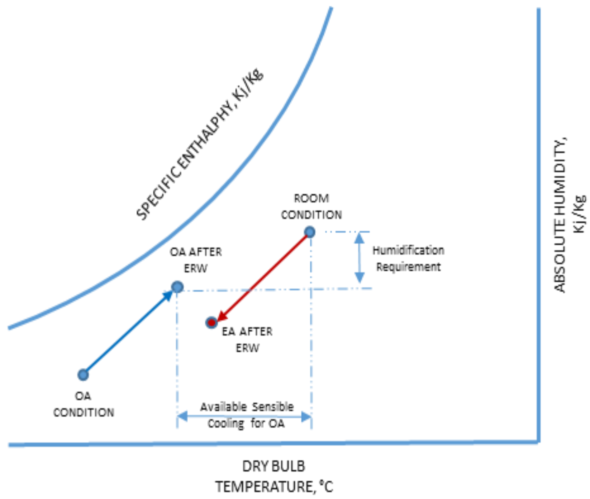


Fig. 25 OA-AHU with less sensible Effectiveness and High Latent Effectiveness

Fig. 24 shows the Psychrometry of the air using an OA-AHU with ERW during winter season (above 0°C). The solid line represent sensible and latent heat recovery while the dashed line represent sensible heat recovery only. Sensible heat recovery is applied for contaminated exhaust. Unlike in the summer season, higher effectiveness for both sensible and latent recover is always desirable. For winter season, selection of energy recovery equipment as well as its effectiveness depends upon the application and high effectiveness is not always desirable. Consider a properly zoned floor an Office Building (or other application such as Residential or Healthcare) where all rooms attached to the façade (external rooms) is served by a dedicated AHU and all internal rooms by another dedicated AHU. Zoning results in energy savings since the external rooms will require heating during winter (especially if the façade is made up of glass) and the internal rooms will still require cooling. Mixing internal and external rooms will result in excessive heating for external rooms during winter since cooling is applied followed by heating. The same scenario is true if DOAS (Dedicated Outdoor Air System) is used for fresh air. Cold air during winter can be used naturally to cool internal spaces provide it is heated to exceed the room dew point to avoid condensation. Outdoor air for external rooms is required to be heated inline to the room adiabatic dry bulb temperature or additional heating can be added to the room recirculating unit if outdoor air heating only exceeds the room dew point temperature. If an ERW is used to recover the energy of the exhaust air, high total effectiveness is required for the OA-AHU serving the external rooms to reduce heating and humidification. The OA-AHU serving the internal rooms require less sensible effectiveness and high latent effectiveness (as shown in Fig. 25) since the outside air can be used for the cooling requirement of the internal rooms. Sensible heat recovery only requires to exceed the room dew point to avoid condensation while high latent recovery is required to reduce humidification required to attain the required room %RH.

If the building is located where dual season is experienced (such as Al Khobar and Jeddah) and the designer selects an ERW for OA-AHU serving internal rooms with high total effectiveness, energy savings is expected during summer but energy is wasted during winter due to high recovery of sensible heat, reducing the OA cooling capability. If the rotation of the wheel is controlled to reduce sensible heat recovery, latent heat recovery will also reduce which results in increased humidification for the room. To reduce the power consumption especially during winter, the designer will require to provide two types of energy recovery equipment.

b. Heat Pipes - recovers sensible heat only and normally used for contaminated exhaust. Preferred for sensible energy recovery due to lower pressure drop compared to Fixed Plate.

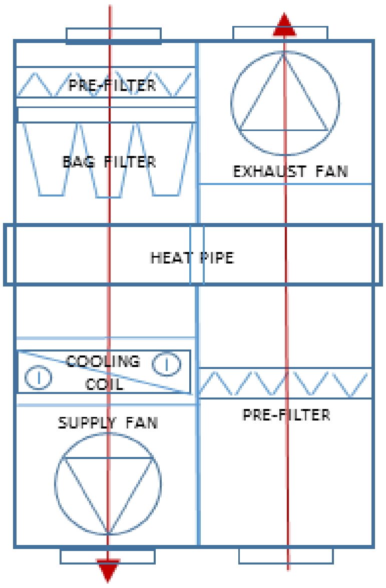


Fig.26 OA-AHU with Heat Pipe for Sensible Energy Recovery using Heat Pipe

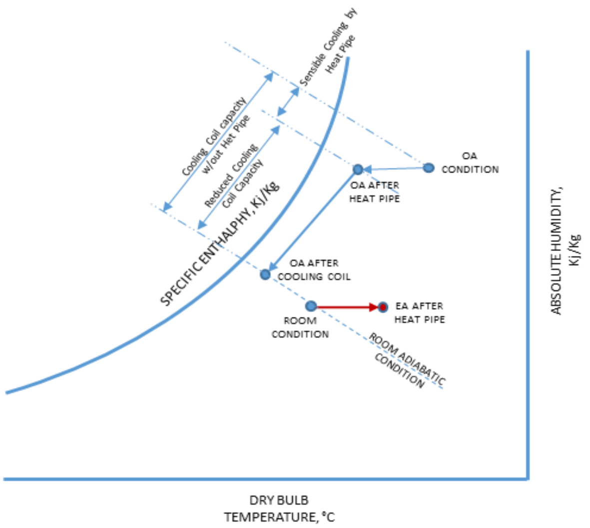


Fig. 27 Psychrometry Showing Reduction in Cooling Coil Requirements

c. Pleated Membrane and Fixed Plate Heat Exchanger - recover both sensible and latent heat or sensible heat only. Effectiveness lower than that of the Energy Recovery Wheel but no issues with cross contamination.

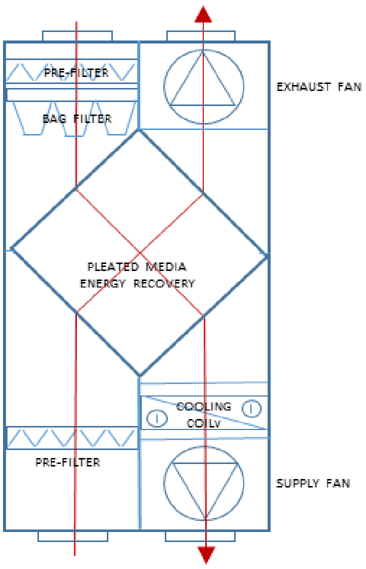


Fig. 28 OA-AHU with Pleated Energy Recovery Media For Sensible and Latent Heat Recovery

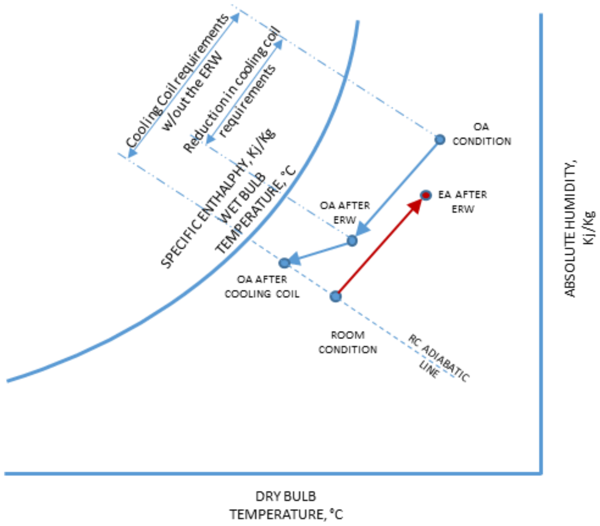


Fig. 29 Psychrometry of Pleated Media with Sensible and Latent Heat Recovery

2. Across a cooling coil to provide cooling and reheat for special application (such as high latent load application) thus reducing requirement for cooling while providing reheat. Heat Pipe for this kind of application saves about 35% to 50% of the electric power and applicable for Dancing Halls, Auditoriums, Natatoriums, Conference Halls, and other similar type of use.

a. Heat Pipes – see Fig. 30 and Fig.31

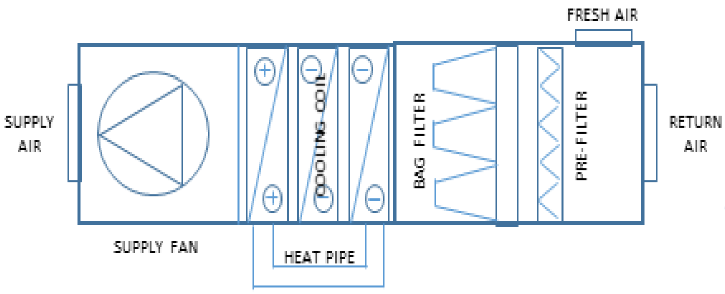


Fig. 30 Mixed Air AHU with Heat Pipe

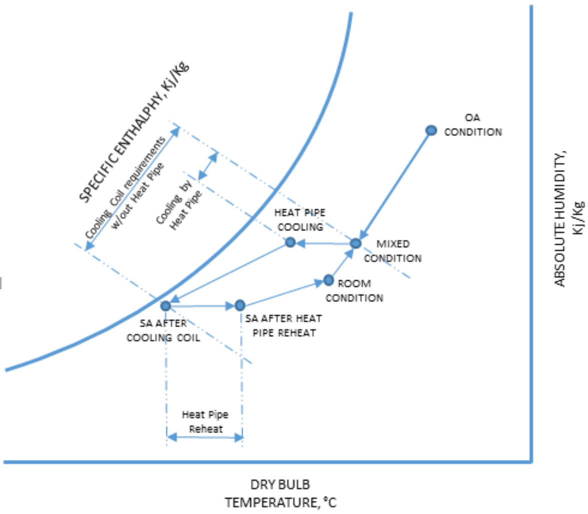


Fig.31 Psychrometry of Recirculating AHU with Heat Pipe

b. Pleated Membrane and Fixed Plate Heat Exchanger – Psychrometry is typical to Fig. 32

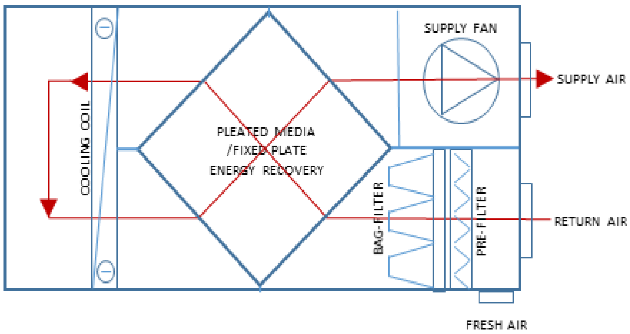


Fig. 32 Mixed Air AHU with Fixed Plate / Pleated Media Heat Exchanger

3. Across the cooling coil of an Outdoor AHU for DOAS application to provide both cooling and reheat.

a. Heat Pipes

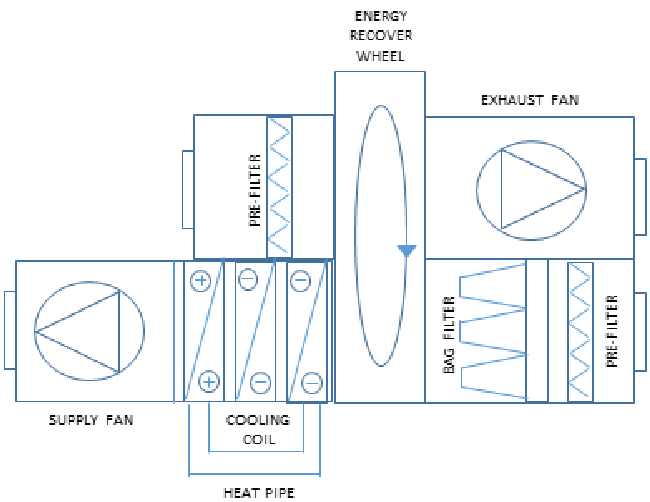


Fig.33 OA-AHU with Energy Recovery Wheel and Heat Pipe

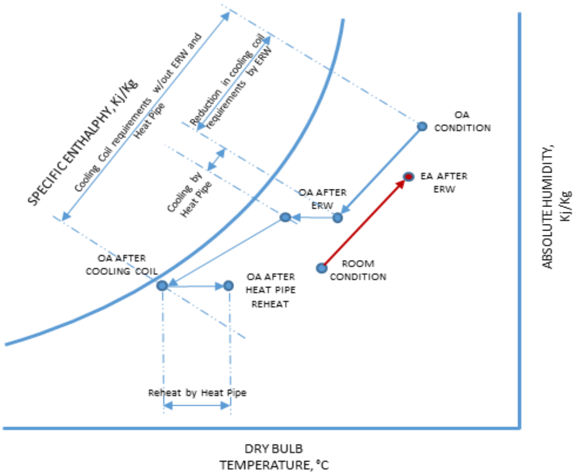


Fig. 34 OA-AHU with ERW and Heat Pipe

Fig. 33 shows a typical OA-AHU with ERW and Heat Pipe while Fig.34 shows the Psychrometric Process of the Unit. The condition after the cooling coil needs to go below the room adiabatic condition to provide additional cooling and latent load capacity specially for radiant cooling application (chilled beams and ceilings where sensible cooling capacity are limited with no latent load capacity). Reheat is required to exceed room dew point temperature to avoid condensation. Power savings can reach up to 60% (inclusive of media pressure drop) in humid and hot ambient condition compared to pure Dx System and electric reheat.

Energy Recovery Equipment saves considerable amount of electrical energy thru mass and heat transfer but pressure drop across the equipment must always be considered. Most manufacturers specify from 1000 L/s to 1500 L/s as the margin for having a considerable energy savings for AHU at the design ambient and exhaust condition. When ambient condition falls (to around 30°C during summer), the equipment must be provided with means to by-pass the energy recovery media since the power produced by the medium pressure drop and airflow exceeds the energy that is recovered. Consult the manufacturer for specific data.

6.9 Combining Fixed Speed Chillers, Variable Speed Chillers, Large Capacity, and Small Capacity Chillers

Fixed Speed Chillers (FSC) are chillers running at constant speed and have several steps for capacity control. Capacity control can be either hot gas bypass, sliding plates, or other means of capacity control. Variable Speed Chillers (VSC) have “thousands” of capacity control within its speed variation range. Variable Speed Chillers are less efficiency at full load compare to its FSC counterparts at same capacity and design lift as indicated in ASHRAE Equipment Manual (see extract below) and chiller manufacturers.

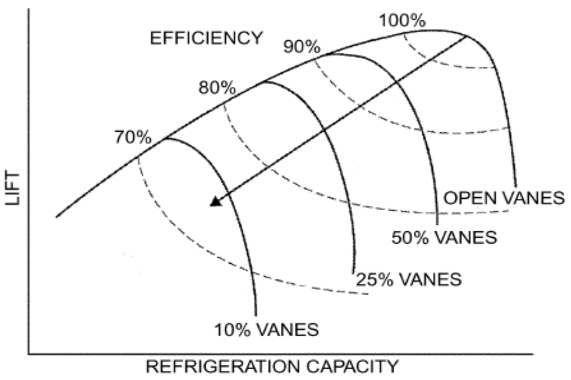


Fig. 35Extract from ASHRAE Equipment Handbook for Fixed Speed Chiller Performance

Centrifugal Compressor Performance at Constant Speed
(Carrier 2004)

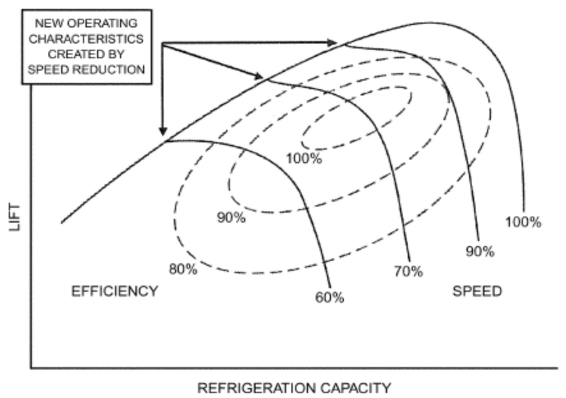


Fig. 36 Extract from ASHRAE Equipment Handbook for Variable Speed Chiller Performance

Variable-Speed Centrifugal Compressor Performance
(Carrier 2004)

FSC are very efficient at full load and efficiency degrades during part load (with the exceptions of modern limitless stepper control chillers) while VSC are efficient at part load. The graph below (Figure 37) represent the current characteristics of both type of chillers at the same capacity and lift (report from other manufacturers varies with different type of chiller, data is for Centrifugal and Screw Chillers). Note that characteristics are not always guaranteed since technological advancement can always change the performance of any HVAC Equipment.

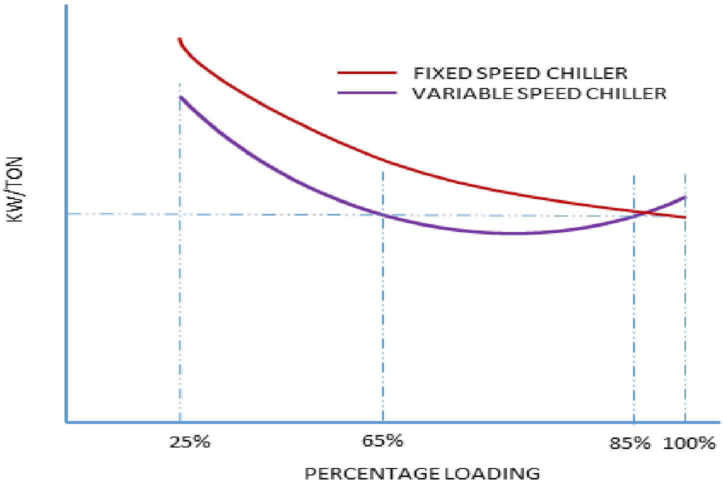


Fig. 37 Fixed and Variable Speed Chiller Performance

Below (Figure 38) is an extract from Trane regarding characteristics of Adjustable Frequency Drive (AFD) or Variable Speed Drive (VSD) Centrifugal Chillers. Centrifugal chillers are the most efficient type of chiller.

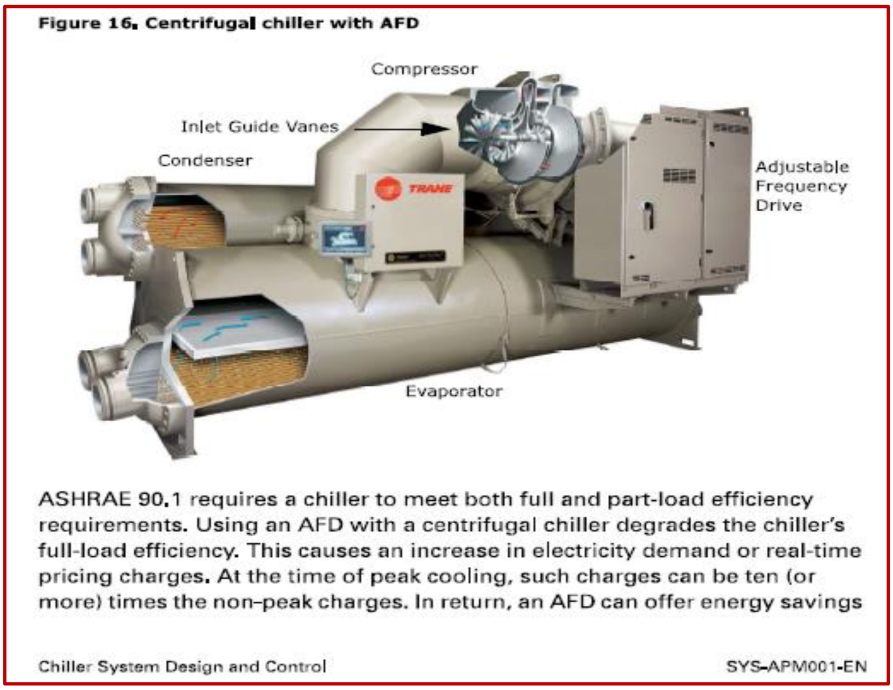


Fig. 38 Trane Variable Speed Chiller Performance

In large projects utilizing multiple chillers, it is required to mix VSC and FSC for energy and first cost savings (fixed speed chiller is about 15% to 20% cheaper than variable speed drive depending on capacity). The system must be design to maximize capacity of FSC by using PICV (Pressure Independent Control Valves) and utilizing the VSC during part load or excess load (to that of the FSC at full capacity) from 65% to 85% of its capacity to maximized energy consumption. The BMS plays an important part as well as the field device accuracy in measuring flowrate (which is the representation of the building load) to calculate the excess flow from the total number of required fixed speed chiller to run.

Almost all HVAC designers utilized equally size chillers as a default design but there are significant energy savings when providing smaller sized chillers (say 1/2 of the size of all other chillers) to handle very low cooling load. For instance, if the required minimum load is 70 Tons, then it is advantageous to run one (1) 100T VSD chiller to maximize efficiency. If the required load is 40 Tons, then one (1) 50T VSD chiller would be preferential. If the building load is 430 Tons and up, running five (5) 100 Tons fixed speed chillers are required since capacity at shared loading (except for CHW piping design where there is preferential loading) will be 86%.

Fig. 39 - Fixed Speed Chiller and Variable Speed Chiller testing at Duke University

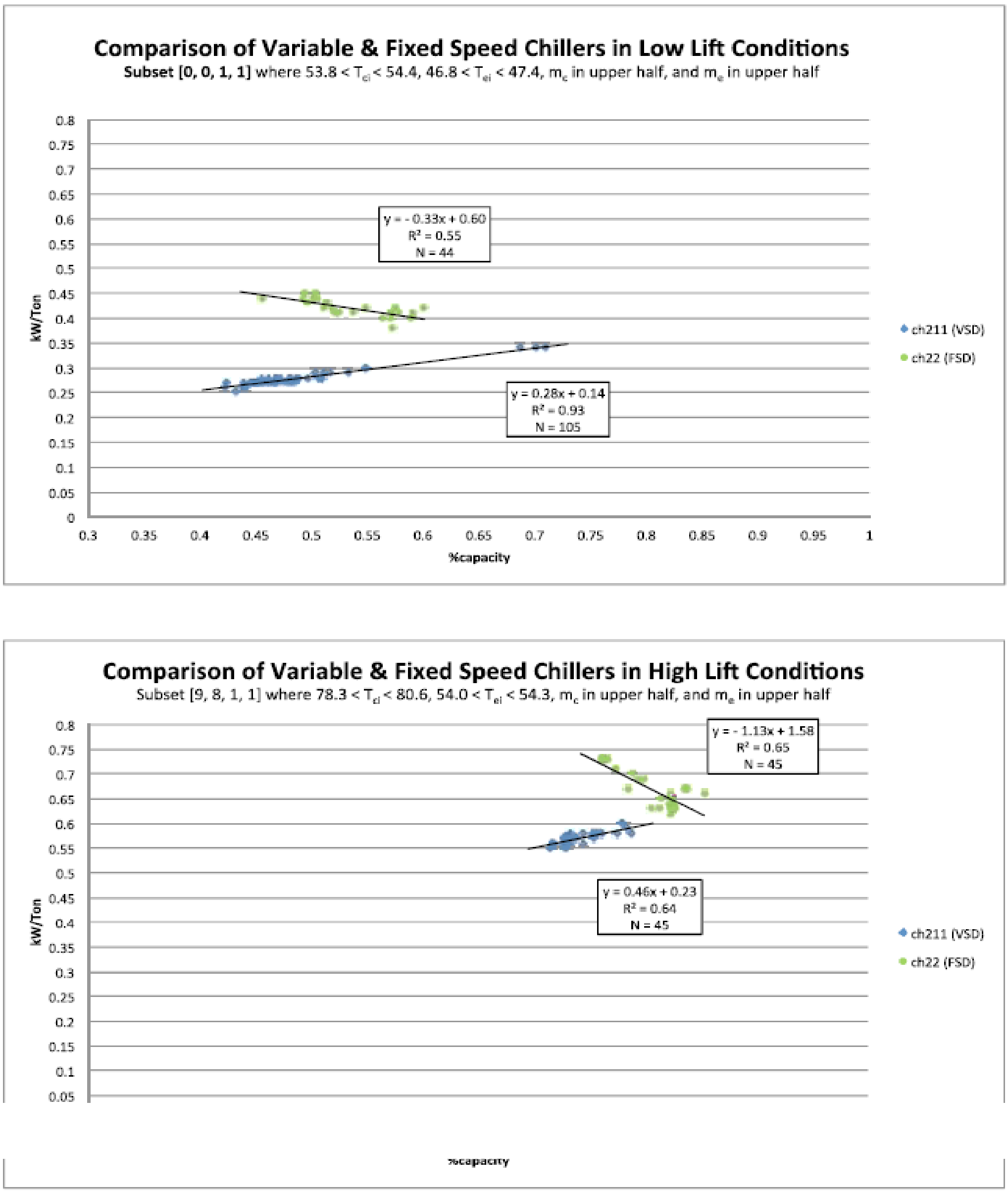


FIGURE 2. CHILLER NO. 211 (VSD) & CHILLER NO. 22 (FSD) COMBINED REGRESSION PLOTS UNDER LOW AND HIGH LIFT CONDITIONS

Operatic Strategy must be thorc , in a hospital which requires 9000 Tons cooling load and the minimum calculated load is 750 Tons due to minimum ACH, reheat requirement, and large heat dissipation in medical equipment. If chillers are sized for 220 Tons each to resolved length of units during site delivery so that 41 units are required, then only five (5) chillers are required to be VSD and the rest to be FSD since loads higher than 1100 Tons which will require six (6) chillers to run will have shared loading of 85% and higher. There will be no requirement for smaller chiller capacity to handle very low load. This technique will result not only in energy efficiency but also considerable decrease in Project Capital Expenses.

It is also an important factor in view of power savings to select a VSD chiller with the lowest minimum flow, both in the evaporator and condenser (for variable flow condenser) avoiding by-pass water to reduce pressure loss in the evaporator and condenser; providing the lowest speed reduction for pumps; and lower chiller power consumption, thereby reducing power requirements. Chiller with minimum evaporator flowrate up to 21% or lower can be selected to reduce power consumption during minimum load.

6.10 Utilizing Displacement Ventilation for Large and High Ceiling Space

Displacement Ventilation is a technique used in HVAC for the main purpose of reducing the power required compared to mixed-air type air-conditioning system. Mixed-air air-conditioning requires placement of supply diffusers at high level and primarily relies on jet velocities to induce the secondary air in the room thereby promoting air mixing. Air mixing through induction creates a homogenous temperature within the room and eliminating stratification. In Displacement Ventilation, supply air is introduce to the room at low level and the return air at ceiling level (or in the walls at level above the controlled zone) thus intentionally promoting stratification and separation between supply air and stale air. This technique is very efficient in removing air contaminants and enhanced Indoor Air Quality (IAQ) since convective plume rises which carries air contaminants and exhausted. Zone Distribution Effectiveness is 1.2 as per ASHRAE 62.1 compared to 1.0 for Mixed Air System, which result in substantial reduction in the required outdoor air by ASHRAE to attain the target IAQ. Air is supplied at a temperature higher than mixed-air ventilation (above 16°C) to avoid occupant discomfort (such as cold feet and cold draft) and at low face velocity (within the range of 0.25 m/s to 0.35 m/s, or 0.5 m/s as the maximum limit). Cooling load as well as the quantity of supply air is calculated based on the considered comfort zone to space height ratio (comfort zone can be 2 meters above the floor), supply air temperature, and occupied zone temperature gradient (must not exceed 3°C from foot to head). Convective capacity of the Displacement Ventilation is limited to 50 w/m2. Radiant cooling system can be used in combination with Displacement Ventilation to offset space heat gain and increase convective capacity per m2 of floor area. Displacement Ventilation is used for application with ceiling greater than 3 meters in height and commonly used in large open office set-up, restaurants, theatres, auditoriums, hospital and cleanrooms, casinos, etc.

As stated above, reduction in power requirements are primarily due to (1) 20% reduction in the required outdoor air to attain the IAQ by ASHRAE 62.1 especially in high humid area such as the Middle East; (2) and reduction of supply air flow and cooling requirements due to flexibility in the consideration of comfort zone height++

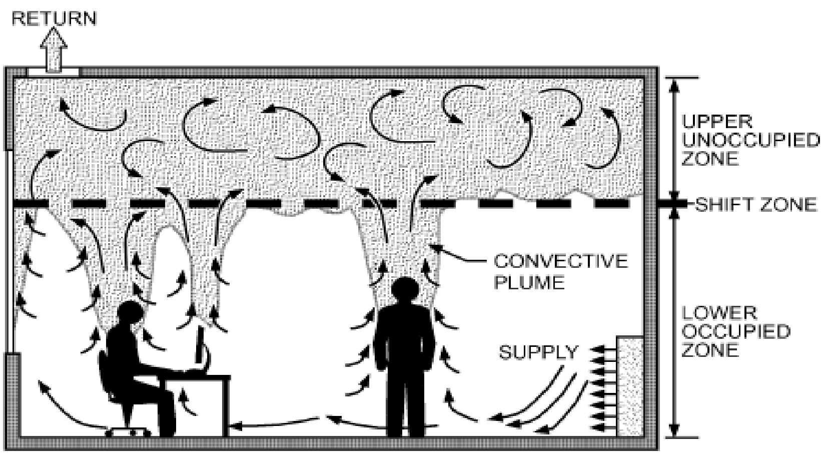


Fig. 40 Displacement Ventilation

6.11 Increasing Chilled Water and Condenser Water Delta T

Most HVAC Designers and chiller manufacturer are aware about the most efficient Delta T that produce the 2.4gpm/Ton for chilled water system and 3.0gpm/Ton for condenser water which is 5.55 °C. Increasing the Delta T from this parameter decrease the efficiency of the chiller especially if the widening is due to decreased chilled water supply temperature and increased outgoing condenser water (or air) temperature. Increasing the chilled water supply temperature and decreasing the outgoing condenser water (or air) temperature while maintaining the design Delta T increase the chiller efficiency *(see Reset Control Technique in "System Controls Section 6.28.1.a" for the separate discussion about this topic regarding energy savings).* Although increasing the Delta T reduced chiller efficiency, the pump power required for the same cooling reduces significantly as well as the Cooling Tower fan power due to reduced flow (see section 6.12 for exact information on Chiller-Cooling Tower Energy Balance on what ambient wet-bulb temperature applicable). Additional benefits for increasing the Delta T is reduced piping size; smaller coils (and pressure drop) for AHU and FCU; lower pressure drop for evaporator and condenser which results in lower Capital Expenditure and power savings.

For chilled water system, modern chillers can handle chilled water supply as low as 1.1°C (see Trane document extract below) and chilled water return as high as 14°C (even up to 16°C for non %RH critical application). Lower chilled water supply results in lower coil ADP (Apparatus Dew Point) and lower off-coil temperature resulting in lower airflow for mixed-air recirculating AHU and FCU, thus lowering power consumption for fans. Lower air flow results in smaller ducting and reduced duct supports for lower Capital Expenditures. Lower off-coil also eliminates need of reheat for non-critical high latent load application; and higher latent capacity for DOAS in-case of radiant cooling system thus reducing required airflow rate to offset room latent load. Designer are advised to coordinate closely the chiller minimum required flowrate when designing and selecting chillers and design Delta T.

Chilled-Water Temperatures

Chilled water (without antifreeze) at 34°F (1.1°C) is possible with some chillers that use sophisticated evaporator-design and chiller-control methods.

Currently, comfort cooling systems are designed with chilled-water supply temperatures that range from 44°F [6.7°C] to 38°F [3.3°C], and, in some cases, as low as 34°F [1.1°C]. Reasons to decrease the chilled-water temperature include the following:

- The system design more readily accommodates wider temperature differences (lower flow rates) than the standard rating conditions (see "Selecting flow rates" on page 30).
- Lower water temperature allows lower air temperatures (and flows) to be selected, resulting in reduced airside installed and operating costs.
- Colder water in the same chilled-water coil may provide better dehumidification.
- Colder water can be used to increase the capacity of an existing chilled-water distribution system. In some instances, this can save significant capital expenditures to add capacity to large central plants that have reached their flow limits.

Incoming water to the chiller condenser is limited by the design wet bulb temperature and the "approach" of the Cooling Tower. The prevailing wet bulb temperature depends on the meteorological data on the location of the project. Cooling Tower best approach in the market is 2.8°C (considering power efficiency point-of- view for moderate wet-bulb temperature) using square-pattern sprinklers, while the incoming water to the chiller ranges from 29.5°C to 31°C (lower value can be used for low prevailing %RH location) . When selecting the appropriate CT range, coordination with chiller manufacturer must be done to ensure minimum allowable flowrate to avoid premature fouling of the condenser especially for variable flow condenser type chiller. Lowering the required "approach" and "range" increases the tower footprint and first cost.

6.12 Chiller and Cooling Tower Energy Balance

In any HVAC System, the power consumed by the chiller accounts to 85% of the power consumed by the system or 60% of the building power consumption (see Fig. 41A, extract from CTI- Cooling Technology

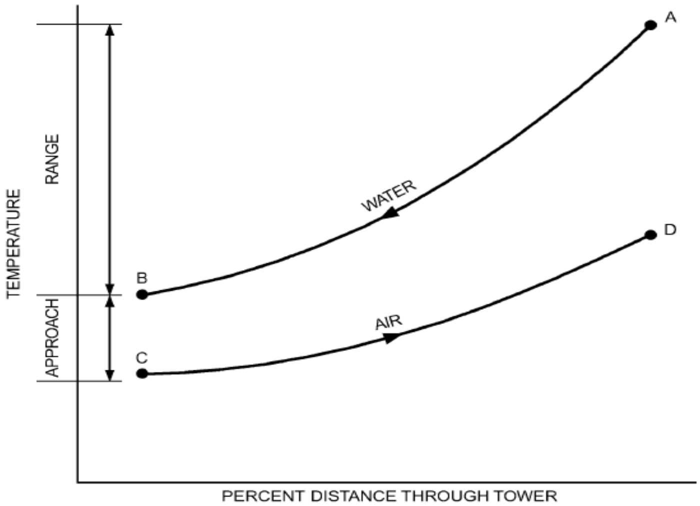


Fig.41 - Temperature Relationship between Water and Air in Counter flow Cooling Tower

Institute) while the tower including the condenser circuit accounts for 6-10% of the building power consumption. As discussed in section 6.11, it is ideal to design the chiller condenser with increased Delta T so that the Cooling Tower "Range" can be maximized but the technique will reduce the efficiency of the chiller. Theoretically, the wider the "Range" the more efficient is the condenser pumping and CT fan power consumption which compensates and exceeds the lowering of the chiller power efficiency. "Range" is the difference in temperature of the incoming water to the outgoing water from the CT and "Approach" is the difference in temperature between the outgoing water to the wet-bulb temperature of the air and is the representation of Cooling Tower Efficiency (see Fig. 41). The lower the "approach", the lower is the power consumption of the tower fan per ton of evaporator cooling, but both the "approach" and "range" is greatly affected by the wet-bulb temperature. The lower the wet-bulb temperature, the wider the "approach" at constant cooling load and the fan power increase. Because of this counter intuitive condition, it is required to know at what condition the combination of all equipment power will be at the lowest per rejected heat by the condenser. Energy Balance Study plays an important part to optimize energy.

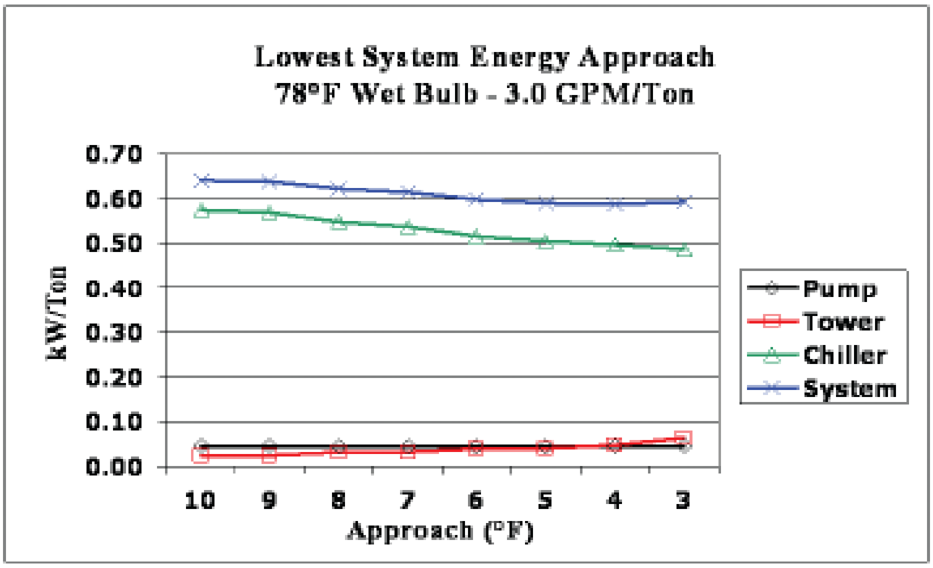


Fig. 41A– Power Consumption of HVAC Equipment

At higher ambient wet-bulb temperature, energy optimization for chiller and CT does not rely only with wider CT "range" but by the energy balance between the chiller heat dissipation, design condenser Delta T, and CT "approach" as shown in the following CTI Cooling Tower data (Fig. 42 to 45).

From the below figure, it can be concluded that wider Delta T (or higher CT "range") for condenser water is only beneficial for energy savings at 66°F (18.9°C) wb and below. Interesting to see that the higher "approach" (from 5-10°F) as well as wider "range" has no further influence in chiller and Cooling Tower power consumption (Fig. 30) at lower wet-bulb temperature, unlike in higher wet-bulb (from 66°F and above), therefore lower "approach" is mandatory. For higher ambient wet-bulb temperature, the standard 3.0 gpm/ton based on 10°F (5.5°C) delta T or "range" and smaller "approach" will be at advantage.

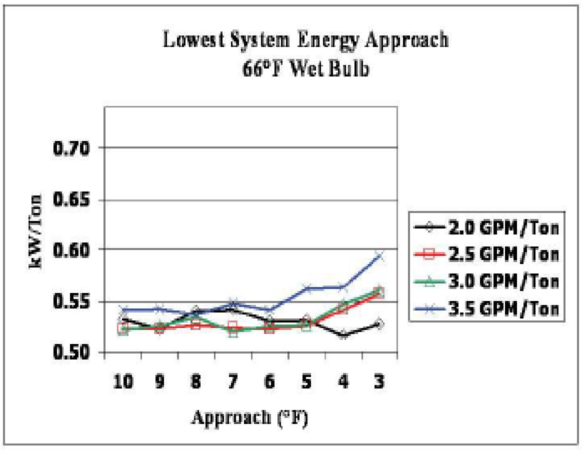


Fig. 42 – Chiller/CT Power Consumption per Ton at 66°F

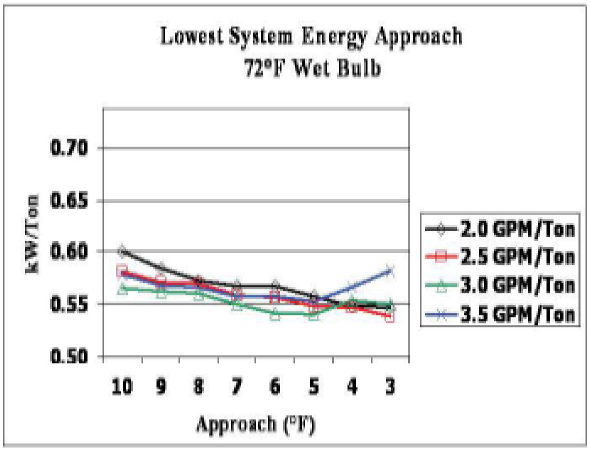


Fig. 43 – Chiller/CT Power Consumption per Ton at 72°F

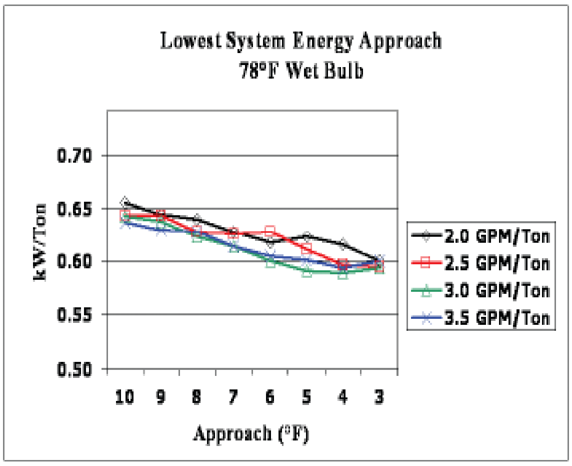


Fig. 44 – Chiller/CT Power Consumption per Ton at 78°F

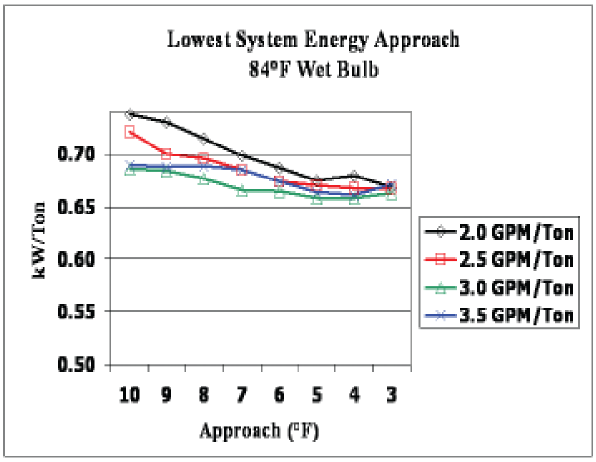


Fig. 45 – Chiller/CT Power Consumption per Ton at 84°F

The designer is therefore advised to conduct a thorough study of the project's meteorological data before deciding for wider "range" (lower condenser flow) and "approach". Based on Fig. 45, improper design and specification can yield to system energy penalty of 6% or higher (or above 4.2% of the building power consumption).

6.13 Free-Cooling Chillers

Technological advancement in chillers incorporates free-cooling technique both for air cooled and water cooled chillers. For building comfort design of 24°C and 55% RH (where dew point is 15°C) and when ambient air is lower than 13°C, or if the ambient air wet-bulb is low enough to produce 13°C or lower (at saturation) water on the cooling tower basin, this can be used to pre-cool the chilled water returning to the chiller to reduce the work and power utilized by the chiller compressor. When the chilled water distribution system is properly design and PICV is used to attain the design Delta T, proper chiller reset control (based on outgoing temperature from the free-cooling heat changer) can be applied to maximize chillers capacity during low cooling demand.

If the chilled water Delta T is design at 14°C, then each degree C from 13°C (considering an approach of 1°C) accounts for 7% savings in the power consumed by the running chiller, provided that the process fluid flow (air or water) which absorbs the rejected heat flow is properly design.

6.14 Heat Recovery Chillers (HRC)

The chiller condenser dissipates large amount of high-temperature heat which can be recovered and utilized for other heating, such as domestic hot water system. Utilizing Heat Recovery Chillers reduce or eliminate the burden of electricity used for electric water heating which accounts for 15-20% of the building total power consumption without or with minimal first cost implication. To maximize the heat dissipated by the condenser and to suit the heating requirements, chilled water system arrangement should be design so that the HRC will have the “preferential loading”. “Preferential Loading” ensure that the HRC will always run at full capacity even at minimal building cooling load requirements.

The designer of the HVAC System must consider the balance between the heating requirements and the HRC condenser heat dissipation at minimum building cooling load.

6.15 Double-effect Absorption Chillers

In certain cases, there are projects such as turbine power plants which disposed the heat exhausted by the turbine directly into the atmosphere or cooled by sea water cooling system. The heat disposed can be re-used using double-effect absorption chillers which uses a double stage evaporating effect using medium-grade heat source (heat source from 120°C to 185°C). Energy savings utilizing this type of chiller depends on the amount of available heat source.

Currently technology has develop solar hot water heating system as source for medium-grade heating capable to be utilized for double effect absorption chillers to reduce or eliminate energy burden from electric chillers.

6.16 Ammonia Chillers

Use of ammonia chillers were limited in the past to refrigeration system due to flammability and toxicity if accidentally released in high concentration. Due to the current competition to reduce energy consumption, ammonia chillers are becoming very competitive in the market and the current concentrations for most chiller manufacturer. Ammonia has the highest enthalpy at evaporating temperature of 4.4°C from liquid to vapor saturation compare to any existing refrigerant which results in lower refrigerant flowrate per kJ of heat absorbed in the evaporator. This reduce flowrate results in smaller compressor and lower power consumption.

Comparing theoretical full load Coefficient of Performance (COP) between Ammonia and R-134a at 4.4°C evaporating temperature and 50°C condenser temperature reveals better characteristics for Ammonia (Ammonia is appx. 8.3% efficient than R-134a), but due to compressor design and limitation (Ammonia chillers largely uses reciprocating compressors which has higher friction losses) this advantage is reduced. Refer to the below extract from Johnson Controls and York:

Advantage of using ammonia includes (1) Zero Global Warning Potential and Ozone Depletion Potential, (2) naturally available and very cheap, and (3) high efficiency at full-load is constantly maintained at part load for Variable Speed Drive, and (4) qualification for LEED EA credit 4- Enhanced Refrigerant Management. If the local regulation permits and integrating safety systems (such as ammonia detection and emergency exhaust system), the use of Ammonia Chillers must be taken into priority considering energy consumption reduction.

- What are the key markets for ammonia today?

Industrial Refrigeration: Today, ammonia remains the most cost-effective and energy-efficient option for all types of industrial equipment. In fact, R717 makes up 15% of the total refrigerant market. R717 is expected to remain the preferred choice for large installations once ozone-depleting substances will be ruled out under international agreements. In the US and Canada ammonia is well regulated and enjoys a wide-spread use. In Europe, R717 has been widely adopted for industrial refrigeration in the UK and Germany but is more tightly regulated in France, Belgium, the Netherlands and Italy, and it is consequently less common. It is the most common alternative to HFCs for larger systems in Scandinavia, mainly as a result of restrictions and taxation on greenhouse gases. First installations in Australia have proved the efficiency of R717 plants. As a most recent example, ammonia freezes 6000 head of lamb every day in one of Australia's largest meat processing companies, being capable of freezing 1610 bulk packed export meat cartons with a 24-hour turn around.

Commercial Refrigeration: The use of ammonia in cascade supermarket refrigeration systems is growing, especially in countries with stringent limitations on the use of HCFCs and HFCs, such as Scandinavia.

Chillers: Although the use of R717 is still limited, this market is expected to grow. As a most recent example, a central ammonia chilling plant provides continuous supply of hot and chilled water for heating and air conditioning to Terminal 5 of London's airport Heathrow.

Heat pumps: R717 has been applied in medium-size and large capacity heat pumps, mainly in Scandinavia, Germany, Switzerland, and the Netherlands.

6.17 Evaporative Cooling

Evaporative cooling is a technique used to cool the air by the evaporation of water instead of vapor compression cycle with the use of refrigerant, thus extremely reducing electrical power consumption while sacrificing increase in water consumption. Humidification or dehumidification of the air which occurs during cooling process depends primarily on the temperature of the sprayed water. Spraying or misting increases the contact surface between the air and water, thus increasing the cooling effect. Evaporative cooling is used primarily in ventilation air cooling such as outdoor air and recirculating air conditioning if source of cheap and clean water is in abundance. In evaporative cooling, three (3) main factors plays an important role namely; (1) dry-bulb temperature, (2) wet-bulb temperature, and the (3) spray water temperature. The wet-bulb temperature is the lowest temperature that the saturated air can attain if sufficient recirculated water is sprayed. The dry-bulb temperature (or it can be substituted by %RH), dictates the amount of water that needs to be evaporated to attain the required sensible cooling at constant wet-bulb. Providing external cooling to the spray water results in lower dry-bulb temperature, and if water is substantially cooled below the air dew point before spraying then dehumidification will occur. If the spray water temperature is above the air dew point, humidification will happen. If external heating is provided to the spray water below the air dry-bulb temperature, humidification and cooling above the constant wet-bulb temperature line will occur, such as in the case of cooling towers.

When evaporative cooling is used for an Open Type Cooling Tower, HVAC system power is reduced by half since the chiller is the main consumer of electrical power which accounts for 25-30% reduction in building power. When Closed Type Cooling Tower, power reduction is decreased by 3-5% due to additional pump for the process fluid circuit.

Evaporative cooling can be further improve using two-stage evaporative cooling technique where water is sprayed in exhaust air from the building to reduce the dry-bulb temperature before passing thru an air heat exchanger (indirect evaporative cooling). Hot outdoor air (OA) passes thru the other side of the heat exchanger where thermal exchange happens, resulting in lower dry-bulb temperature of incoming outdoor air. The OA then goes to direct evaporative cooling as described above to further reduce the dry-bulb temperature.

6.18 Night Pre-cooling

This technique is applied for day-occupied type of building and requires to run the mechanical Air Handling Unit to pre-cool the internal walls of the building by means of mechanical cooling or natural ambient air cooling during the nighttime hours. Night Pre-cooling is applicable where ambient air is colder and dry during the night time hours or the cost of electricity is cheaper during nighttime. Building thermal mass (external walls and internal wall) acts as heat sinks to store coldness and uses it during daytime to absorb heat and flatten the energy consumption of the building during early peak cooling hours.

Night Pre-cooling using natural ambient air is applicable when it is colder and dry at night compare to daytime. The technique is also effective for flushing of air pollutants and other harmful volatile organic compound (VOC) emissions for better IAQ (Indoor Air Quality). The HVAC designer must consider the effect of higher %RH(Relative Humidity) in the building internal components to avoid absorption of moisture that acts as latent load (desorption) during the activation of cooling system at daytime.

Mechanical cooling or LDAC (see section 6.5 for LDAC details) is required for the application to reduce the moisture content when it is colder during the night hours. Mechanical cooling is also utilized for building with high thermal mass, both external and internal walls that acts as heat sinks (stores coldness) and reduce cooling requirements during the early peak cooling hours. Using the building thermal mass as heat sinks shifts the peak energy consumption during the day into the night hours, and is mostly applicable for project with higher utility rates during the day.

The time of operation depends upon the characteristic of the building and the purpose of Night Pre-cooling. If the purpose is for attaining IAQ for low thermal mass, the AHU can be activated 15 to 30 min. before occupancy. For high thermal mass building (say with 10 hrs. thermal time lagging), the west wall will have the peak sun radiation at 2pm and reach the internal wall surface by appx. 12am. It will require appx. half of the thermal time lag (4 hrs. or less) to remove the heat stored in the external walls since both side are colder. If the building will be occupied at 8am, then natural air Night Precooling must be applied at appx. 4am, or appx. 2-3 hrs. before occupancy for mechanical type Night Pre-cooling. The accurate time where the technique can be best activated based on the building thermal mass characteristics and usage can be determine using HVAC Software.

6.19 Radiant Cooling and Heating

Radiant Cooling and Heating uses temperature controlled building indoor surfaces such as walls, floors, and ceilings which are cooled or heated by recirculating hydronic system. Thermal radiation is transmitted in a straight line in the speed of light between two bodies of different temperature with no or very minimal effect on the surrounding air. Most common technological advancements are chilled ceiling and chilled beams which are either passive or active system (Hybrid or Load Sharing HVAC System). Active System (Hybrid) utilized other HVAC system such as DOAS (Dedicated Outdoor Air System) for primary air flow, and is necessary to increase the capacity of the Radiant Panel to suit some application since Passive System has very limited capacity (in watts/m2). Human body emits 40-45% radiant heat, 40% latent heat, and remaining as convective heat. Household appliance (not related to cooking) and office equipment emits 80% radiant heat and 20% convective heat. Fluorescent lighting emits 50-50% radiant and convective. Mixed-air air-conditioning system set at 24°C and 50%RH for optimum comfort, operates in convective and latent heat principle for cooling. Active Radiant Cooling primarily operates in the principle of thermal radiation, and secondary to convective and latent transfer so that the commonly used comfort cooling parameters do not apply, and higher room dry-bulb temperature can be use (for offices, 25-26°C is normally acceptable to have same comfort as in convective-latent cooling optimum parameter). Chilled water Delta T is limited (usually 3°C) due to increased supply chilled water temperature (usually 13°C for 14°C room dew-point temperature) to avoid condensation. Outdoor air dry-bulb temperature must be low enough to off-set latent load and reheated to avoid condensation (see section 6.8 for DOAS design using radiant panel).

Radiant cooling or heating has limited application and is applied to projects where the heat transfer ratio of thermal radiation to the convective and latent heat must be 50% or higher. Infiltration is purely convective load (and latent for humid areas), therefore building envelop plays an important role to the success of this technique. Active radiant cooling/heating has limited cooling and heating capacity (high limit is appx. 100 watts/m2) and cannot be applied for high density cooling and heating load application.

Energy savings using this technique primarily comes from the fact that the comfort cooling parameter (such as room dry-bulb temperature) is increased resulting in the decrease of sensible cooling load. Active radiant cooling and heating greatly reduce fan power compared to mixed-air recirculation system, since outdoor air required is usually sufficient to attain the required entrainment ratio (ratio of the secondary-air to primary-air) for induction purpose. Since CHW supply temperature is higher (especially for higher allowable room %RH application), free cooling can easily be adopted as means for saving energy using economizers (see section 6.13 for Free Cooling concept and section 6.23 for HVAC Economizers) to reduce power consumed by the chillers.

Coordination between the HVAC designer and architect pose a tremendous challenge for project utilizing this technique. First cost as well as installation cost is higher compare to other HVAC System and the system must be properly design to avoid condensation. The HVAC designer is strongly advised to carefully study the effect of offsetting the design latent load resulting in high outdoor airflow for application with high density occupation.

6.20 Demand Control Ventilation (DCV)

DCV is an energy savings technique that reduces supply air for ventilation (for outdoor and make-up air); or exhaust air (in-case of centralized kitchen exhaust system); or combination of both, as per the reduction in demand.

For outdoor air ventilation using DCV, the technique is more appropriate for medium to high density occupation. Variation in outdoor air supply is monitored thru CO2 sensor or Occupancy Sensor (Infrared) link to the Building Management System (BMS) controlling the supply motorized volume dampers.

DCV used for centralized kitchen exhaust system requires close coordination between the specialized hood and its DCV controller supplier, the main HVAC contractor, and BMS vendor/installer. Exhaust air flow control is conducted thru hood face temperature sensor and infrared sensor (smoke sensor) monitoring. Data collected is send to the hood DCV controller which control the motorized volume damper for increase or reduction in airflow. Exhaust airflow monitoring is provided thru flowmeter installed in each hood main exhaust connection. Constant balance between the exhaust flowrate and make-up air must be ensure at all times and this is done by hood DCV controller interface with the BMS controller, with provision of correct programming and algorithm to modulate the make-up air VAV Boxes to the hood. The BMS in turn commands the AHU controller to adjust the AHU fan speed due to the reduction in make-up air flow.

DCV utilized for Parking Exhaust System, either ducted or with the use of jet fans uses the same principle as for the kitchen DCV except that the make-up air is not commonly used since car parking have large openings for make-up air. Fan speed (3 ACH (Air Change Hour) and 6 ACH) are normally varied to compensate for the demand as determined by CO2 sensor.

Energy savings for DCV used in outdoor air and kitchen ventilation primarily comes from the power savings from chillers, CHW pumping, fan, and AHU operating power due to reduced cooling requirements. DCV for Car Parking exhaust results in power savings mainly for exhaust and jet fans.

6.21 Variable Air Volume (VAV) System

VAV System is an air distribution strategy to reduce the supply of ventilation air to the occupied space during low cooling or heating load due to changes in solar load, occupancy, and equipment heat dissipation. VAV System should be applied for external zones (rooms attached to the building façade or floor attached to roof slabs) or when occupancy and equipment heat dissipation are cyclically changing. In VAV System application with occupancies, types of occupancy must be properly zone since ASHRAE 62.1 requires strict compliance to the amount of outdoor air to be delivered in the space per occupant especially in reduced supply air flow rate. Multiple Assembly type of occupancy (which requires 2.8L/s per person) must not be mixed with Art Classroom type of occupancy (which requires 9.5 L/s), since the outdoor air to supply air ratio of occupancy with highest requirement must be used in determining the total amount of outdoor air to be supplied to the AHU. Improper zoning result in tremendous increase in outdoor air resulting in increase in energy consumption to treat and cool the increased outdoor air requirement, which is counter-intuitive to the purpose of VAV System. Energy optimization for VAV System comes mainly from the fan power savings.

For clean rooms and other application with requirements for “minimum air changes” and “directional control”, the designer must take caution of the difference between the calculated airflow using the room volume and air-change to that of the calculated air flow due to cooling load. If the resulting total airflow due to minimum air-change is higher or close to cooling load total air flow, the VAV System must not be used since there is no opportunity for fan power savings.

Concerning energy optimization, room zoning using VAV System poses a tremendous challenge for the HVAC Designer and Architects. In actual practice, proper zoning is always unconsidered during design due to complications especially in healthcare and other type of occupancy where FLS (Fire and Life Safety) Codes requires separate zoning and Engineered Smoke Control System.

6.22 Thermal Storage System - Ice Storage and Chilled Water Storage System

Thermal Storage System works in the principle of adding or removing heat to the storing medium for use in other time. During the nighttime where solar load is at the lowest (so as the building cooling load) or there is substantial reduction in cooling load, the CHW System shifts the load to the thermal storing medium so that all of the system chillers remain running at full load. During the day that cooling is at peak, thermally stored energy is used for auxiliary cooling. Application of the technique and energy optimization depends on the characteristic of the building load and type of chiller used. Selecting this technique without fully understanding the above mentioned consideration will result in energy penalty. The following sections explains what the designer should consider:

1. Characteristic of the Building Load – nowadays some building owners and engineers are aware that increasing the building mass decreases the required cooling/heating requirements and the magnitude of energy savings depends on the type of occupation and time of use. A 24-hr operating, high thermal mass building has stable 24 hr. thermal load profile while a 10-hr. operating, high thermal mass building has dip in thermal load profile at late night time. A 24-hr low thermal mass building has dip in thermal load profile once the sun set. Projects with stable thermal profile must not employ the use of Thermal Storage technique.
2. Type of chiller used – as discussed in section 6.9, VSD (Variable Speed Drive) centrifugal chiller is lower in efficiency in full load compare to its FSD (Fixed Speed Drive) counterparts, while the VSD type is very efficient in part load compare to FSD. Theoretically, FSD is required for system that used Thermal Storage System but this is not the actual case. Good HVAC designer knows the problem with low-delta T and this forces HVAC designer to use VSD Centrifugal Chillers (especially for district cooling) since chillers will always perform at part load although there is sufficient load to run at full load. See section 6.1 for the proposed solution to this problem using PICV.

Thermal storage generally is useful if the cost of electricity at night is cheaper than daytime and might not be applicable for the KSA. Just as the Night-Precooling Technique, it is used to shift chiller cooling load from daytime to nighttime, flattening the energy consumption of the building during early peak cooling hours. It also reduce the size and capacity of electrical services since Thermal Storage do not require electricity, thus lowering the project first cost.

Space is a major consideration for locating large tanks. Ice Storage has smaller tank footprint since water latent capacity is used as advantage for thermal storage but poses tremendous challenge to the designer due to thermal expansion of water and ice. Water expands from temperature below 4°C and greatly expands when tuning to ice.

6.23 Air and Water Economizer

Air economizer pertains to the arrangement of duct, dampers, sensors, equipment, and controls in air distribution system to utilize cold outdoor air and by-pass ducting and equipment resistances during cold or moderate climate to eliminate or reduce need for mechanical cooling, thus saving energy.

Water economizer pertains to the arrangement of piping, motorized control valve, sensors, equipment, heat exchanger, and controls in hydronic distribution system to utilize low outdoor air dry-bulb or low wet-bulb temperature during cold, moderate, and dry climate to eliminate or reduce need for mechanical cooling, thus saving energy. Water economizer can be build-in to water cooled condenser chillers equipped with free-cooling system (see section 6.13 for explanation) or a separate system for air-cooled chiller.

6.24 Avoiding Fixed High Resistance Ancillaries in Air Distribution Index Point

In hospital projects where it is customary to located Hepa Filters in air terminals (or air outlets) as required by ASHRAE Application Handbook, it is advisable to locate the filter in the discharge side of the supply fan internal to the AHU in certain case for the reason of power savings. For example, a Centralized AHU is to serve a zone where majority are normal Patient Rooms (with low ACH - Air Change per Hour requirements) and Protective Isolation Rooms (where Hepa Filter is required), and all rooms are located in the façade with high solar heat gain so that the calculated AHU minimum airflow can reach the motor minimum speed (from 25% to 30% of the full speed). If the Hepa filter is to be located at the outlet of the terminal serving the Isolation Room, then the setting of the pressure sensor located at ¾ of the duct circuit will be appx. 350 to 400 pascals (note that newly installed Hepa Filter has pressure drop of 125 to 150 pascal and replacement is needed when it reaches 300 pascal hence pressure sensor is set at this condition plus the pressure drop from the point where the sensor is installed up to the air outlet of the index branch). If the total pressure drop of the system is 1000 pascal (including AHU internal pressure drop), then the settings will limit the fan speed reduction to 55% of the full speed as shown in the performance curve below (Fig. 46), using fan affinity law (e.g. pressure is directly proportional to the square of the speed).

If the Hepa Filter is located internal to the AHU (or in the main duct closer to the AHU), then the pressure sensor setting will be from 50 pa to 100 pascals. The new setting will result in full range of speed adjustment from 100% speed to 25% of the full speed, resulting in 33% fan energy savings.

This technique is not applicable in healthcare if majority of the rooms served by the Centralized AHU have high minimum ACH (such as Surgical Rooms, Trauma Room, Nursery Suite, Procedure Rooms, Triage, Emergency Waiting Rooms, Isolation Rooms, Radiology Room, etc.). In this case, either placing the pressure sensor immediately in the discharge or close to the index point has no difference concerning power consumption since the minimum ACH will limit the speed range adjustment even the pressure sensor is located close to the index point.

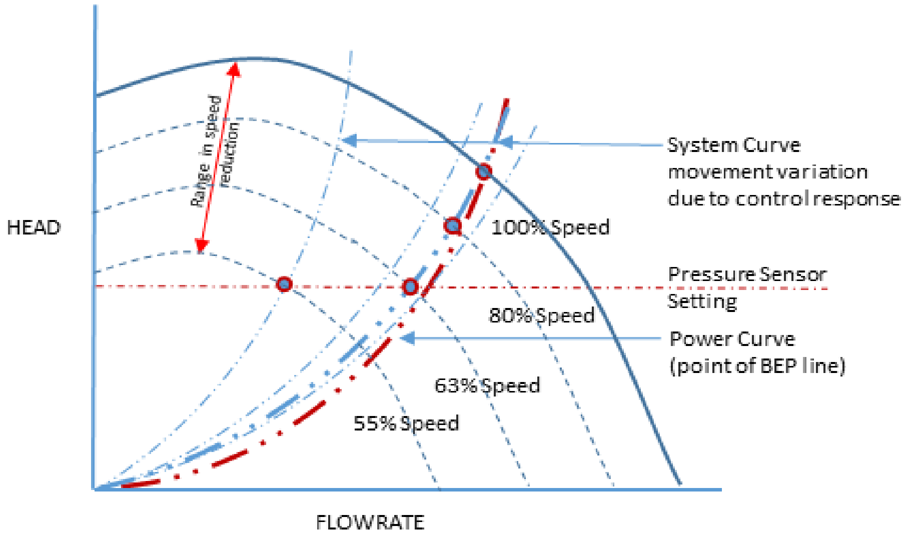


Fig. 46 – AHU Fan (centrifugal airfoil) Performance when Hepa Filter is located close to the Index Point

6.25 Selecting the Proper Head for AHU, Pumps, and Fans

Designers customarily increase the head of AHU fans, pumps, and other fans during the initial stage of the design to calculate the power requirement of mechanical equipment for the electrical team to progress with the electrical design. The mechanical equipment power has no basis for head calculation since the ducting plan or piping circuit is not yet finalized. Every design firm knows that the last stage of the design (from 90% to 100%) is very critical for mechanical and electrical team to adjust all loads and finalize the equipment schedule. This procedure in design is always neglected specially in small design firms to beat submission schedule resulting in highly oversized equipment for mechanical and electrical systems and larger cables. This highly oversized equipment for mechanical and electrical system results in higher project capital expenses, and building services system inefficient performance.

During construction, it is always a “best practice” to recalculate the equipment head based on the layout of piping and ducting as shown in the approved shop drawing to maximize power consumption and reduce the loading in the electrical system. Unfortunately, this “best practice” is always not implemented and the equipment are procured using the contract information to avoid deductive variation. When fans and pumps are selected at the best market available efficiency, the indicated efficiency is based on a given flowrate and head and this point is called the BEP (Best Efficiency Point). Deviating away from the BEP results in degrading of mechanical equipment efficiency.

As shown in the below example (Figure 47), the BEP at full speed based on design information (excessive head) has 70% pump efficiency. Correcting the efficiency based on actual head results in System Curve shifting to the right of the Power Curve results in 65% efficiency. Recalculation and using the correct head does not only result in higher efficiency (or lower power consumption) but savings in equipment first cost due to smaller footprint.

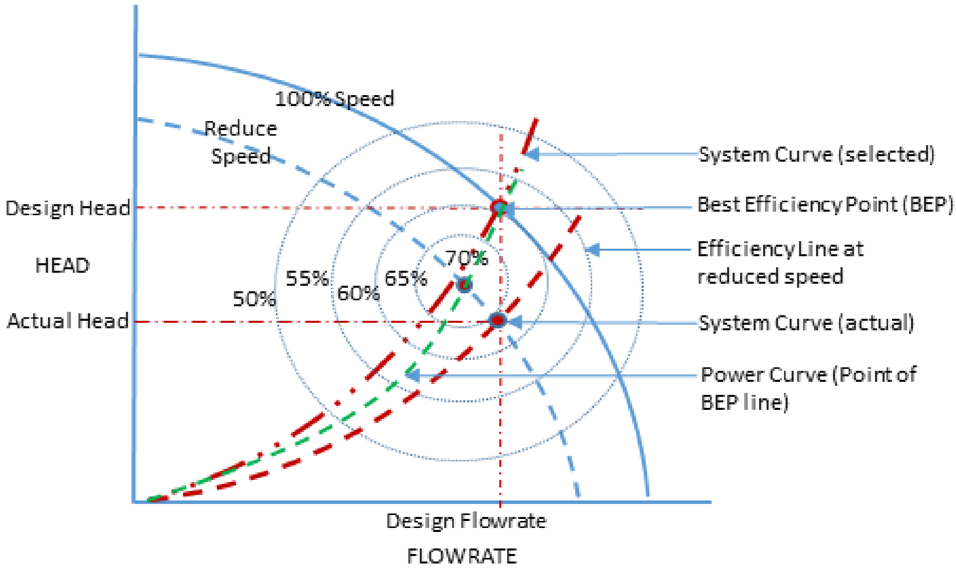


Fig. 47 – Effect of Excessive Oversizing of Pump

6.26 Magnetic Induction Lighting (MIL) and LED Lighting System

According to US CBECS (Commercial Building Energy Consumption Survey), 10% of the building power is consumed by lighting system as of 2012 in the United States which is dominated by standard (tube) fluorescent lighting (see Fig.48). MIL lamps differs from standard fluorescent lamps since it uses electro-magnetic fields through electro-magnets instead of electrodes (or filaments) to excite mercury particles and produce UV lights. MIL lamps have the following advantages over standard fluorescent and other lamp types:

1. Lower first cost resulting on lower project capital expenses.
2. Higher efficiency (efficacy) in lumens per watts (see Fig. 49 from American Green Power). Efficacy is about half of tube fluorescent which means that there is opportunity to reduce the power consumed by lighting appx. by another 5%. Since the lamp has high efficacy, lamps are colder and reduces cooling load due to light radiation and heat convection.
3. Longest life span compare to other bulb types (see Fig. 50 from American Green Power). Lifespan is double compare to that of LED lighting.
4. Lower maintenance cost since replacement is cheaper (see Fig. 51 from American Green Power).
5. Lower carbon emission and lower mercury consumption compare to tube and compact fluorescent lighting.

The architect shall consider selection of lighting lamp types based on building requirements especially when it comes to architectural lighting effects, which LED lighting takes advantage over the MIL.

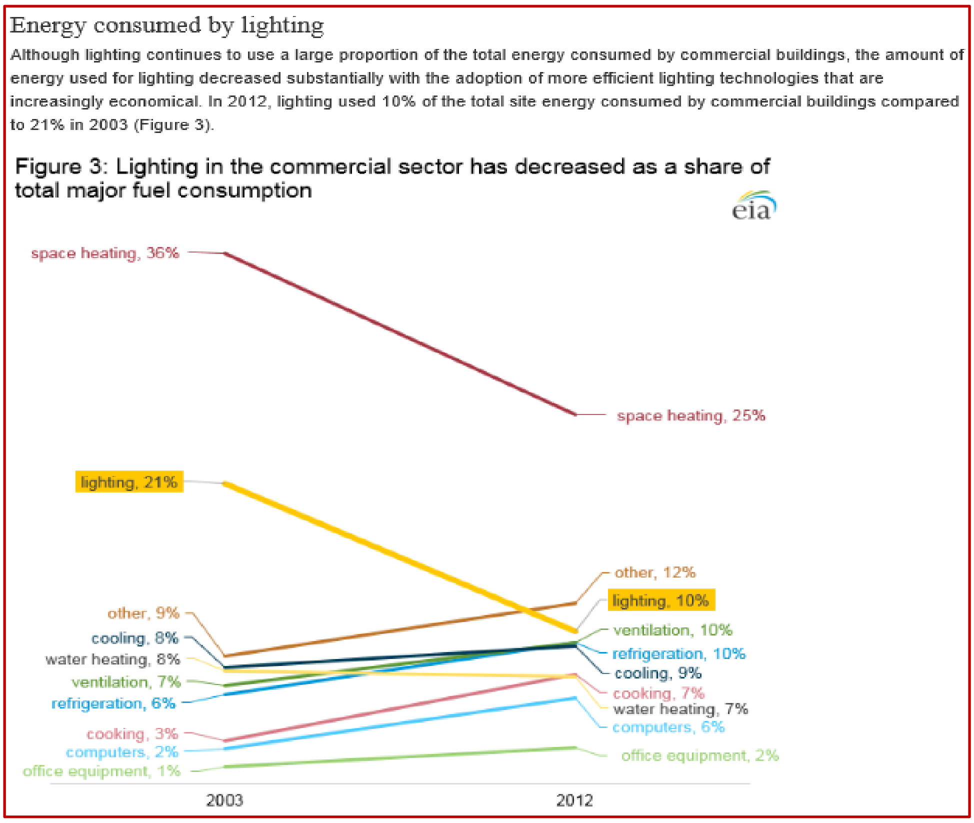


Fig. 48 CBECS Survey for lighting power consumption from 2003 to 2012 as percentage of total building consumption

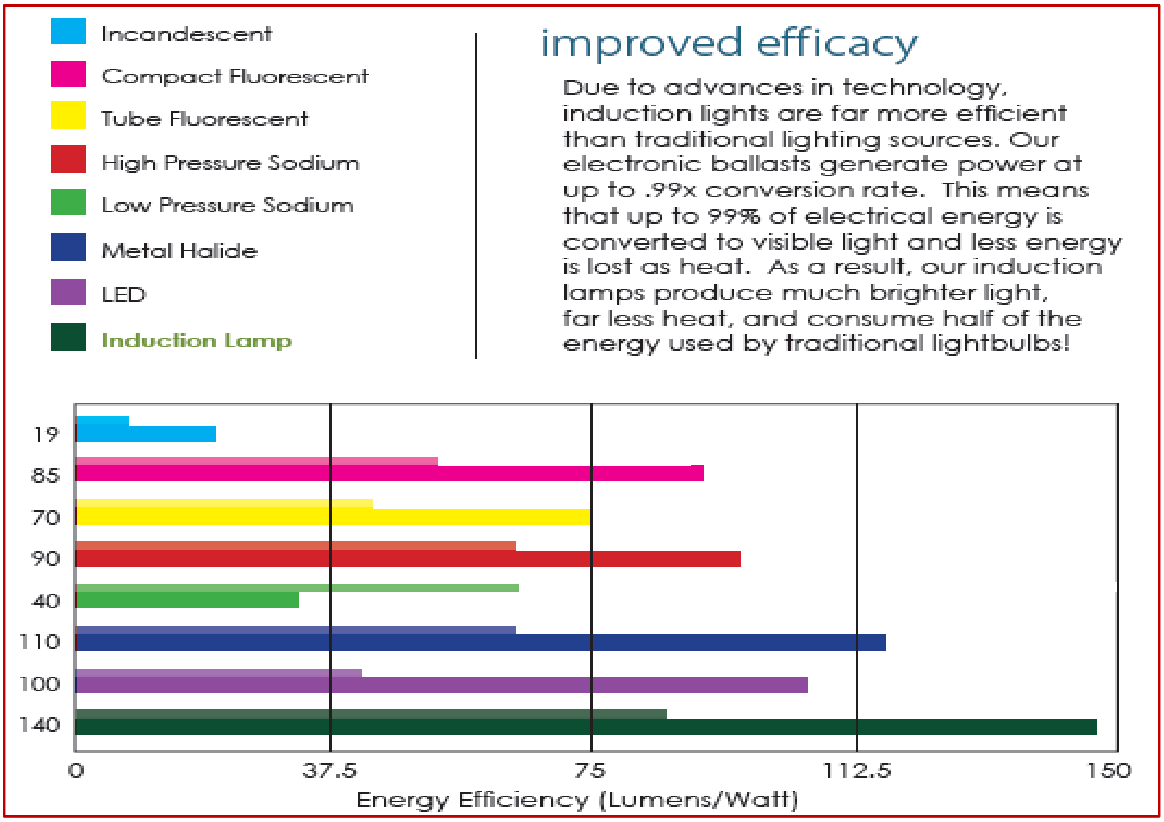


Fig. 49 American Green Power - Various Lamp Efficacy

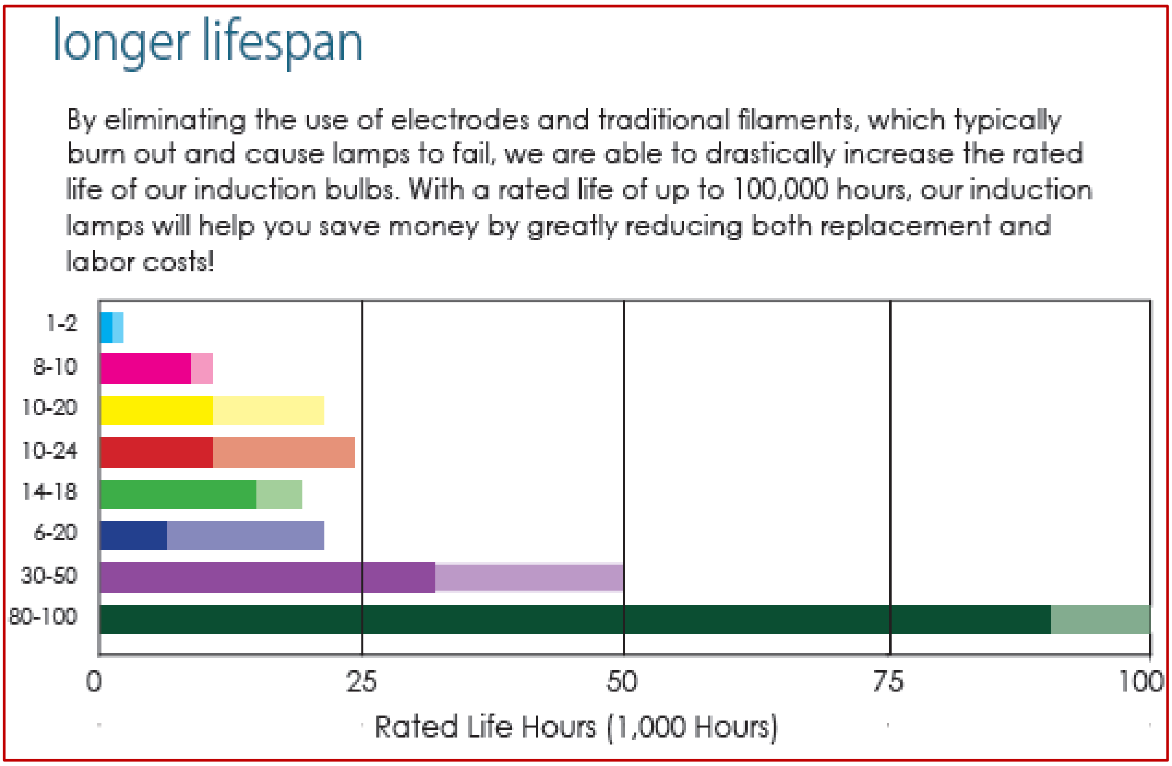


Fig. 50 American Green Power - Various Lamp Life-Span

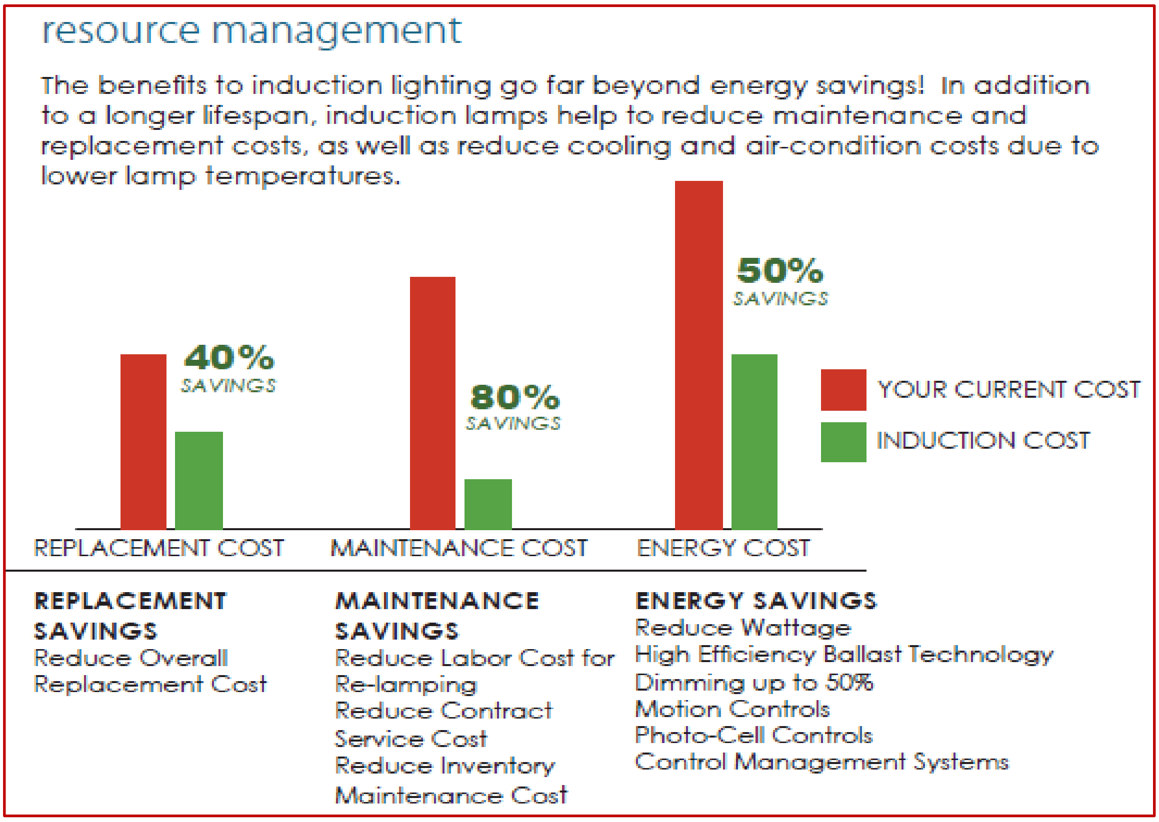


Fig. 51 American Green Power- Lamp Life Maintenance and Replacement Cost

6.27 Daylight Harvesting Technique (DHT)

DHT is a technique use to utilize natural lighting for indoor building use during the day by any or combination of the following method:

1. Use of glazing material with lower U-value, lower SHGC (Sensible Heat Gain Coefficient), higher SC (Shading Coefficient), lower RHG (Relative Heat Gain), higher VT (Visible Transmittance), and higher LSG Ratio (Light to Solar Gain Ratio). Higher LSG glass allows passage of visible light but blocking most of solar radiation and convective heat. Latest technological advancement made this glass characteristic possible. Applicable especially in west, east, and south exposure where solar load is greater.
2. Use of “Smart Glass” (Electro-chromic glass) which changes from clear to translucent to opaque and vise-versa upon application of low electric voltage, or maintaining visible light transmittance throughout varying solar load. Applicable especially in west, east, and south exposure where solar load is higher.
3. Integrated Design - In coordination with the architect and with the use of high LSG glass or “Smart Glass”, locate occupancy which requires higher lighting intensity close to the glass façade and lower light intensity occupancy at the inner location.
4. Use of LSG glass, “Smart Glass”, or standard glazing for north exposure.
5. Use of “Light Conveyors” or “Solar Tracking Skylight” as shown in Figure 52 below.
6. Integration of electric lighting system to DHT sensors and Building Automation System in case of low solar radiation.

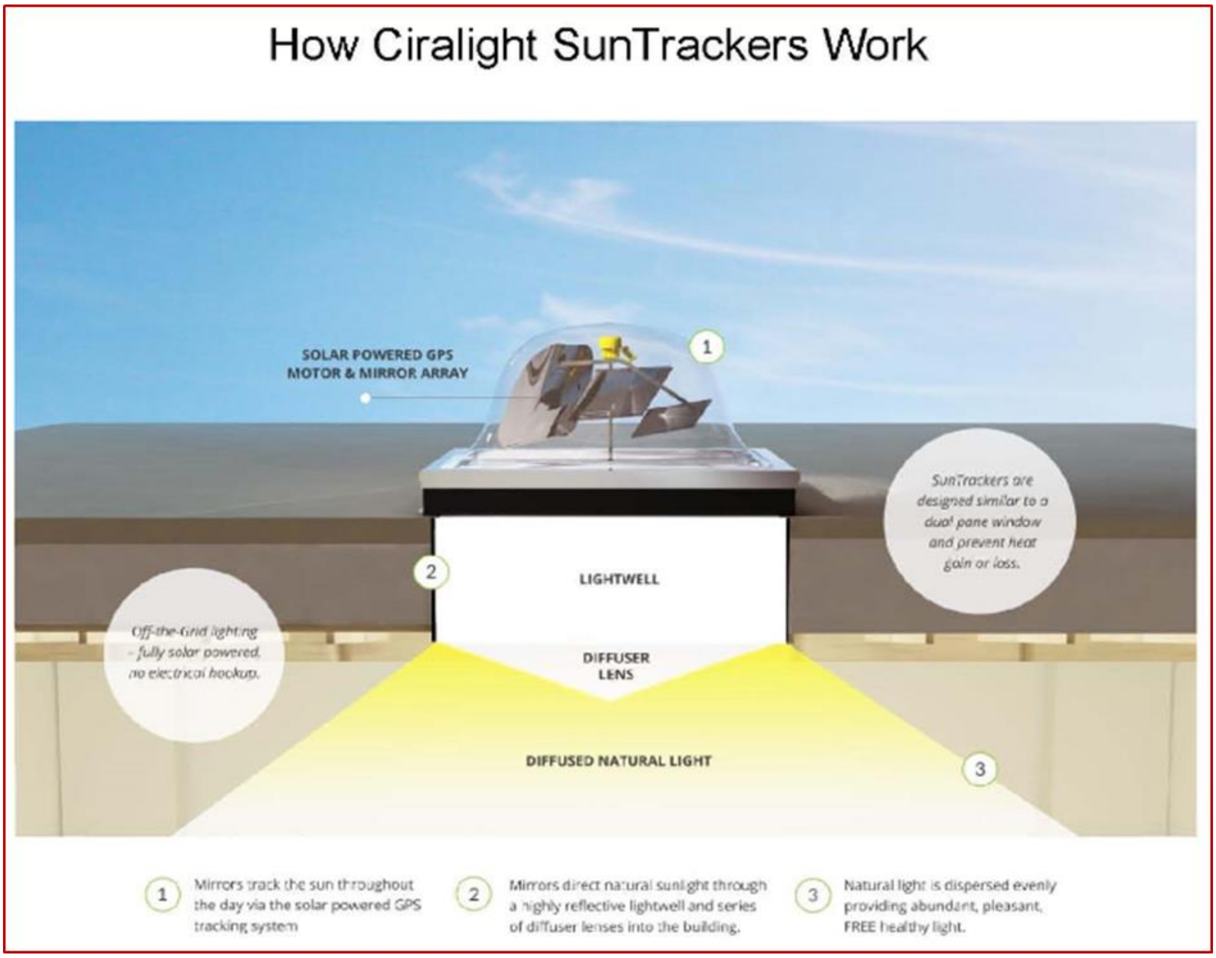


Fig. 52 Ciralight Sun Tracking Skylight

6.28 Building Automation System Controls Optimization

This section applies to control strategy that can be used for HVAC system and equipment for the purpose of energy savings optimization:

1. Reset Control

   A PI (Proportional-Integral) algorithm programed to the BMS controller where the set-point is reset to a new parameter (called desired control point) based on a change in the reference input variable within a time sampling interval. Reset control strategy is used but not limited in the following application for energy optimization;

   a. Chilled Water Supply Temperature (CWST) Reset

      As discussed in section 6.11, chiller efficiency increases as the chilled water supply increases while delta T is maintained. Reduction in building cooling requirement can best be represented by the return chilled water temperature, or by the outdoor temperature since most of the building load is from solar heat gain. Use of reset based on chilled water return or outdoor temperature is based on thermal-coupling characteristics of the building and chilled water circuit design. For a constant flow chilled water system and large thermal mass building, the outdoor air temperature has no relationship with the building load and chilled water supply temperature reset based on chilled water return temperature will be the best option. If the return chilled water temperature based reset is used for constant flow circuit, lower returning CHW results in BMS controller increasing the supply CHW temperature and maintaining the balance between the two parameters to have a constant returning chilled water temperature.

      If the outdoor (ambient) air temperature reset is used, lower outdoor temperature results in increased supply CHW temperature until thermodynamic balance is met. This technique results in significant energy savings but affects the %RH of the building. HVAC designers must consider the maximum allowable tolerance for relative humidity as dictated by ASHRAE for non-critical application, and must not be exceeded. Relative Humidity override can be incorporated into the controls to ensure that the higher allowable limit of %RH will not be exceeded.

   b. Cold Deck Temperature (CDT) and Hot Deck Temperature (HDT) Reset

      Off-coil temperature of mixed-air recirculating cooling coils designed for 24°C and 50%RH comfort cooling are set at 13°C and this is based on peak cooling demand when the solar heat gain is at peak. When the outdoor temperature falls, off-coil temperature setting can be increased to reduce the cooling requirement which results in savings of considerable amount of energy. Since the off-coil temperature is increased then the dehumidification capability of the coil is decreased. The room relative humidity is affected and it is required that the BMS controller be programmed with %RH override to ensure that the higher allowable limit of %RH will not be exceeded. HDT Reset operates in the same principle as the CDT and is applicable for heating system during winter, and applicable for external room zone mostly with low thermal mass.

   c. Mixed-Air Temperature (MAT) Reset

      For Mixed-Air AHU used for location with dual season considering that the building is properly zoned, the temperature of the mixed air (between return and outdoor air) is set at or close to 13°C during winter when the outdoor condition is below 13°C, thus eliminating or reducing cooling requirements for internal room zones. This is done by increasing the outdoor air quantity and exhausting most of the returning air. The technique is best suited for application with tolerable fluctuation in %RH since increasing the outdoor air during winter will reduce room %RH at constant humidification.

      For the external zones especially for low thermal mass façade (e.g. made of glass), heating will be required during winter. Increasing the MAT by reducing the amount of outdoor air during low occupancy (especially in dense occupancy type) will yield to energy savings since heating and humidification requirement will be reduced. Strategy to determine reduction or increase in occupancy can be done in many ways such as installing a C02 sensor in the main return branch or selective sub-branches.

   d. Condenser Water Temperature (CWT) Reset for Cooling Tower VFD Controlled Fan

As discussed in section 6.11, the lower the condenser water temperature going to the chiller then the chiller gain efficiency (or require less power to generate same cooling capacity). Cooling Tower basin water is usually set at 29.5°C controlled by the tower VFD fan. When the ambient air wet-bulb temperature is low enough, then the basin water temperature sensor setting can be reset at lower value to increase the efficiency of chiller, thus decreasing power consumption.

e. Differential Pressure (DP) Reset

As discussed in section 6.1 and 6.2, locating the DPS/T in the index maximized the opportunity to increase the range of speed change for the CHW pump, thus saving energy. Using PICV, lower differential pressure (0.6 bar) can be set for the DPS/T. Using the affinity law, a 4 bar hydronic circuit will yield a minimum 39% pump speed whereas pumps nowadays can run down to 25% of the maximum speed without causing any problem. The concept of DP Reset is to reset the 0.6 bar setting as low as 0.25 bar when all index branches are in low cooling load (low flow). Remember that the 0.6 bar setting is assumed that the actual index circuit is running at full load, which is not always the case.

Applying this technique will require monitoring of (1) prospective indexes control valve positions; and (2) CHW delta T across the coils, and coil differential pressures (as representation of the flow) to calculate the actual partial cooling load of the coils in all prospective indexes.

f. Duct Static Pressure (DSP) Reset

This technique is only applicable for Variable Air Volume distribution system with high resistance ancillaries such as Hepa Filters installed in the supply ducting. Hepa Filters when newly installed has 125-150 pascals pressure drop and needs to be replaced when it reaches 300 pascals. Filter installed in one index point can be measured and monitor to become the basis for Static Pressure Reset to maximize range of speed reduction for fans to reduce power consumption.

2. Space Temperature Set-back Control

Applicable for non-critical space temperature application (such as high-end hotels, healthcare rooms, medicine factories, semi-conductor facilities, and other clean room application, etc.) which are not allowed to be turned-off but allows space temperature to be increased during non-occupied period for power savings. Technique requires occupancy sensors for successful automatic control implementation.

3. Air Flow Set-back Control

Applicable for application with minimum air change requirement and directional flow control (such as operating rooms, healthcare isolation rooms, medicine factories, other clean room application, etc.) which are not allowed to be turned-off but Standards allows reduce air flow during non-occupied period for power savings. Technique requires occupancy sensors for successful automatic control implementation.

4. BMS-HVAC System and Equipment Optimization

Optimization in HVAC System pertains to the method of operating the system and equipment in the most energy efficient manner. Optimization starts with “controller fine tuning” where PID (Proportional-Integral-Derivative) algorithm gains, parameters, set-points, and timings (time sampling) for a particular equipment in a system are set, and followed by system controllers. The intention of the “fine tuning” is to ensure that the equipment and controller operates and response in a quick, stable, predictable, and repeatable manner, thus eliminating any off-set from the set-point (or control point) in the shortest period of time. A “fine-tuned” controller eliminates quickly the unnecessary power used by equipment running in excess of the set-point.

When equipment are to perform in a system to obtain certain operational functions and to optimize energy consumption, the BMS Sequence of Operation must address the following sample questions;

a. What combination of chiller type and capacity will give the best energy optimization from start-up, part-loading, and full loading basing in their operating (efficiency) characteristics?

b. Based on the building thermal mass characteristics, what reset technique will be beneficial? Based on outdoor air temperature or return chilled water temperature?

c. If Night Precooling is used, what time should the equipment start?

d. What differential pressure value and deviation from the set value (above and below) should be used in chilled water system?

e. What temperature (dry-bulb and wet-bulb) shall the economizer be set to start free-cooling for water cooled condenser chillers? What ambient temperature shall air economizer start?

f. What set-back control value best correspond to the change of monitored value? Is the response of the monitored value linear or equal-percentage to the change in controlled value for accurate and fast response?

g. What outdoor temperature shall the energy recovery by-pass should open during winter and summer?

These are some of the questions that must be addressed during T&C to “optimized” operation of the HVAC System and Equipment. Data to substantiate controller operation must be based on calculation, software analysis result, or proper engineering judgement based on previous project experience.

6.29 Testing and Commissioning

During the Testing and Commissioning stage, certain activities are needed to be implemented strictly according to the standards to ensure that means of wasting of energy are avoided. T&C activities that affects power consumption are as follows:

1. Air duct leakages – HVAC standards mandates certain percentage of leakage for air ducting which are as follows:

Duct Pressure Class	Static Pressure Limit		Allowable Air Leakage (% of total air flow)
	Positive	Negative	
Class A – Low Pressure	500 Pa	500 Pa	6%
Class B – Medium Pressure	1000 Pa	750 Pa	3%
Class C – High Pressure	2000 Pa	Above 750 Pa	2%

2. Leakage above this limit increases AHU fan, chiller, and pump power consumption since above ceiling spaces, duct shafts are unnecessarily cooled by leakage air. Duct sealant class shall coincide with the duct pressure class rating to avoid excessive leakages.

3. Piping Leakages – pipes must be hydraulically tested to the required pressure (see T&C document EPM-KT0-GL-000003 Pre-commissioning Stage) to avoid even minor leakages which can result to pump activation and hunting.

4. For Non-Pressure Independent air and water distribution system, air and hydronic balancing must be conducted and deviation of terminal flowrate must be within the tolerance. Unbalanced distribution results in excessive pump and fan power. Pressure Independent Systems (with the use of PICV for hydronic system and VAV for air distribution) automatically provide balancing which results in energy savings. Future rebalancing required for Non-Pressure Independent System are also avoided which is an operational cost savings.

5. Building envelop must be air leak tested to ensure building façade tightness and reduce infiltration or air pressurization requirements especially for hot and humid climate.

6. Building Management System must be properly commissioned, fine-tuned, and optimized. Control algorithm must be verified that properly functioning as per the accepted control strategy. Refer to BMS Control Optimization Strategy section 6.28.

7. HVAC System, equipment, and the BMS controller must be properly tested and commissioned, finely tuned, and optimized.

6.30 Post-Occupancy Commissioning

1. Ensure balance in electrical phases to avoid unwanted current flowing in neutrals and to the phases resulting in energy losses in terms of heat (over heating of inductive loads such as motors and transformers).
2. Ensure THD (Total Harmonic Distortion) is treated. THD results in energy loss in terms of heat (over heating inductive loads such as motors and transformers).
3. Ensure proper cable termination of electrical and mechanical equipment to avoid over heating which is a loss of power. Conduct thermal scanning to verify.
4. Provide proper maintenance of equipment such as regular replacement of filters, regular lubrication of moving/rotating equipment, periodical replacement of fan belts, etc. to maintain equipment efficiency.
5. Conduct re-commissioning as recommended by Codes and Standards or by the Commissioning Authority recommendation especially on duct leakages, electrical phases balance, harmonics, hydronic and air system balance.
6. With the use of energy meters, determine deterioration of equipment due to wear and tear and decide if replacement will be required based on life-cycle cost analysis.

6.31 Other Energy Saving Techniques

1. Architectural design must comply or exceed ASHRAE 90.1 (Energy Standard for Buildings except Low Rise Residential Building) Energy Cost Budget (ECB) method or Performance Rating Criteria (PRC) minimum requirements such as:
   a. Building envelop vertical fenestration should not exceed 40% of the gross wall area of the spaces to be air-conditioned with U-value, SHGC (Solar Heat Gain Coefficient), and LSG (Light to Solar Gain) Ratio compliance to the minimum requirements based on climate zoning.
   b. Building sky-light fenestration should not exceed 3% of the gross roof area for spaces to be air-conditioned with U-value, SHGC, and LSG Ratio compliance to the minimum requirements based on climate zoning.
   c. Provision of ante-room between areas of unconditioned and conditioned space.
   d. Provision of external and internal shades.
   e. Leakages in building envelop, fenestration, and external doors shall comply with the minimum requirements. Improve method of sealing façade joints especially when glazing is used to reduce infiltration.
   f. Building envelop (walls, curtain walls, roof slabs, doors, inclusive of insulation) shall comply with minimum U-value for each component and over-all U-values for assemblies.
   g. Building geometry- orienting the building where the longest side faces the north as possible while considering prevailing winds (which affect infiltration, use of natural and cross ventilation, and daylight harvesting) and minimizing fenestration exposure in south, west, and east. Computer modelling can help to identify best option for floor plate configuration and building orientation relative to prevailing air direction.
   h. Increasing the building thermal mass.
   i. Light fixture type shall be selected to comply with the maximum lighting density in watts/m2 for interior and exterior lighting.
2. Mechanical design must comply or exceed ASHRAE 90.1 (Energy Standard for Buildings except Low Rise Residential Building) Energy Cost Budget (ECB) method or Performance Rating Criteria (PRC) minimum requirements such as:
   a. Mechanical equipment shall comply with minimum efficiencies at Standard and Non-Standard Rating Operating Condition.
   b. HVAC System to be selected must be best suited based on building thermal characteristics, thermal load profile, and project meteorological data.

c. HVAC System must comply with the requirement of ECB or exceed ASHRAE 90.1 requirements based on Performance Rating Criteria requirements.

d. Simultaneous cooling and electrical heating for humidity control must be avoided.

e. Duct sealing and leakages shall comply with minimum requirements.

f. Motorized damper leakages complies with the minimum requirements of Standard AMCA 500.

g. Duct and piping insulation thickness and R-values shall comply with minimum requirements for internal and external installation.

h. External duct and piping shall comply with cladding requirements to shield from solar radiation.

i. Implementation of Building Automation System to automatically start and shut-down based on scheduled timing and provide control for part-loading.

j. Implementation of control algorithm for reset controls, interlocks, overrides, and equipment protection.

k. Implementation of Optimum Start Controllers (Low Voltage Starters) for large HVAC and other building equipment.

l. Use of Energy Recovery Equipment with effectiveness compliance to the minimum requirements.

m. Use of air and water economizer with pressure drop complying with the maximum pressure drop.

n. Fan and Pump motor shall be selected compliance to the minimum nominal full-load efficiency.

o. Service hot water heating equipment efficiencies shall comply with the minimum requirements. Heating operational and safety controls and parameters shall comply with the minimum requirements.

3. Electrical design must comply or exceed ASHRAE 90.1 (Energy Standard for Buildings except Low Rise Residential Building) Energy Cost Budget (ECB) method or Performance Rating Criteria (PRC) minimum requirements such as:

   a. Feeder conductor maximum voltage drop shall not exceed 2% at design load.

   b. Branch circuit conductor maximum voltage drop shall not exceed 3% at design load.

   c. Percentage of receptacle covered by automatic control shall comply with minimum requirements except receptacles for use of equipment requiring continuous operation and if automatic control will compromise occupant safety or security.

   d. Energy metering shall comply with the requirements with the exception as stated in the Standard.

   e. Low Voltage (≤ 600 volts) Dry Type Distribution Transformers shall comply with the minimum efficiency and requirements of Energy Policy Act of 2005.

4. Energy Optimization related to Architectural Design:

   a. Use of LED Lighting and Magnetic Induction Lighting (MIL) - see section 6.26. MIL shall be preferred over LED for general application except in some special application requiring architectural lighting effects. MIL first cost is lower than LED, life expectancy is higher, and power consumption is half of that LED lighting.

   b. Reduce the percentage of fenestration and glazing. Build block walls at the back of glazing to reduce solar load and increase thermal mass.

   c. Specifying low water consuming or waterless plumbing fixtures, thus reducing flowrate requires for domestic water and reduce water and power consumption.

   d. Specifying low pressure plumbing fixtures to reduce power drawn by pumps.

   e. Specifying low power consuming appliances (computers, house hold appliance, and other electronic equipment).

   f. Avoiding of water features inside the building especially for humid climates.

g. Design building to maximize natural ventilation especially during mid-season by stack and cross ventilation (by using operable windows).

h. Design building to maximize natural ventilation and lighting for car parking. Use open and above ground parking as much as possible and take advantage of stack effect and cross ventilation.

i. Providing thermal breaks for external structures like balcony, window frames, and external shades.

j. Provide greeneries in roofs and façade to reflect and absorb solar radiation.

k. Use of thermal paints with light color for building façade to increase reflectance of radiant heat and reduce conductive heat gain.

l. Utilized daylight harvesting technique (see section 6.27)

5. Energy Optimization related to Mechanical Design:

a. Utilization of the energy saving techniques covered in this document.

b. Specifying high efficiency mechanical equipment (pumps, fans, chillers, Dx Units, heat pumps, etc.) available in the market.

c. Use of solar hot water system for domestic water heating instead of electric water heaters.

d. Use of round ducting or square ducting to reduce friction loss in air distribution.

e. Use centralized cooling system (chillers and CHW system, VRV (Variable Refrigerant Volume) Direct Expansion System, centralized AHU and air distribution system, etc.) instead of small package units and split type units for medium to large scale projects. Central HVAC equipment has lower KWe/ton (Kw of electricity required per nominal ton produced) compare to smaller units. For example, modern FSD Centrifugal Chillers has 0.8 KWe/ton at standard full-load condition compare to 1.2 KWe/ton for split-units. Best industry IPLV (Integrated Part Load Value) of modern ASD Centrifugal chillers can be as low as 0.2 KWe/ton compared to 0.45 KWe/ton for screw chillers.

f. Avoid hydronic air removal system that requires to be installed in the CHW System main line. Use Reflex atmospheric or vacuum de-aeration system which is a combined air removal, expansion vessel, and make-up water system.

g. Use steam in laundry equipment heating instead of electric equipment thereby reducing power consumption. Cost of fuel for boiler is much cheaper compare to processed/distributed electricity.

h. Use LPG or LNG (Liquefied Natural Gas) for kitchen equipment instead of electric equipment thereby reducing power consumption. Cost of LPG or LNG is much cheaper compare to processed /distributed electricity.

i. Use renewable energy instead of municipality grid supplied power sources (photo-voltaic, air turbines, biomass, and other renewable energy)

j. Specifying ECM (Electronically Commutated Motors) starters for small fans and pumps.

k. Hot water branch recirculation line shall connect close to the fixture it serve to reduce or eliminate wastage of water to draw hot water, thus saving power for pump.

l. Reduction of supply hot water temperature as the project allows.

m. Increasing the thickness of hot water and chilled water insulation to reduce thermal losses.

n. In coordination with the architect, provide energy modeling to determine best floor-plate form, dimension, and orientation considering solar radiation and wind direction for energy optimization. Energy modelling shall also determine best orientation of fenestrations (glazing) and solid walls considering irradiation (from neighboring buildings and ground) as well as the solar reflections (from glazing) from other buildings for congested urban areas.

6. Energy Optimization related to Electrical Design:

a. Integrate lighting system to automatic control such as BMS for scheduling.

b. Specify high efficiency transformers.

c. Use capacitor banks to compensate inductive loads thereby increasing power factor close to unity and maximize power.

d. Consider photo-voltaic and other renewable electrical power source.

e. Use high voltage power source as possible for larger motor to reduce wiring losses. This technique is widely used in district cooling chillers and pumps.

f. Use of photo-electric sensor and automatic control for outdoor lighting.

g. Provide energy metering to monitor increase of energy consumption over time (due to building material failure and degradation, poor maintenance, and equipment wear & tear).

h. Educate building occupants and with the help of metering system, imposed penalty for building users exceeding maximum allowance for power consumption.

7.0 WATER SAVINGS TECHNIQUE

7.1 Design of Hot Water Recirculation System

One major source of water wastage in hotels, residential apartments, hospitals, and other application using centralized hot water system is the waiting time to draw hot water in faucet and showers. As a standard design guideline, discharging hot water at its design temperature within 10 secs. upon opening a faucet or a shower is acceptable. This is done by routing the hot water mainline as close as possible to the faucets and showers and providing recirculation line with balancing valves for long branches. To reduce or eliminate the time for hot water withdrawal, each shower and faucet must be provided with recirculation and low flow automatic balancing (or temperature balancing valve) with tapping point located immediately at the back of faucets and shower mixing valves. This technique will not only significantly reduce water wastage but will also reduce the power associated with pumping.

7.2 Low Flow, Timed Flush, and Waterless Plumbing Fixtures

1. Water Closets

   Water closet shall be the lowest capacity per flush available in the market and type shall be as follows;

   a. For Tanked Type:

      (1) For women's toilet – dual flush 1.9/3.6 LPF(Liter Per Flush) *(subject to change due to technological advancement)*

      (2) For man's toilet with urinals – single flush 3 LPF*(subject to change due to technological advancement)*

      (3) For man's toilet w/out urinals - dual flush 1.9/3.6 LPF*(subject to change due to technological advancement)*

   Single and Dual- Flush toilet indicated above is the lowest capacity per flush currently available in the market. To minimize water consumption, toilets using waterless or low flow urinals requires single flush WC since it requires less water for full flushing.

   b. For Flush Valve Type:

      (1) For man and women's toilet – 4 LPF *(subject to change due to technological advancement)*

2. Urinals

   For water consumption optimization, waterless urinals will seem to be the best option but actual installation of this type raises a lot of concern regarding foul smell and hygiene. Alternative option is to provide timed flush instead of waterless urinal to reduce water consumption. For highly dense occupancy, low flow urinals (1.9 LPF) with timed-flushed will be the best option.

7.3 Gray Water System and Treated Sewer Effluent (TSE)

Building utilizing double sanitary stack system (soil and waste stack) has opportunity to re-use the gray water (treated) for water closet flushing, irrigation, and other non-potable water usage. Gray water is the waste water produce without fecal matter or with lesser pathogens and requires only secondary waste water treatment for re-use. Gray water source can be from showers, wash basins, washing machines, sinks, etc. Secondary waste water treatment involves chemical reactions and bacteriological processes to remove the suspended organic matter. The treated waste water, normally termed as “Treated Sewer Effluent” or TSE is then pumped back for re-use for the following system;

1. For Water Closet flushing

   Water closet water line used for TSE is separate to the domestic water line used for basins, shower, sinks, and other clean water application.

2. For Irrigation System of greeneries

7.4 Irrigation System Sprinkler Types and Control System

Irrigation uses different type of sprinkler heads which ranges from impact sprinklers, pop-up spray heads, rotor sprinklers, and nozzle sprays. This types of heads creates very fine droplets which increases evaporation, thus large quantity of water is loss to the atmosphere. To eliminate or reduce evaporation in irrigation system, the following should suffice;

1. The contact of water to the air must be reduced by avoiding sprinkler head that breaks the water into fine mist. Use of rotary sprinklers and drippers will be more appropriate.
2. The higher the wind velocity, the greater is the evaporation. Velocity of air must be at minimum during irrigation and air velocity sensor will be required for automatic irrigation to avoid activation during high winds.
3. The irrigation tank must be insulated or shaded to protect the water from gaining heat from solar radiation and to keep it cooler.
4. To eliminate evaporation, the ambient air dew point must be the same or higher compared to the temperature of the irrigation water. Air dew point is advised to be monitored in automatic irrigation system as well as the temperature of the water. The larger the difference in temperature, the greater is the evaporation.

Relative humidity most of the time shoots-up at night as well as the dew point temperature, and drop as the sun rise early in the morning (as well as the dew point temperature). It is for this reason that it is ideally reasonable to irrigate at night and early in the morning before the sun rise to eliminate or reduce evaporation. Rain water sensors shall be used to avoid irrigating greeneries when raining.

7.5 Other Water Saving Technique

1. Energy Optimization related to Architectural Design:
   a. Specify low flow plumbing fixtures with aeration technology for faucets and showers.
   b. Use fake greeneries or greeneries that consumes less water.
   c. Use easy to clean tiles (or finishes) in bathrooms, kitchens, and laundries to reduce water for cleaning.
   d. Specify appliances (washing machines, etc.) that requires less water to operate.
   e. Reduce surface area of water ponds for external and indoor ones.
2. Energy Optimization related to Mechanical Design:
   a. Harvest A/C condensate in humid climate for re-use in plumbing fixtures and irrigation.
   b. Use municipal TSE or local TSE from WWTP (Waste Water Treatment Plant) for irrigation to reduce potable water consumption.
   c. Avoid deaerating type nozzles for water features to reduce evaporation.

d. Provide lower system pressure for domestic water to reduce water consumption.

e. Use washer tunnels with water recycling system for laundry instead of individual washer extractor.

f. Reduce boiler blowdown by providing effective water treatment for make-up water. Blowdown should be recover, cooled, and used for TSE water for irrigation or WC flushing.

g. Use side stream filtration for cooling tower condenser water instead of blow down technique.

h. Ensure proper pressure testing for pipes is conducted during construction to avoid water leakage during operation.

i. Cool indoor water features to reduce or eliminate evaporation especially for high latent load application.

j. Provide water metering. Educate building occupants and imposed penalty for building users exceeding maximum allowance for water consumption.