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OPTIMIZED CONTROL STRATEGIES FOR A TYPICAL WATER LOOP HEAT PUMP SYSTEM

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By

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OPTIMIZED CONTROL STRATEGIES FOR A TYPICAL

WATER LOOP HEAT PUMP SYSTEM

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Water Loop Heat Pump (WLHP) System has been widely utilized in the Heating, Ventilating and Air Conditioning (HVAC) industry for several decades. There is no doubt about the energy saving potential of this type of system from the design and construction perspective. However, there are still many unsolved problems, such as high loop pump energy consumption, low heat pump efficiency and high electricity cost due to improper operation, although the system design intension was to save energy. Thus, how to control the WSHP to realize its energy-efficient value is an innovative but practical topic on which this thesis will focus.

To achieve this goal, three optimal WLHP control strategies will be discussed in this thesis:

(1) Improve the water loop circulation pump control by using a differential pressure reset method;

(2) Optimize the water loop temperature control by resetting the fluid cooler supply temperature;

(3) Modify the building schedule and heat pump fan operation mode to save energy without sacrificing occupant comfort.

These new control strategies were implemented on a middle school building equipped with a very typical WLHP system in a building mechanical system commissioning project. Several experiments were conducted in order to demonstrate the energy savings and other benefits compared with the existing control and operation.

It is concluded from the results of the experiments that the new loop pump control reduced the pump speed by 16.3% on average and has the potential to save 40% to 70% pumping energy; the improved water loop temperature control saved 12.8% of heat pump electricity consumption; and the new HVAC operating schedule and heat pump fan mode are able to achieve 20% to 30% of total electricity savings annually.

ACKNOWLEDGEMENTS

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NOMENCLATURE

English Letter Symbols

$Btuh$	= The heating or cooling load of a particular zone, in Btu per hour;
C	= Differential pressure required by the coil, psi;
COP	= Coefficient of performance;
c	= Specific heat, Btu/lb-°F;
DBT	= Outside air dry bulb temperature, °F;
DP	= Differential pressure, psi;
EER	= Energy Efficiency Ratio;
GPM	= The flow rate to meet the heating or cooling demand, in gallons per minute;
H	= Pump head, psi;
HP	= Heat Pump;
KW	= Heat pump compressor power, KW;
$\overline{m}$	= Loop water mass flow rate lb/h;
N	= Pump speed, rpm;
OAT	= Outside air temperature, °F;
P	= Water pressure, psi;

Q = Building load, Btu/h;

$\bar{Q}$ = Actual flow rate, gpm;

RWT = Return water temperature, also fluid cooler inlet water temperature, °F;

S = Flow rate coefficient;

SWT = Supply water temperature, also fluid cooler outlet water temperature, °F;

T = Temperature, °F;

ΔT = Temperature difference between supply and return water, °F;

v = Pump speed in percentage, %;

WBT = Outside air wet bulb temperature, °F.

Greek Letter Symbols

ω = Pump speed ratio, i.e. actual speed divided by design speed, %;

Δ = Difference, for temperature.

Subscripts

$actual$ = Actual condition

ad = Added

$balanced$ = Balanced condition

c = Cooling

$design$ = Design condition

$\max$	= Maximum value
$\min$	= Minimum value
s	= Supply
$setpoint$	= Setpoint value
r	= Return
h	= Heating
p	= Pressure
0	= Specified condition
1	= Loop differential sensor location
2	= Most remote coil location

Chapter 1 INTRODUCTION

Water loop heat pump (WLHP) systems have gained more and more popularity in commercial buildings and institutional buildings ever since their debut in California more than forty years ago. The WLHP systems are notable for their high efficiency and energy recovery ability. However, improper control and operation will undermine these advantages. Although many research projects related to WLHP systems have been undertaken from 1970s, most of them were concentrated on the system design (Spitler 2005). Not much focus was given to the control and operation of the water loop system. This thesis therefore aims to find out several optimal control strategies for a WLHP system and demonstrate their feasibility through their application on a school building.

In the initial design of a WLHP system, the heat pump water loop was directly connected to heap pump units without using any isolation valves. Later, people added isolation valves to the system so that the valves close when the heat pump compressors are not working. More than twenty years ago, variable frequency drives (VFD) became accepted by the HVAC industry and were widely adopted to control the pump speed corresponding to different water flow rates. Yet how to properly control the pump speed in a WLHP system remains a question for most practitioners and engineers.

In practice, water loop circulation pump speed is often controlled to maintain a constant high loop differential pressure (Delta-P or DP) to ensure the safety of the system with sufficient water flow. The DP is often based on the system design condition. This conservative method not only wastes pumping energy, but also leads to a high-pressure working condition for the control valves and fittings. Leaking problems and valve operating difficulties may occur as a result. In Chapter 3, an improved loop pump control strategy will be discussed and compared with the constant DP setpoint control method. This novel pump control method uses a reset variable DP setpoint determined by the actual flow rate to control the pump speed.

Another issue is related with the water loop temperature control. A common control method is to let the loop temperature float between a high limit and a low limit, usually set at 90 °F and 60 °F. The cooling tower starts to absorb heat from the water loop and reject the heat to the outside atmosphere if the loop temperature is above the high limit, whereas the boiler starts to add heat to the water loop if the loop temperature is below the low limit. This method is fairly easy to implement. However, it is not efficient. The heat pump compressor power consumption is closely related with the heat pump coefficient of performance (COP). In a heat pump vapor compression cycle, the COP increases as the difference or "lift" between the condensing temperature and evaporating temperature decreases. On the water side of a water-to-air heat pump, the heat pump COP increases as the water loop temperature increases in the heating mode since it offers a higher temperature heat source. In the same manner, the COP increases as the loop temperature drops in the cooling mode since the heat sink temperature is lower. In this scenario, the heat pump tends to work more efficiently with a low loop temperature in the summer and with a high loop temperature in the winter. Chapter 4 will provide a method to determine an optimal loop temperature in the summer by properly operating the cooling tower. This optimal loop temperature in the cooling dominated season not only makes the heat pumps run more efficiently, but also saves the circulation pump power.

Other common problems in this type of system are related with the heat pump operating schedule and heat pump fan operating mode. The building operating schedule is an important factor that can influence the total building energy consumption. In the case study building, the existing schedule allowed too much unnecessary hours to operate the heat pumps. In addition, the heat pump fans were set to run continuously no matter if the compressors are activated or not. This can waste much energy especially in the part load seasons when most heat pump compressors work less than two hours in a day. For a detailed study about these problems, improved operating schedule and fan operating mode will be presented and compared with the existing schedule and mode in terms of daily electricity consumption in Chapter 5. A Carrier HAP model will also be created and compared with the actual energy consumption under different schedules and operating modes.

In the final part of the thesis, potential energy savings through the utilization of these optimal control methods will be summarized. Suggestions for future research in this area will also be provided.

Chapter 2 BACKGROUND AND LITERATURE REVIEW

In this chapter, some background information on water loop heat pump (WLHP) and cooling tower /fluid cooler will be given. Meanwhile, a review of several control strategies on WLHP systems will also be presented.

2. 1 Water Loop Heat Pump Fundamentals

2.1.1 History of Heat Pump Systems

The introduction of heat pump is one of the greatest milestones in physics and thermodynamics. The growth in understanding of the refrigeration and heat pump process began after Joule demonstrated the principle that the temperature of the gas can be changed by altering the gas pressure. Piazzi Smythe was the first to propose a cooling machine using this principle. The heat pump theory was first described by William Thomson (also known as Lord Kelvin) in 1852 and the first heat pump was developed by Peter Ritter von Rittinger in 1855 to 1857. After that, for a long period, the development of the heat pump lagged behind that of the refrigeration equipment until the 1920s and 1930s.

One critical progressive period for the heat pump systems was after the 1930s' world economic crisis when a dozen of heat pump systems were applied in schools, hospitals and commercial buildings in Europe and United States. Geothermal heat pump system was developed in Britain and United States around 1950s after the initial introduction of the technology at the Commonwealth Building in Portland, Oregon. In the 1960s, the reversible domestic air-to-air heat pumps became popular in the United States. At the same time, water loop heat pump system was applied first at California.

Following the oil crisis in the early 1970s, heat pump applications increased worldwide and more large systems incorporated heat recovery into their design and operation in the interests of energy conservation. Countries like China, Japan, Sweden and New Zealand have put much effort on the research and application of heat pump systems. In the 21st century, with new problems and concerns such as global warming, CO_2 emissions and global energy tension emerging, heat pump, as an energy efficient and environmental friendly device, continues to contribute itself to the goal of sustainability.

2.1.2 Benefits of Heat Pumps

Direct combustion or heating using electric resistance is never the most efficient way of using energy. Heat pumps are more efficient than electric heating and combustion heating because one share of electric energy can generate 3 to 5 times of heating energy, based on the concept of COP. Also, heat pumps can reduce the amount of CO_2 generated in a combustion process. According to a report from International Energy Agency in 2004 (IEA 2004), the potential CO_2 reduction achieved by heat pumps can be 6.6 billion tons of CO_2 for heating buildings. In addition, 1.0 billion tons can be saved by residential and commercial heat pumps, assuming that they can provide 30% of the heating for buildings, with an emission reduction of as much as 50%. Heat pumps are also known by their ability to use low temperature heat and renewable energy. These features make them very attractive to property owners, designers and engineers.

2.1. 3 Heat Pump Theories

A heat pump is a machine or device that extracts heat from a source at a lower temperature and transfers it to a sink at a higher temperature by using mechanical work or a high-temperature heat source. Most modern heat pumps use a refrigeration cycle like the one shown in **Fig. 2-1**.

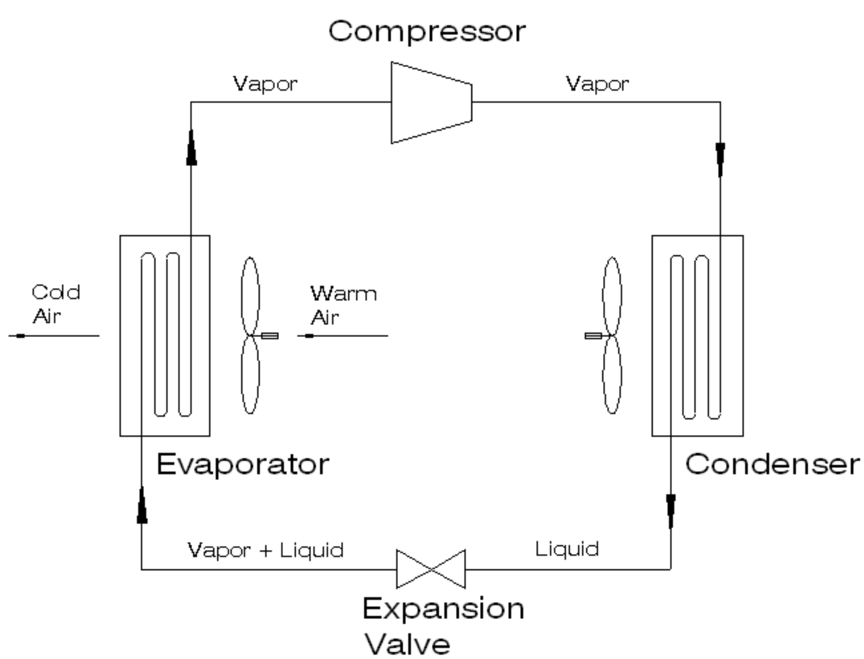

Fig.2-1 Typical Single Stage Vapor Compression Refrigeration

Refrigerators, air conditioners and chillers are all examples of heat pumps, except that they operate only in the cooling mode. In common practice, a heat pump can provide heating and cooling alternately by the control of the reversing valve. The reversing valve can change the direction of the heat pump refrigerant flow and therefore switch the heat pump operation from cooling to heating or vice versa. **Fig. 2-2** and **Fig. 2-3** illustrate the cooling mode and the heating mode for a typical air-to-air heat pump unit. Note the reversing valve position change between the two diagrams.

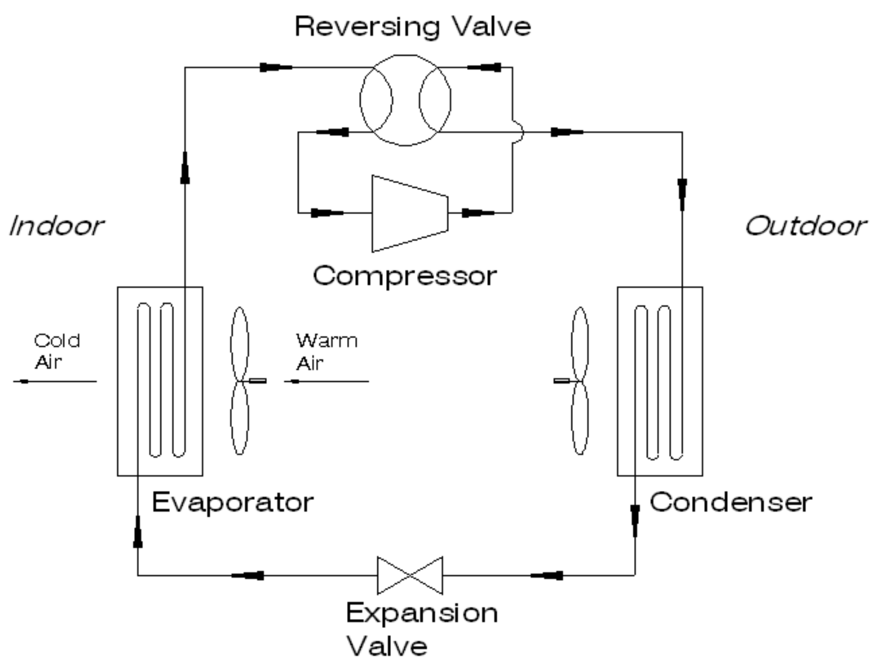

Fig.2-2 Heat Pump in the Cooling Mode

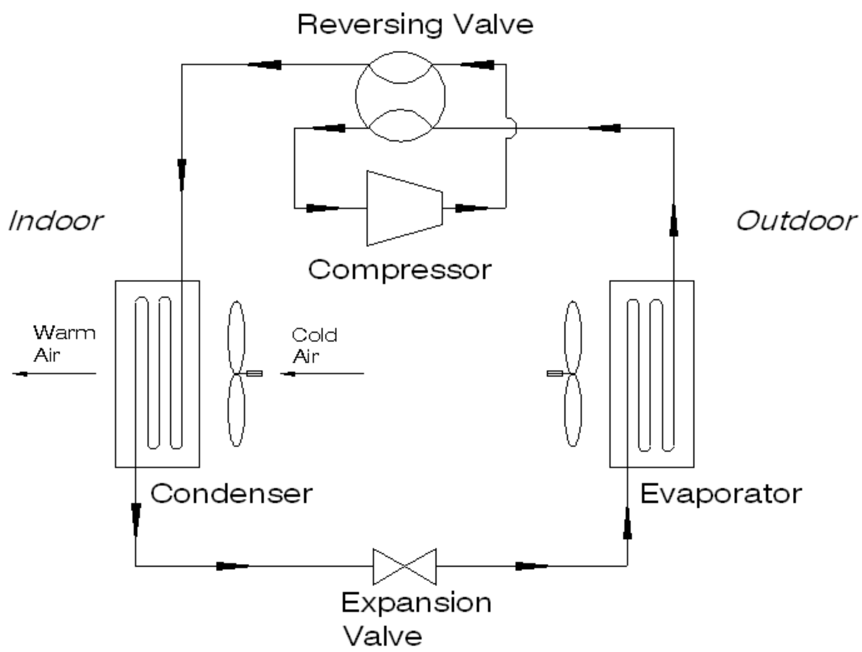

Fig.2-3 Heat Pump in the Heating Mode

Based on the theories above, heat pumps are able to move heat energy from one environment to another in dual directions. This allows a heat pump to either bring heat into an occupied space or take it out. The refrigerant is used to transfer heat within the heat pump. It absorbs heat as it vaporizes in the evaporator and releases heat as it condenses in the condenser. In the cooling mode, the indoor coil serves as the evaporator that absorbs heat from the conditioned space and the outdoor coil is the condenser that ejects heat to the outside. In the heating mode, the indoor coil turns into the condenser and the outdoor coil serves as the evaporator. The heat pump absorbs heat from the outside evaporator and rejects heat to the conditioned space through the inside condenser. The key component that makes such a change is the 4-way reversing valve which allows heat to be pumped in either direction. The compressor provides power and drives the refrigerant flow in the heat pump cycle.

2.1.4 Heat Sources and Heat Sinks

In thermodynamics, a heat source is defined as an object that generates thermal energy to a specified unit, and a heat sink is an object that absorbs thermal energy. In general, the only parameter that determines a heat source or a heat sink is the temperature difference. For example, if a window-mounted air-to-air heat pump unit is in the cooling mode, the outside air is the heat sink for the outside condenser, since the condensing temperature is higher than the outside air temperature. In the heating mode, the ambient air is the heat source for the outside evaporator, since the evaporating temperature is lower than the ambient air temperature. In heat pump applications, heat source and heat sink can be alternated with each other. Air, water, ground, etc. can be treated as either heat source or heat sink.

2.1.5 Types of Heat Pumps

Four heat pump types are presented in *2008 ASHRAE Handbook—HVAC Systems and Equipment*. They are air-to-air heat pumps, water-to-air heat pumps, water-to-water heat pumps and ground-coupled heat pumps. This part of the thesis only discusses two types that are vastly used, the air-to-air heat pump system and the water source heat pump system.

2.1.5.1 Air-to-Air Heat Pump System

This type of heat pumps is the most widely used in residential and light commercial buildings. Sometimes they are called packaged thermal heat pumps. Outside air is used as the heat source in the winter and as the heat sink in the summer. The heat pumps are usually discretely distributed and installed in a split manner, similar to typical window-mounted air conditioners. The capital cost and the installation cost is relatively low for the air-to-air heat pumps. The installation is fairly easy and flexible. However, this type of heat pump tends to be subject to a low COP compared with other types, especially in cold weather or extremely hot weather. High operating cost may occur in the peak load seasons. Defrosting in the heating-dominated season is another concern for this type of heat pumps. The defrosting energy cost can be over 10% of the total heat pump energy consumption in the humid and cold weather. In a highly-populated area, the air-to-air heat pumps can aggravate the heat island effect in the summer. Noise problem and relatively short life span are also common drawbacks for this type of heat pumps.

2.1.5.2 Water Source Heat Pump System

This type of heat pumps can be seen in many commercial, institutional and large residential buildings. The definition of water source heat pumps is broad and quite blurry sometimes. In general, a water source heat pump system either uses water as the medium to transfer heat with the natural outside heat source or sink, as in a geothermal heat pump system; or is equipped with supplemental heating or cooling devices to add heat to or remove heat from the water loop, as in a conventional water loop heat pump system. Sometimes, these two types are combined for better performance and lower cost. For a better understanding, three subcategories of water source heat pumps are given below.

(1) Conventional Water Loop Heat Pump System

The water loop heat pump (WLHP) system is also known as the California system, since it was developed and put into application first in California in the 1960s. This type of heat pumps uses existing internal load to balance the heating or cooling load of the conditioned zone without employing other natural heat source or heat sink such as ground, well water or surface water, etc. All the heat pumps in both the building core and perimeter areas are hydronically connected with a common closed water loop that circulates usually 60 °F to 90 °F water throughout the building. More often than not, auxiliary heating or cooling devices such as boilers or cooling towers are installed and they are turned on when the internal heat is insufficient to meet the load requirement in the peak heating or cooling seasons.

Most time during a year, the zone load is unbalanced and different from each other. The interior zone of the building is always cooling-dominated while the exterior zone of the building is heating-dominated in the cold weather. The zones on the sunny side of the building may require cooling while the zones on the shady side of the building may require heating. Therefore heat pumps in different zones may operate in the opposite modes. The units in the heating mode extract heat from the water loop while the units in the cooling mode reject heat to the water loop. Heat that could otherwise be wasted in a traditional central chiller-boiler system can be recovered and redistributed to where it is needed through the circulation of the water loop without simultaneously running both the chiller and the boiler. If more heat pump units are heating rather than cooling, the loop water temperature will drop. The boiler will start if the loop temperature is too low. Conversely, if more heat pump units are cooling than heating, the loop water temperature will rise. The cooling tower will start if the loop temperature is too high. In this case, the heat pump water loop can help balancing the heating and cooling load of each zone especially in partial load seasons, and at the same time ensure a good operating COP for all the heat pump units by giving a relatively stable heat source or sink loop water temperature. Moreover, compared with the central chilled/hot water-fan coil system, the installation cost of the water loop heat pump system is lower. This type of system is also suitable for the sites with limitations in geological conditions or outdoor spaces where large geothermal heat exchangers are not well qualified. **Fig. 2-4** shows a simplified diagram of a conventional water loop heat pump system.

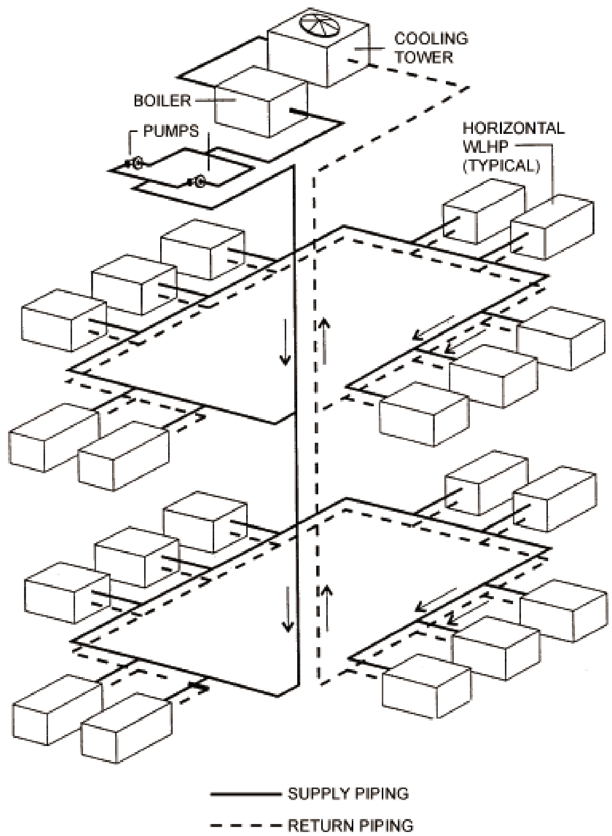

Fig.2-4 Conventional Water Loop Heat Pump System from 2008 ASHRAE Handbook

(2) Geothermal Heat Pump System

The geothermal heat pump is a broad definition that may include ground-coupled heat pump, ground water heat pump, well water heat pump, surface water heat pump, waste water heat pump, etc. This type of heat pumps uses ground or water as the heat source or sink, and uses air to add heat to or remove heat from the conditioned zone, or use water as a medium for heating and cooling. The geothermal heat pump can be classified into the following categories based on the type of heat source or sink: (1) Groundwater heat pump, which uses groundwater in the well as a heat source or sink; (2) Surface water heat pump, which uses surface water from a lake, river or pond as a heat source or sink; (3) Wastewater-source heat pump, which uses wastewater as a heat source or sink. Usually the waste water should be treated sewage or grey water. If raw waste water is directly used, special consideration shall be given on the design of the heat exchangers; and (4) Ground-coupled heat pump (GCHP), which uses geothermal heat exchanger to transfer heat to or from the ground. The geothermal heat exchanger can be installed horizontally, vertically, in a coil or in other configurations. Some ground-coupled heat pumps may use direct-expansion (DX) coils to transfer heat between the refrigerant and the ground.

Depending on whether the ground water is directly used or not, the geothermal heat pump system can also be divided into ground water source (open loop) system and ground-coupled (closed loop) system. An open loop system uses a surface or underground water source such as a lake, river or well as the heat source or sink. In a closed loop system, no water is taken from the ground or disposed of.

The temperature of earth or ground water 3 ft or below the ground surface is relatively stable because of the large heat capacity and thermal inertia of the ground. For example, the ground temperature throughout the United States generally ranges from 40 °F to 70 °F. Therefore, the ground temperature can be higher than the ambient air temperature in the winter and lower than the ambient air temperature in the summer. The loop temperature can be maintained at a level that gives a higher efficiency for the heat pump units. Also, by installing the ground heat exchangers, the need for supplemental heating or cooling load can be reduced or even eliminated. Considerable amount of operating cost can be saved compared with other types of systems. An electrically driven GCHP operating in the heating mode typical energy costs are between 50-70% less than for electric resistance heating, depending on climatic conditions, and at least 25% less than for an air-source heat pump (IEA 2002). Given these advantages, geothermal systems are very favorable and attractive to building owners. In the Leadership in Energy and Environmental Design (LEED) Standard 2.2, the geothermal system could earn up to 2 points under Energy and Atmosphere, Credit 1, depending on the energy rates and location.

However, this type of system often has a higher first cost than the conventional water loop heat pump system. For example in the US the total cost for an installed vertical ground heat exchanger including materials, drilling, backfilling, etc. is typically between 45 and 70 USD per meter (IEA 2002). This makes short period investment unattractive and the payback period may be longer than other systems. In addition, more emphasis shall be given to the design of the ground heat exchangers. Undersizing or improper design of the ground heat exchangers may lead to a ground temperature rise in the long run and compromise the system performance. Oversizing of the heat exchanger may result in high system initial cost and pumping energy cost. Local or municipal requirements for surface or ground water should also be followed. Finally, extra space is required to install the ground loops or heat exchangers.

(3) Hybrid Geothermal Heat Pump System

A cost-effective alternative for the conventional water loop heat pump system and the geothermal heat pump system is the combination of the above two types, called hybrid geothermal heat pump system. For most buildings, the peak heating load is not equal to the peak cooling load. Therefore, in such a system, the ground heat exchanger size can be reduced and an auxiliary heat rejecter or heat supplier is used to handle the excess cooling load or heating load. *ASHRAE 2007* and Xu (2007) both provided a design procedure for the ground-coupled heat exchanger in a hybrid system, and suggested that the ground heat exchanger can be designed to meet the small peak load while the supplemental cooling towers or boilers are only responsible for the difference between the large and small peak loads. In this way, the excessive capital cost for the geothermal heat pump system and the high operating cost for the conventional water loop heat pump system can be both decreased. This type of system is also very suitable for the sites where not enough space is allowed for the installation of a large ground heat exchanger.

2.2 Cooling Tower / Fluid Cooler Fundamentals

2.2.1 Cooling Tower Theories

In an HVAC system, a cooling tower is a device that rejects waste heat from the condenser water to the ambient air through a series of water/air heat and mass transfer processes. The condenser water can either come from the condenser of a chiller or come from a heat pump water loop. The cooling tower cools the water by a combination of heat and mass transfer in which the condenser water is in direct or indirect contact with the drafted outside air. Mass transfer and latent heat transfer are the dominating mechanisms that take place in a wet cooling tower process. The air stream is drafted by cooling tower fans and vaporizes the water. Latent heat can be released from water to the surrounding air and thereby the entering water temperature is decreased. 70% to 80% of the total cooling tower heat transfer is caused by the evaporative effect of latent heat of vaporization. Sensible heat transfer is also responsible for part of the cooling tower operation. There is a tendency for the water to be cooled by the surrounding air if certain temperature difference exists between them. Since a large amount of latent heat can be transferred from the water to the air, cooling towers are more efficient for large HVAC systems than the air-cooled heat exchangers in which only sensible heat is removed. A cooling tower can cool the condenser water to 5 °F to 7 °F above the ambient air wet bulb temperature.

Two important parameters in the cooling tower theories should be introduced first. One is called range. It is defined as the temperature difference between the cooling tower inlet and outlet water. In other words, the value of the range is also equal to the temperature rise of the chiller condenser or the heat pump water loop. The range is a function of the cooling load for the cooling tower and the water flow rate. There is no connection between the range and the cooling tower size or capacity. The larger the cooling load or the smaller the tower water flow rate, the greater the range. In HVAC industry, vast condenser water systems are designed with a 10 °F temperature difference. The design criteria for water chillers is 85 °F condenser water supply temperature and 95 °F condenser water return temperature. Therefore, most cooling towers are usually designed with a 10 °F range.

The concept of web-bulb temperature assumes that water can be cooled to be infinitely close to the ambient air wet bulb temperature. However, in an actual wet cooling tower, this can be hardly realized. The reason is that temperature difference is an indispensible element to complete a heat transfer process and there will be no heat transfer if the temperature difference is zero. The smaller the difference between the ambient wet bulb temperature and tower water temperature, the less sufficient the water-air heat transfer, the more the tower fan power consumed. In this case, another term called approach, short for the approach to the wet bulb temperature, is created. It is the difference between the cooling tower leaving water temperature and the wet bulb temperature of the ambient air or the air entering the cooling tower. In the cooling tower industry, it is uncommon to ensure any approach less than 5 °F. The approach is a function of cooling tower capacity. Therefore, a large cooling tower can possibly produce a closer approach to the wet bulb temperature than a small cooling tower. Once a cooling tower is selected with a given size or capacity and is working under a certain cooling load, the approach is only related with the ambient air wet bulb temperature and the required condenser water supply temperature. A change in wet bulb temperature or a change in range may result in a change in approach. The approach is the key factor that reflects the performance of the cooling tower.

Fig. 2-5 illustrates the temperature relationship in a typical counter-flow cooling tower. Point A and point B are entering and leaving tower water temperatures respectively. Point C and point D are entering and leaving tower air wet bulb temperatures respectively. The range is the vertical distance between A and B, while the approach is the vertical distance between B and C.

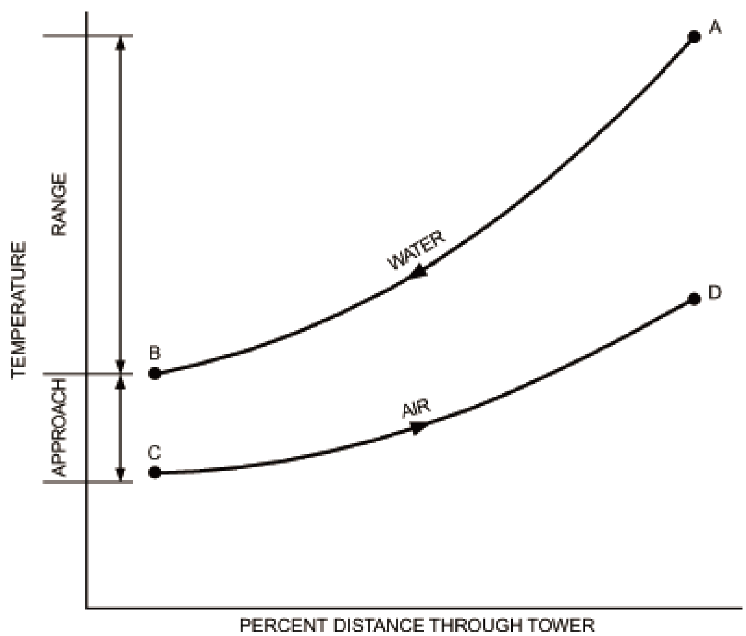

Fig.2-5 Temperature Relationship between Water and Air in Counter-flow Cooling Tower

from ASHRAE 2008 Handbook Chapter 39

2.2.2 Cooling Tower Types

Although almost all cooling towers are evaporative, mechanical draft type, their configurations can be varied based on the air/water flow direction, the tower fan location and whether the condenser water is in direct contact with the outside air. Three classifications are listed in the following.

2.2.2.1 Crossflow vs. Counterflow

In a crossflow cooling tower, water and air flow streams are oriented 90 degrees to each other, i.e., the water flows vertically downward while the air flows horizontally. This type of cooling tower requires small condenser water pumping energy and fan energy. However, large footprint is a drawback for the installation.

In a counterflow cooling tower, water and air flow in opposite directions. The counterflow tower is taller and with smaller footprint compared with a crossflow tower with the same capacity. Closer approach can be achieved since the air and water flow directions facilitate their mixing and improve the heat transfer. On the other side, this type of tower generally has high pump and fan power requirement due to the location of the spray nozzle and the large static pressure loss.

2.2.2.2 Forced Draft vs. Induced Draft

In a forced draft type, the tower fan is located at the air intakes of the tower, and the fill or heat exchanger inside the tower is under positive pressure; whereas in an induced draft type, the tower fan is located at the air outlet of the tower and the fill or heat exchanger is under negative pressure.

There is generally no advantage or disadvantage whether the cooling tower is with a forced draft fan or with an induced draft fan. However, the forced draft fan tower should be avoided in applications where the tower is used in the winter with icing potentials. The forced draft fan is located in the entering cold air stream rather than in the warm leaving air stream, making them highly susceptible to icing and potential mechanical failure.

Fig. 2-6 shows a typical forced draft crossflow cooling tower, while **Fig. 2-7** gives a picture of a typical induced draft counterflow cooling tower.

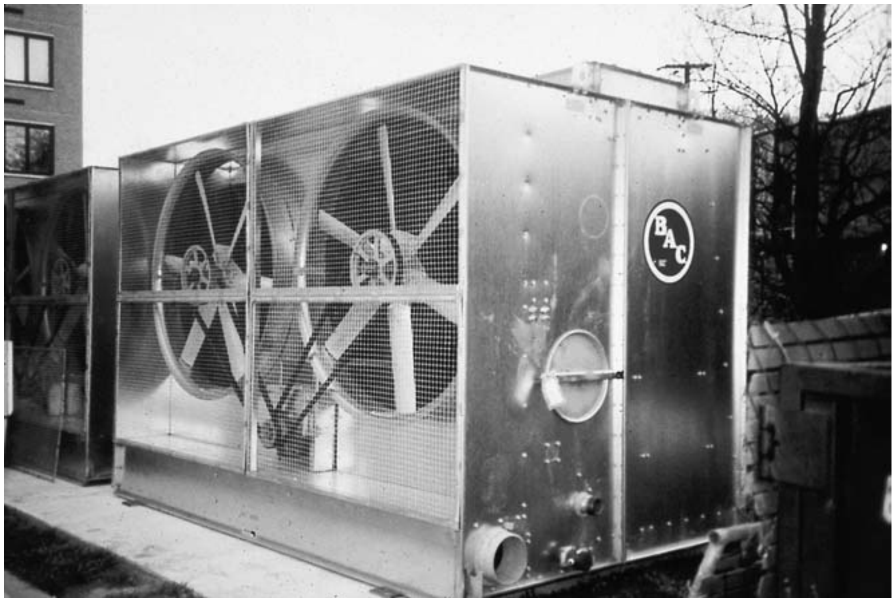

Fig.2-6 Typical Forced Draft Crossflow Cooling Tower (Courtesy of the Baltimore Aircoil Company Baltimore, MD)

Fig.2-7 Typical Induced Draft Counterflow Cooling Tower (Courtesy of the Evapco Inc., Taneytown, MD)

2.2.2.3 Open Circuit vs. Closed Circuit

For an open circuit cooling tower, the condenser water is exposed to the outside air. Usually there is no spray pump in the cooling tower. The condenser water pump should be sized with enough head to overcome the lift from the tower basin to the water spray nozzles on the top of the tower. Generally, an open circuit tower has a better heat transfer performance compared with a closed circuit tower. However, since condenser water is in direct contact with the atmosphere, one important concern is the condenser water treatment to prevent contamination.

For a closed circuit cooling tower, the condenser water is not in direct contact with the outside air. It requires a closed-circuit heat exchanger that is exposed to the air/water cascades similar with the fill of an open circuit cooling tower. Sometimes, this type of cooling tower is also called fluid cooler. The heat transfer of a closed circuit cooling tower may be worse than an open cooling tower. Yet this type of tower offers an alternative for winter tower operation using glycol and also simplifies the water treatment process for the condenser water. A diagram of a typical fluid cooler is shown in **Fig. 2-8**.

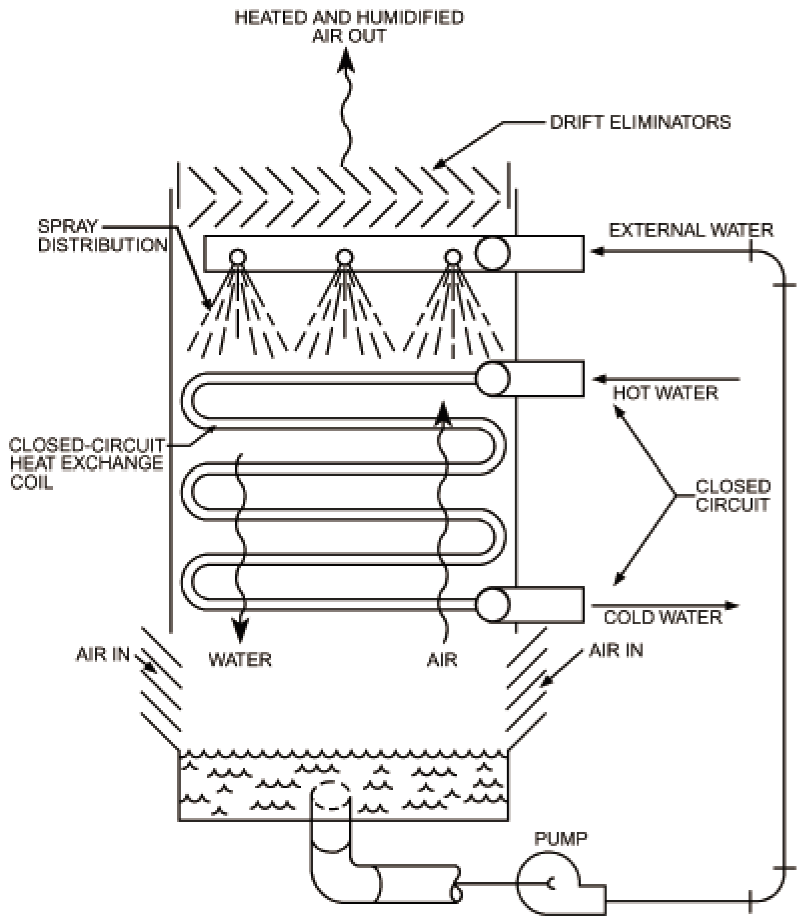

Fig.2-8 Typical Closed Circuit Cooling Tower / Fluid Cooler from ASHRAE 2008 Chapter 39

2.3 Typical Heat Pump Water Loop Control Strategies

In this section, common control strategies for variable speed pumps and temperature control for the heat pump water loop will be reviewed and discussed.

2.3.1 Common Variable Speed Pump Control Strategies

In the past few decades, the cost of variable frequency drives (VFD) has dropped significantly. More and more VFDs have been employed in the HVAC industry in controlling pumps and fans because of their potential energy savings and reliability. The installation of VFDs is becoming a trend especially for large systems. *ASHRAE Standard 90.1-2007 Energy Standard for Buildings except Low-rise Residential Buildings* requires that any heat pump system with more than a 10 horsepower circulation pump shall be with variable water flow. Since most of the researches that have been conducted were related with the variable pump control strategies in chilled/hot water systems, how to control the pump in a WLHP system is still a problem that needs to be addressed, although the control methods can be similar between these two types of systems. Four typical variable speed pump control methods that are commonly used in chilled/hot water systems will be discussed in the following section.

2.3.1.1 Control the Loop Pump Speed to Maintain a Constant Differential Pressure Setpoint

One common pump control method is to adjust the pump speed based on a constant differential pressure (DP) setpoint. The differential pressure sensors can be located on the main pipelines, for example, on both the suction side and discharge side of a pump; or can be connected to a certain terminal water coil for a selected zone. The pump speed is modulated to maintain a fixed differential pressure setpoint. The setpoint is usually determined based on the water loop design pressure when the pump is set at the maximum or a relatively high speed.

There is no question with this method if the system is always working at the design condition with all the control valves completely or widely open. However, buildings operate most time under the part load condition in which not much water flow is required in the same amount as the design flow. Therefore, the hydronic system resistance characteristics are different from the design. With this method, the control valves are forced to close as the load or flow is reduced to follow the high pressure setpoint. As a result, the circulation pump is forced to run at high speeds and large amount of pumping energy is wasted on the throttling valves with high flow resistance.

Another issue is that with a high system resistance, the pump operation points may fall into the unstable zone of the pump curve as the (Q_1, H_0) point shown in **Fig. 2-9**. The pump may oscillate between the two head-flow combinations. Usually in the unstable zone, the pump may be subjected to low efficiency as can be found in a typical pump flow-head-efficiency characteristic diagram.

Other inherent problems of this method may include valve control difficulties and water leakage at high pressure, noise accompanied with high pump speed and high system resistance.

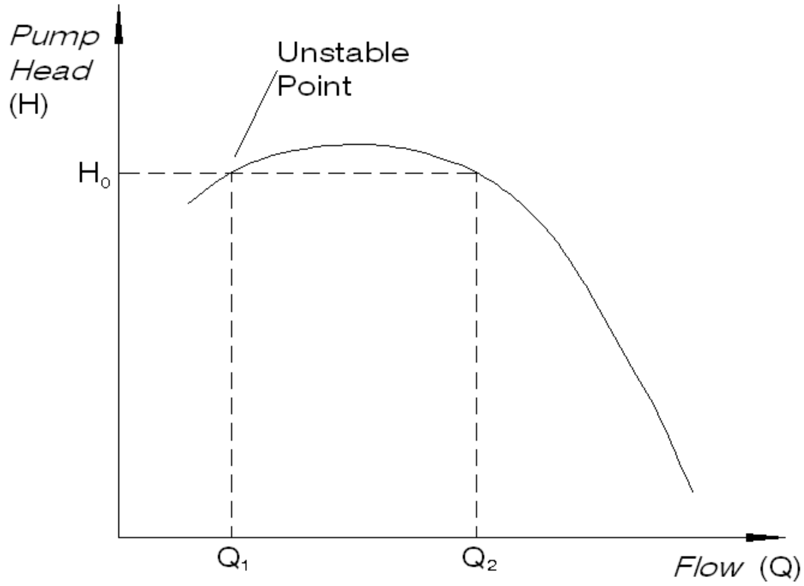

Fig.2-9 Unstable Pump Operation Zone on a Typical Pump Curve

2.3.1.2 Control the Loop Pump Speed to Maintain a Reset Differential Pressure Setpoint

The differential pressure setpoint can be reset based on the flow rate, the coil valve position or in some cases, the outside air temperature.

The DP reset based on the outside air temperature assumes that the building load is only related with the outside air temperature. Therefore, this method is only limited to buildings with a proportionally large exterior zone in which the heat flow across the envelope has a great impact on the heating or cooling load. Buildings with large heat gains from occupants, lighting and electronic equipments are not recommended to use this method. **Table 2-1** and **Table 2-2** are examples of resetting the DP based on the outside air temperature for the chilled water system and the hot water system respectively.

Outside air temperature (°F)	50	60	70	80	90
DP setpoint (psig)	6	8	10	12	15

Table 2- 1 Chilled Water DP Setpoint Reset based on Outside Air Temperature

Outside air temperature (°F)	50	60	70	80	90
DP setpoint (psig)	10	8	6	5	4

Table 2-2 Hot Water DP Setpoint Reset based on Outside Air Temperature

Another common DP control method is to modulate the pump speed to maintain the maximum valve opening position at 95% or greater (Liu et al, 2002). *ASHRAE 2007 Handbook*— *HVAC Applications* Chapter 41 recommended that the best pump control strategy for a chilled water set point is to reset the differential pressure setpoint to maintain all discharge air temperatures with at least one control valve in a full open condition. This method can result in relatively constant flow resistance and great pumping energy savings at partial loads. However, in systems with pneumatic controllers or and systems where the coil valve position data cannot be collected, this method is not applicable (Liu et al, 2007). Besides, control valve characteristics should also be taken into account while using this method.

Liu et al (2002) developed a step-by-step reset DP calculation procedure for the chilled water and hot water pumps and implemented the method in a library located on the Texas A&M

University (TAMU) campus at College Station, TX. The procedure uses trial-and-error method to balance the water flow in each coil by modulating the control valves. When the coil DP of all units match their design DP, the balanced flow rate is obtained. Finally, Eq. 2-1 and Eq. 2-2 are used to determine the DP setpoint.

$$S = \frac{P_{s,blanced} - P_{r,balanced}}{(\bar{Q}_{balanced})^2} \qquad \text{(Eq. 2-1)}$$

Where,

S = Flow rate coefficient;

$P_{s,balanced}$ = Water loop supply pressure at the balanced condition;

$P_{r,balanced}$ = Water loop return pressure at the balanced condition;

$\bar{Q}_{balanced}$ = Balanced water flow rate.

$$DP_{setpo\,\mathrm{int}} = S \cdot \bar{Q}^2 + P_{ad} \qquad \text{(Eq. 2-2)}$$

Where,

$DP_{setpo\,\mathrm{int}}$ = Calculated differential pressure setpoint;

$\bar{Q}$ = Actual flow rate;

P_{ad} = The pressure added to the setpoint for control valves to be adjusted for the requirement of flow rate. The value of P_{ad} is usually 1 psi to 2 psi based on field experience.

Liu et al (2005) presented a similar method to address the DP setpoint reset problem in Chapter 5 of the *Continuous Commissioning* [SM] *Guidebook for Federal Energy Managers*. Before resetting the DP, all the three-way valves should be converted into two-way valves by closing the manual valves on the bypass line. The coil that calls for highest DP for water distribution should also be identified. After that, slow down the pump speed until the coil control valve is 85% open. At this time, the water flow rate ($\bar{Q}_0$) and the DP (DP_1) at the loop DP sensor location are measured. This DP is the optimal DP setpoint under the measured flow condition. The DP setpoint can also be determined using Eq. 2-3. If the DP sensor is located at the most remote coil, then modulate the pump speed to maintain this setpoint.

$$DP_{setpoint,1} = DP_1 - DP_2 + C \qquad \text{(Eq. 2-3)}$$

Where,

$DP_{setpoint,1}$ = Calculated differential pressure setpoint when the DP sensor is located at the most remote coil, psi;

DP_1 = Differential pressure measured at the loop sensor position, psi;

DP_2 = Differential pressure measured across the coil, including the valve, psi;

C = Differential pressure required by the coil. It usually varies from 1 psi to 5 psi depending on the size of the coil. Exceptions may happen if the coil is blocked.

If the DP sensor is not located at the most remote coil, the DP setpoint should also be a function of the actual flow rate. Under this circumstance, the DP setpoint is reset based on Eq. 2-4. Generally, it is necessary to set a high limit and a low limit for the setpoint. The low limit should not be lower than 5 psi. The high limit should be calculated by replacing $\bar{Q}$ in Eq. 2-4 with the design flow rate.

$$DP_{setpoint,2} = DP_{setpoint,1} \cdot \left(\frac{\bar{Q}}{\bar{Q}_0} \right)^2 + 2 \qquad \text{(Eq. 2-4)}$$

Where,

$DP_{setpoint,2}$ = Calculated differential pressure setpoint when the DP sensor is not located at the most remote coil, psi;

$\bar{Q}$ = Actual water flow rate, gpm;

$\bar{Q}_0$ = Measured water flow rate when the valve with the highest coil DP is 85% open, gpm;

If the water flow rate is not measured, the differential pressure can be reset based on the pump speed as shown in Eq. 2-5.

$$DP_{setpoint,2} = DP_{setpoint,1} \cdot \left(\frac{v}{v_0} \right)^2 + 2 \qquad \text{(Eq. 2-5)}$$

Where,

v = Actual pump speed, %;

v_0 = Measured pump speed when the valve with the highest coil DP is 85% open, %.

An accurate measurement of the water flow is necessary in order to generate a proper DP setpoint. Liu et al (2007) developed a pump speed control strategy based on a reset loop differential pressure setpoint. The system resistance is obtained by the pump head and the water flow rate, which is calculated by the pump water flow station (PWS). The PWS is able to automatically calculate the water flow rate based on the current pump head, pump speed and pump performance curve. This control method has been tested in many commissioning projects of HVAC water systems and proved to be successful in reducing the pump speed and improving the hydronic system performance.

2.3.1.3 Control the Loop Pump Speed by a Constant Temperature Difference Setpoint

The temperature difference between the supply and return water is an important indicator of the actual heating or cooling load in a building. One rule of thumb equation in calculating the flow rate by the temperature difference and the building load is shown in Eq. 2-6.

$$GPM = \frac{Btuh}{500 \times \Delta T} \qquad \text{(Eq. 2-6)}$$

Where,

GPM = The flow rate to meet the heating or cooling demand, in gallons per minute;

$Btuh$ = The heating or cooling load of a particular zone, in British thermal units per hour;

ΔT = The temperature difference between supply and return water, °F. The designed ΔT for a heating system is usually 20 °F; while for many radiant floor heating and chilled water cooling systems, the designed ΔT is typically 10 °F;

500 = The specific heat of water, in Btu-min/gallon-hr-°F. 8.33 b/gal$\times$60 min/hr$\times$1 Btu/lb-°F = 500 Btu-min/gallon-hr-°F.

In practice, one variable speed pump control method for the chilled water system is to modulate the pumps speed to maintain a certain temperature difference of supply and return water. The temperature difference can be a constant or a variable based on load condition. If the pumps speed is based on a constant temperature difference, a low limit for the pump speed should be provided. This can prevent malfunctions of those flow-sensitive devices in the system such as chillers, heat pumps due to loss of water flow under low building loads.

In Chapter 3 of this thesis, a detailed analysis and discussion will be given on the constant DP setpoint method and the DP reset method, followed by a case study of a WLHP system in a school building, in order to make a comparison of these two pump control methods.

2.3.2 Common Water Loop Temperature Control Strategies

How to control the loop water temperature is another question in optimizing the WLHP system performance. The conventional control method is to maintain the loop temperature between a 60 °F lower limit and a 90 °F upper limit. The boiler is started if the loop temperature is below the lower limit and the cooling tower or fluid cooler is started if the loop temperature is above the upper limit. No measures are taken if the loop temperature is floating between 60 °F and 90 °F. This control strategy is easy to accomplish. However, it is not efficient for heat pump units, because it leads to high condensing temperature during the cooling season and low evaporating temperature during the heating season. The net result is that the COP of the heat pump units is not optimal as it should be.

Peitch (1990) did an investigation on the characteristics of closed water loop heat pump systems to find out opportunities to optimize the overall system performance related to loop water temperatures, loop water flow rates and additional thermal storage. He suggested that for optimal system efficiency, the cooling tower should be operated to cool the loop water down to a temperature approaching the boiler cut-in temperature, since the reduction of the cooling-mode compressor input at the lower entering water temperature will usually more than offset the additional tower energy consumption. The best overall system performance occurs when the loop is controlled to the lowest temperature levels possible during the cooling mode. Similarly, resetting the loop temperature to a higher value can increase the heat pump efficiency during the cold seasons.

Phetteplace and Sullivan (1998) conducted a study on the water loop temperature control strategies of a hybrid GHP system using data collected over a 22-month period. They found that the cooling tower fan and pump were only responsible for 4% of the system energy over the 22-month monitoring period while the heat pumps accounted for 77% of the total energy consumption. Thus, they recommended that the tower setpoint control be revised to either initiate tower operation at a lower fluid setpoint temperature or to operate the tower whenever possible to minimize the ground heat rejection load. Even though this may result in extra cooling tower energy consumption, the system energy consumption could be reduced since the heat pumps would operate at higher efficiency due to lower entering water temperatures. In addition, it was estimated that by using a lower loop water temperature, the circulation pump power consumption could be reduced by as much as 45%.

Palahanska-Mavrov et al (2005) performed a parametric analysis to find out the optimal supply water temperature for the water source heat pump system. The research showed that the water loop heat pump performance is more sensitive to the loop supply water temperature in the cooling mode than in the heating mode. By gathering data from a heat pump manufacturer, the authors found that the cooling energy efficiency ratio (EER) increases from 11 to 20 when the loop supply temperature decreases from 85 °F to 50 °F, and the heating EER increases from 17 to 19.5 when the supply temperature increases from 50 °F to 85 °F. As can be seen in **Fig. 2-10**, the curve of the EER is steeper in the cooling mode than in the heating mode.

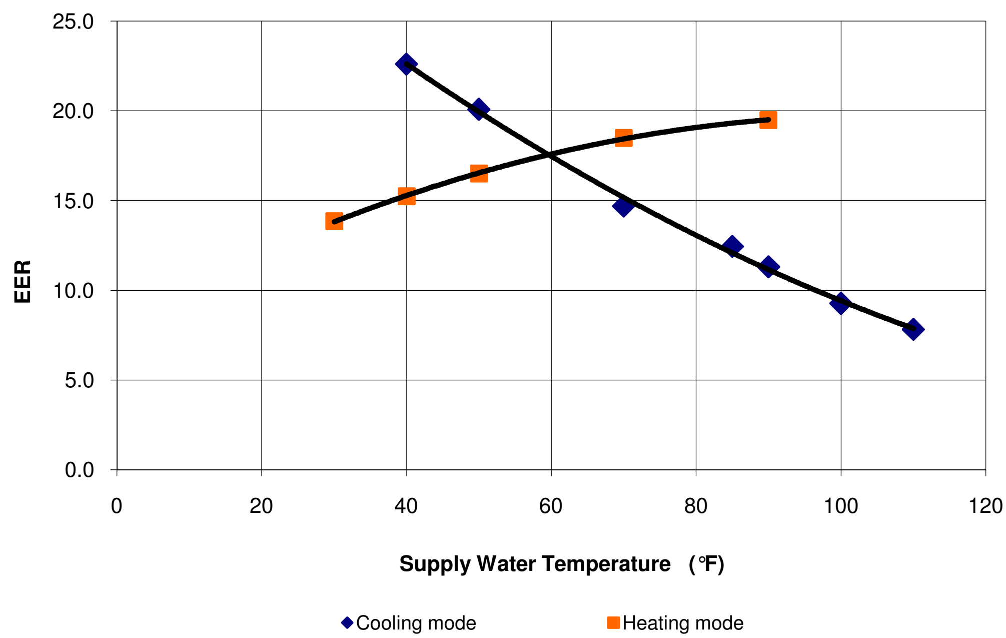

Fig.2-10 Relationship between Loop Supply Water Temperature and Energy Efficiency Ratio (EER)

The paper provided a simplified equation (Eq. 2-7) to demonstrate the relationship between the loop water temperature difference and the heating/cooling load. As can be derived from Eq. 2-7, when the cooling load is greater than the heating load, the return water temperature must be higher than the supply water temperature; when the cooling load is slightly smaller than the heating load, the return water temperature may still be higher than the supply water temperature;

when the cooling load is to a certain extent smaller than the heating load, the return water temperature is lower than the supply water temperature.

$$\Delta T = RWT - SWT = \frac{Q_c - Q_h}{\bar{m}c_p} + \frac{Q_h / COP_h + Q_c / COP_c}{\bar{m}c_p} \qquad \text{(Eq. 2-7)}$$

Where,

ΔT = Loop water temperature difference;

RWT = Loop return water temperature;

SWT = Loop supply water temperature;

Q_c = Building cooling load;

Q_h = Building heating load;

$\bar{m}$ = Loop water mass flow rate;

c_p = Specific heat of water at constant pressure;

COP_h = Heat pump coefficient of performance in the heating mode, each of the large zone heat pumps connected to the loop is assigned the same COP and the same EER regardless of possible differences of the heat pumps in the actual building;

COP_c = Heat pump coefficient of performance in the cooling mode.

A mathematical model was built in their research to analyze the impacts of the interior zone load ratio, the exterior zone partial load ratio and the supply water temperature on the overall heat pump system operation cost. It was concluded that the water source heat pump is most suitable for buildings that have interior zone design load ratio higher than 0.3. In addition, the optimal supply water temperature schedule strongly depends on both the interior zone design load ratio and exterior zone partial load ratio. In the summer operation, the supply water temperature must be 6 °F higher than the corresponding wet bulb temperature at full load conditions and must be at least 3 °F higher than the corresponding wet bulb temperature when the cooling load is 20 % of the design load. It was also recommended that the water temperature should be controlled as low as possible during the primary cooling season and as high as possible during the primary heating season to improve the heat pump efficiency. But the supply water temperature must be higher or equal to 50 °F to ensure system safety.

2.4 Building Schedule and Fan Mode Analysis

Building operating schedule plays an important role in the building energy consumption. Improper schedule or operating mode not only increases the electricity and gas consumption for the HVAC system, but also shortens the lifespan of the equipment.

One major function of the HVAC system is to maintain the room temperature and humidity at a proper level that can meet both the thermal comfort and the indoor air quality requirements for the occupants. However, in certain times of a day or a year, a building can be lightly-occupied or unoccupied. In intermittent occupied buildings, it is unnecessary to maintain the same zone comfort level during the unoccupied period as in the occupied period. An efficient building operating schedule is one that offers the desired thermal comfort to the occupants in the occupied day time while consumes the least amount of energy without compromising the building thermal safety during the unoccupied night time.

2007 ASHRAE Handbook—HVAC Application Chapter 41 gave two definitions for the measures to alter the unoccupied setpoint at night from the day occupied setpoint, namely, night setup and night setback. Night setup is used in summer to raise the zone temperature setpoint during the unoccupied time to reduce the cooling requirement. By contrast, night setback is used in winter to lower the zone temperature setpoint during the unoccupied time to reduce the heating requirement.

The unoccupied setpoint may vary from zone to zone. A temperature difference between the occupied setpoint and the unoccupied setpoint, i.e., the night setup or setback of 5 °F is generally acceptable. Since there is a transition period between the unoccupied time and occupied time, the unoccupied temperature setpoint should not be a value too far from the occupied setpoint in case it takes too long for the building to warm up or cool down.

Liu et al (2002) stated in the *Continuous Commissioning* ^{SM} *Guidebook for Federal Energy Managers* that two of the basic Continuous Commissioning ^{SM} measures are to turn off systems or slow down systems during unoccupied or lightly-occupied hours. The guidebook also suggested that a proper schedule should be developed for each zone based on the system, zone and occupancy characteristics, since turning off the system too early in the evening or turning on the system too late in the morning may cause comfort problems, while turning off the system too late in the evening or turning on the system too early in the morning may result in the loss of considerable energy savings.

Budaiwi (2003) performed a study on the operating schedule of a building in Saudi Arabia using the Visual DOE energy simulation software and concluded that a substantial amount of energy can be saved if proper ventilation and temperature control strategies are employed.

Erickson et al (2009) used a wireless camera sensor network to collect the occupancy data in a large multi-function building and created multivariate Gaussian and agent based models for predicting user mobility patterns in buildings. Using the models, they tried to control the HVAC system in an adaptive manner. Their results showed that a 14% of HVAC energy reduction can be achieved based on occupancy estimates and usage patterns.

In this thesis, a case study of a school building will be presented. The study will demonstrate how much electricity savings can be achieved by using an improved building occupancy schedule and fan operating mode. The study is conducted with both the energy simulation software tools and the field utility measurement.

Chapter 3 LOOP CIRCULATION PUMP CONTROL OPTIMIZATION

In this chapter, an optimal differential pressure (DP) reset pump control method will be discussed and compared with the constant DP pump control method. Since several pump control strategies have been discussed in detail in Chapter 2, this chapter will prove the advantage of the DP reset pump control method through its application in a case study. Pump energy consumption using both DP reset method and constant DP setpoint method will be analyzed.

3.1 Overview of the Case Study Building and the Existing Pump Control Strategy

Bellevue Mission Middle School is located in Bellevue, NE. The school building was originally built in early 1900s with several large-scale overhauls and building expansions during the 1980s and 1990s which made the total floor plan at 128,000 ft^2. The latest mechanical renovation was carried out in the summer of 2009. The renovation replaced almost all the chiller-boiler, Air Handling Unit (AHU), Fan Coil Unit (FCU) systems with a water loop heat pump system that serves 68 heat pump units with a total of 173.5 air condoning tons. It also added five outside air roof top units (RTU) with heat recovery wheels to solve the previous indoor air quality (IAQ) problem. The outside air RTU handles the load of the outside air only, while the heat pump in each zone handles the zone load only.

A diagram of the WLHP system is illustrated in **Fig. 3-1**. Two circulation pumps are operated in a lead-lag sequence in which one pump is on and the other is off, and they rotate every week. The heat pump water loop is connected with two boilers and one fluid cooler. A three-way valve connects the fluid cooler with the loop. If the loop requires heat rejection, the three way valve opens and leads the water through the fluid cooler; otherwise, the three way valve closes and the water bypasses the fluid cooler. If the loop temperature is too low, the boiler and the boiler pump will be started. VFDs are installed for the circulation pumps and for the fluid cooler fans. Heat pump control on the water side is simply determined by weather the compressor is on or off. If the heat pump compressor is on, the isolation valve is opened and the water runs through the heat pump unit; otherwise, the isolation valve is closed and the water bypasses the heat pump unit. The mechanical schedules of the loop pumps, boiler pump, VFDs, fluid cooler and boilers can be seen in **Appendix A** Table A-1 to Table A-4.

The existing loop control strategy forced the circulation pump to maintain the loop pressure at a constant 15 psi at the loop DP sensor location. Problems such as high pump speed, loop DP fluctuation, large noise and valve control difficulties occurred frequently before the new DP reset method was applied.

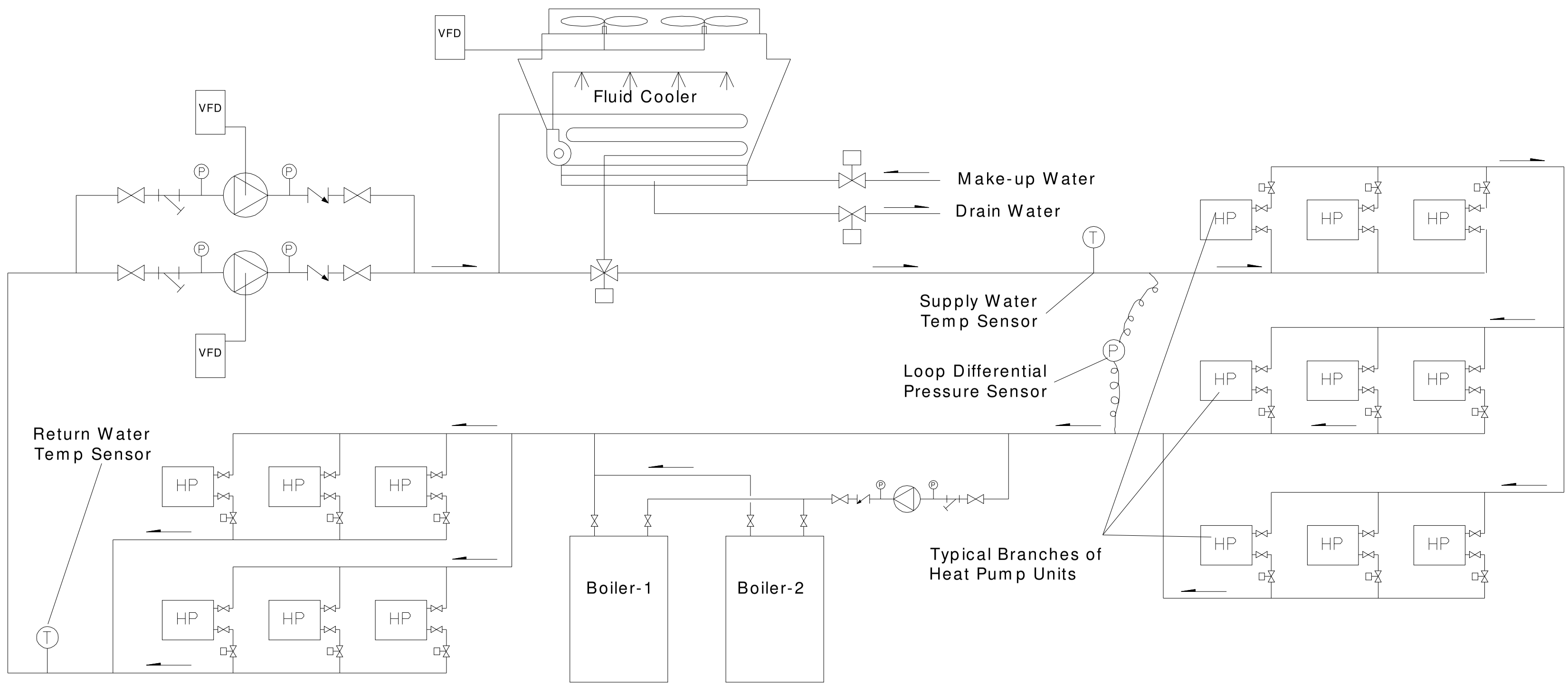

Fig. 3- 1 Water Loop Heat Pump System Diagram

3.2 Differential Pressure Reset Pump Control Theories

A number of DP reset pump control strategies have been mentioned in Chapter 2. The parameters and equations used to determine a reset DP setpoint are very similar to each other. In this case study, the DP setpoint is determined based on Eq. 3-1, which is very close to Eq. 2-10 in Chapter 2. The first term on the right side of the equation calculates the pressure drop on the main supply and the main return pipe. It is a function of the actual flow rate. The second term on the right represents the differential pressure on the most remote heat pump coil which can be treated as a constant value.

$$DP_{setpoint} = (DP_{design} - DP_{\min}) \cdot \left(\frac{\bar{Q}_{actual}}{\bar{Q}_{design}} \right)^2 + DP_{\min} \qquad \text{(Eq. 3-1)}$$

Where,

$DP_{setpoint}$ = Differential pressure setpoint, psi;

DP_{design} = Design differential pressure setpoint, psi;

$DP_{\min}$ = Minimum differential pressure for the heat pump, psi. This is generally the pressure drop across the most remote heat pump, 5 psi in the case study;

$\bar{Q}_{actual}$ = Actual flow rate, gpm;

$\bar{Q}_{design}$ = Design flow rate, gpm.

A significant parameter in order to specify the DP setpoint is the actual loop water flow rate. In the case study building, no flow meter was installed. One pair of DP sensors is installed on the 2/3 way down the water loop. Besides, the pump head can be measured from the pump head sensors. In this case, a method developed by Liu et al (2007) called pump water flow station (PWS) was employed to calculate the real-time loop water flow rate. The PWS is able to measure water flow rate using the existing pump head, pump speed and pump performance curve. The pump head can be regressed as a polynomial function of the water flow rate using the in-situ pump curve as shown in Eq. 3-2.

$$H_{design} = \sum_{i=1}^{n} k_i \cdot \bar{Q}_{design}^{i} \qquad \text{(Eq. 3-2)}$$

Where,

H_{design} = Pump head when the pump is at 100% speed, psi;

$\bar{Q}_{design}$ = Water flow rate when the pump is at 100% speed, gpm;

k_i = Coefficient of the ith term.

Based on the pump affinity law, the water flow is proportional to the pump speed, the pump head is proportional to the square of the pump speed, and finally the pump brake horsepower is proportional to the cube of the pump speed. Therefore,

$$\omega = \frac{N_{actual}}{N_{design}} \tag{Eq. 3-3}$$

$$H_{actual} = \omega^2 \cdot H_{design} \tag{Eq. 3-4}$$

$$\overline{Q}_{actual} = \omega \cdot \overline{Q}_{design} \tag{Eq. 3-5}$$

Where,

N_{actual} = Actual pump speed, rpm;

N_{design} = Design pump speed, rpm;

ω = Pump speed ratio, %;

H_{actual} = Actual pump head, psi;

$\overline{Q}_{actual}$ = Actual water flow rate, gpm.

Eq. 3-2 can be expressed as Eq. 3-6 after the substitution from Eq. 3-3, Eq. 3-4 and Eq. 3-5.

$$\frac{H_{actual}}{\omega^2} = \sum_{i=1}^{n} k_i \cdot \left(\frac{\overline{Q}_{actual}}{\omega^2} \right)^i \tag{Eq. 3-6}$$

Yuill (2003) demonstrated that the second and third order polynomials are with sufficient accuracy for this flow rate model. Therefore, the pump curve can be represented using a second order polynomial (i.e. i=2 for Eq. 3-6) as shown in Eq. 3-7.

$$H_{actual} = k_0 \cdot \omega^2 + k_1 \cdot \omega \cdot \overline{Q}_{actual} + k_2 \cdot \overline{Q}_{actual}^2 \tag{Eq. 3-7}$$

Eq. 3-7 can be fitted by manufacturer's pump curve or in-situ measured pump curve, where k_0, k_1 and k_2 can be determined. Using the second order regression of the pump curve, the pump water flow can be calculated based on the measured pump head and pump speed as in Eq. 3-8.

$$\bar{Q}_{actual} = \frac{-k_1 \cdot \omega - \sqrt{k_1^2 \cdot \omega^2 - 4 \cdot k_2 \cdot (k_0 \cdot \omega^2 - H_{actual})}}{2 \cdot k_2} \qquad \text{(Eq. 3-8)}$$

3.3 Differential Pressure Reset Pump Control Case Study

The pump curve from the manufacturer's catalog as shown in **Fig. 3-2** was used directly to obtain the second order regression polynomial. The point at the bold black right angle indicates the design point. Later in the experiment, the flow rate and the pump head were both verified using pressure gauge and ultrasonic flow meter based on the in-situ pump curve recommendation from Liu, Wang and Liu's paper (2007). The actual water flow and pump head was found to be agreed with the curve in general, with a 5.3% error on average for seven testing points at the design flow rate with different flow resistance. Therefore, the manufacturer's curve was reliable for the water flow calculation.

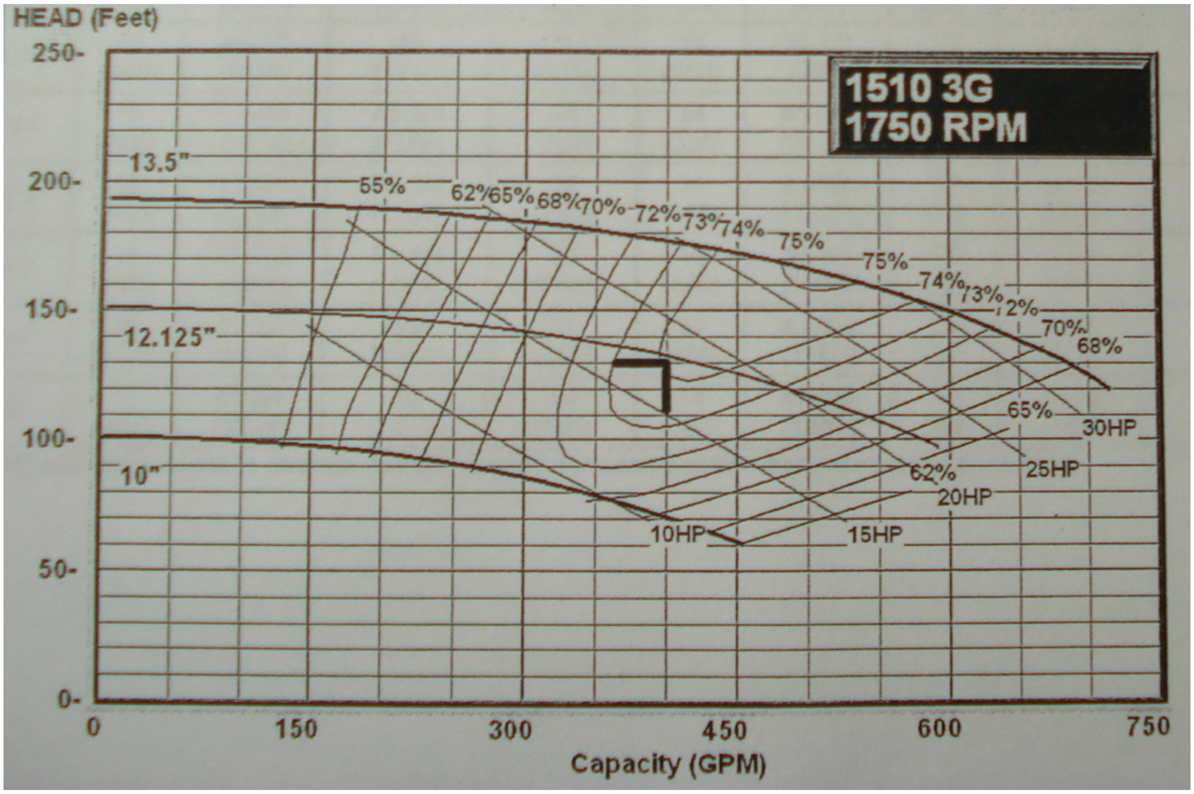

Fig. 3-2 Loop Water Circulation Pump Curve from Manufacturer's Catalog

The water flow rate was calculated based on real-time pump head and pump speed according to Eq. 3-8 and the DP setpoint was calculated using Eq. 3-1. During the process, several tests were carried out to find out value of the $DP_{\min}$ term in Eq. 3-1. Since it was difficult to measure the pressure drop of the most remote heat pump given the complexity of the loop piping, a rule-of-thumb value of 4 psi of $DP_{\min}$ was used at first. However, it was reported that a few of heat pumps by the end of the loop tripped off several times. After that, the $DP_{\min}$ was set at 5 psi and no heat pump tripping problems occurred. To ensure the heat pumps are working with sufficient water flow, a 6 psi to 16 psi range was set up for the DP setpoint and the controller PID loop was retuned to make the pump speed respond faster to the change of the loop DP. Inputs and outputs of this pump control method can be seen from an EES (Energy Equation Solver, developed by Purdue University) graphic model as shown in **Fig. 3-3**.

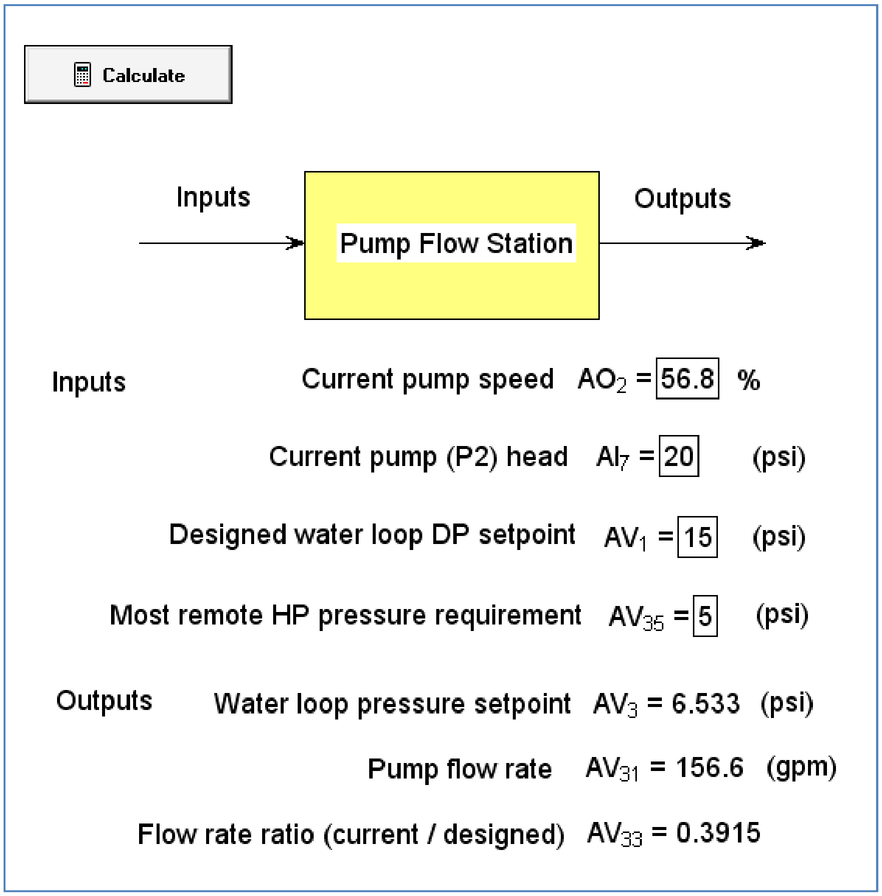

Fig. 3-3 Pump Flow Station Program Interface in EES software

Pump speed data were trended every two minutes from May 3, 2011 to May 5, 2011. The outside air temperatures ranged between 47.8 °F and 76.1 °F in the two-day period (47.8 °F to 71.5 °F in the first day and 53.9 °F to 76.1 °F in the second day). The old constant DP pump control method was used during the first day and the new DP reset control method was used during the second day. The detailed pump speed in these two days can be seen in **Fig. 3-4**. The old pump control method forced the pump to run with 55% to 75% speed most of the time with an average of 60.04%, while the new DP reset method reduced the pump speed to between 38% and 55% speed with an average of 43.72%, even though the outside temperature was slightly higher on the second day of the experiment. On average, there was a 16.3% of pump speed difference between these two methods. The DP reset method saved 60.9% pumping energy compared with the constant DP pump control method from the pump KW trending data of the two days. The pump KW curve of the two days is illustrated in **Fig. 3-5.**

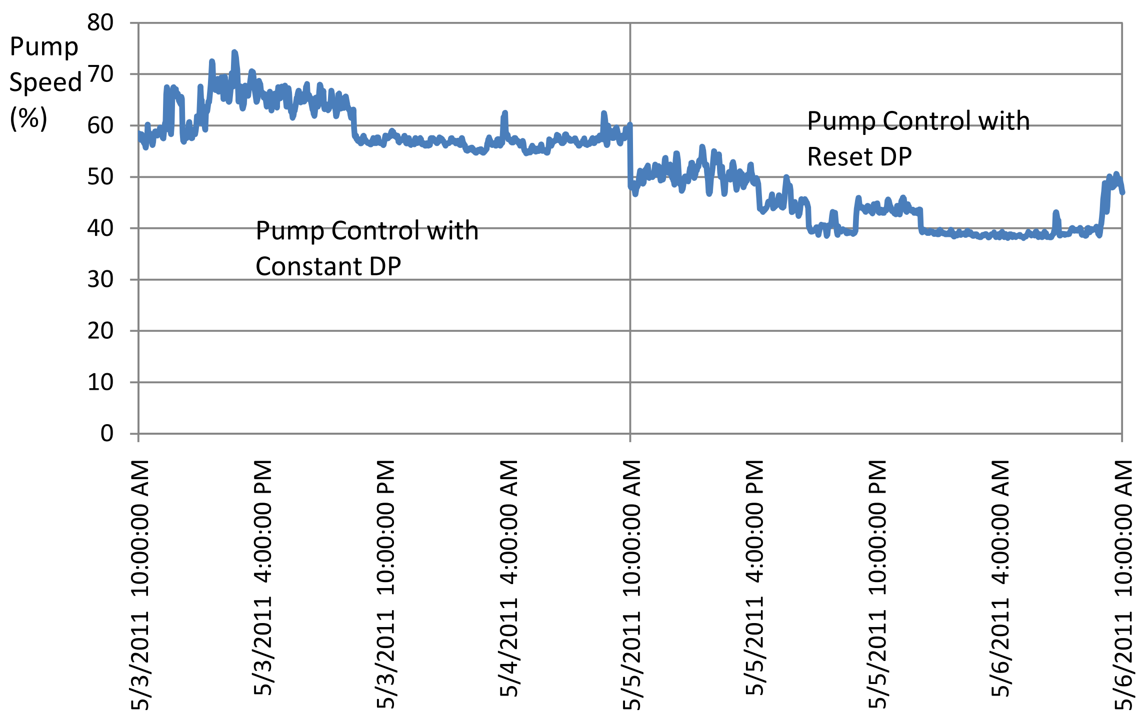

Fig. 3-4 Pump Speed from Constant DP Method vs. Pump Speed from DP Reset Method

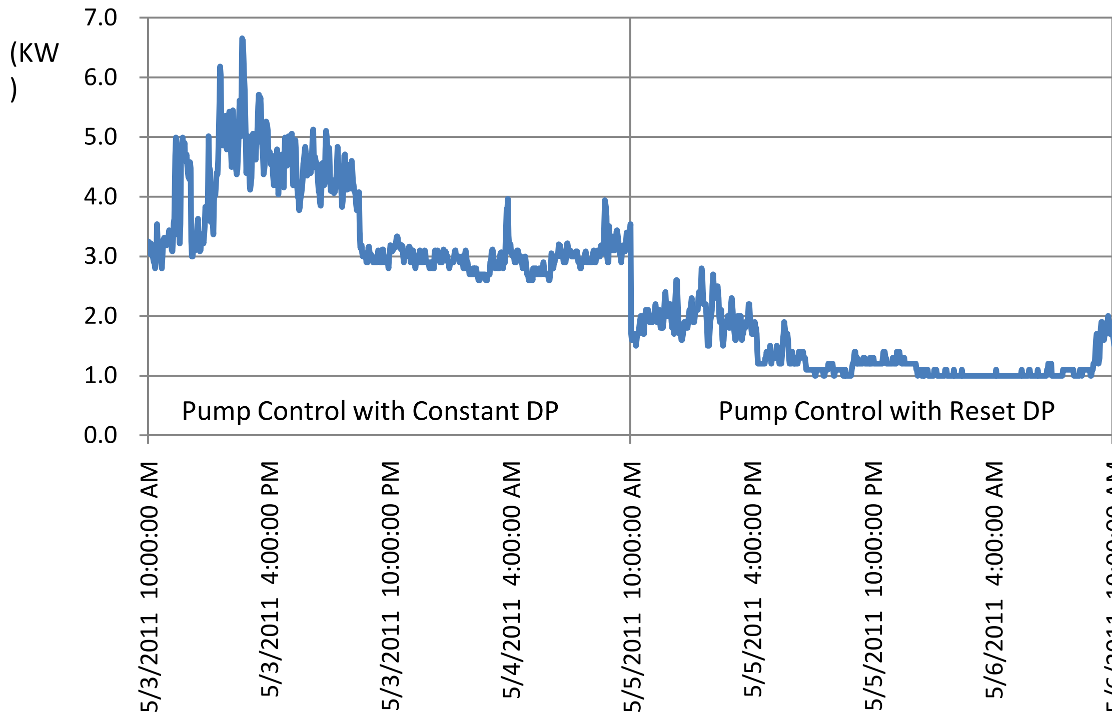

Fig. 3-5 Pump Power from Constant DP Method vs. Pump Speed from DP Reset Method

Another fact worth mentioning is that the loop water temperature profiles of these two days are very similar. The loop temperature was between 64 °F and 81 °F for both days, which means that the new control method did not sacrifice the heat pump COP or EER to gain the loop pump energy savings. The loop temperature control and fluid cooler operation strategy will be discussed in Chapter 4.

Fig. 3-6 shows the differential pressure from both days of the experiment. It can be seen that in the first day when the DP was maintained at 15 psi setpoint, the loop pressure was high and unstable even with several overshoots of 6 psi. In the second day, the DP was reset every 100 seconds and a low minimum setpoint of 6 psi was applied. In addition, the DP was more stable compared with the old method. Since the outside air temperature was not too high and economizers were activated for the RTUs when the experiment was conducted, the loop DP setpoint was maintained at the minimum 6 psi most of the time in the second day. It should be noted that the 6 psi DP minimum setpoint was determined based on several trial-and-errors. The heat pumps at the far end of the loop away from the circulation pump experienced more risks of malfunctions if the loop DP setpoint is below 6 psi.

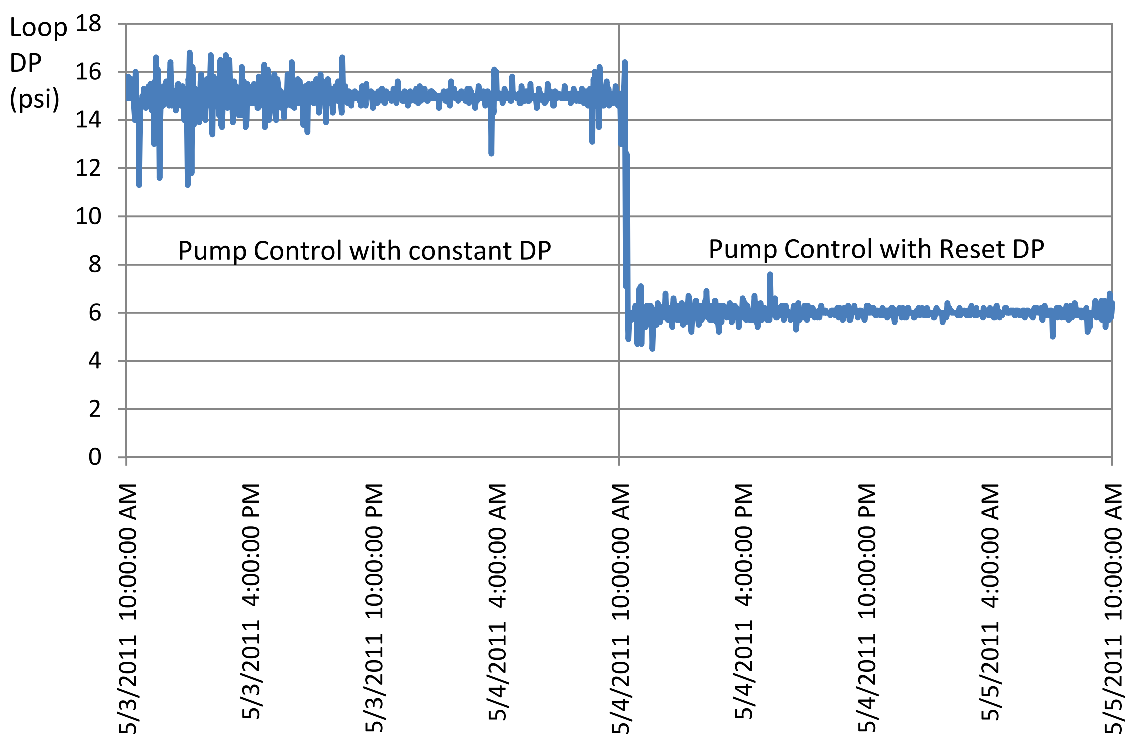

Fig. 3-6 Loop DP from Constant DP Method vs. Loop DP from DP Reset Method

It may not be clear to see the DP setpoint change in **Fig. 3-6** since the weather was mild and free cooling was allowed from the heat-recovery RTUs. Thus, it seemed the loop pressure was maintained at the lowest 6 psi from the trending data for the most time in the second day.

However, in hot or extremely cold weather, the DP setpoint change can be seen more clearly. **Fig. 3-7** shows the DP setpoint change along with outside air temperature on Sunday, May 8, 2011 when the school was unoccupied. **Fig. 3-8** shows the setpoint OAT relationship on Monday, May 9, 2011. It can be seen that occupancy level and outside air temperature are two important factors that influence the loop DP setpoint.

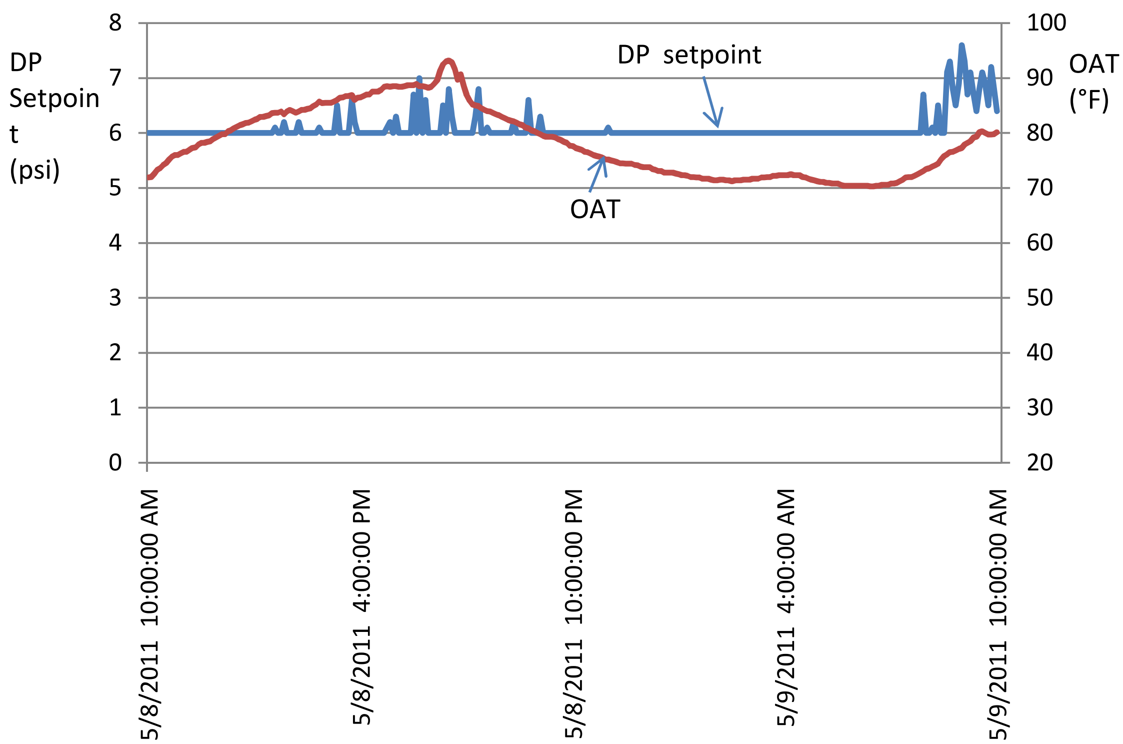

Fig. 3-7 Reset DP Setpoint vs. Outside Air Temperature on weekend

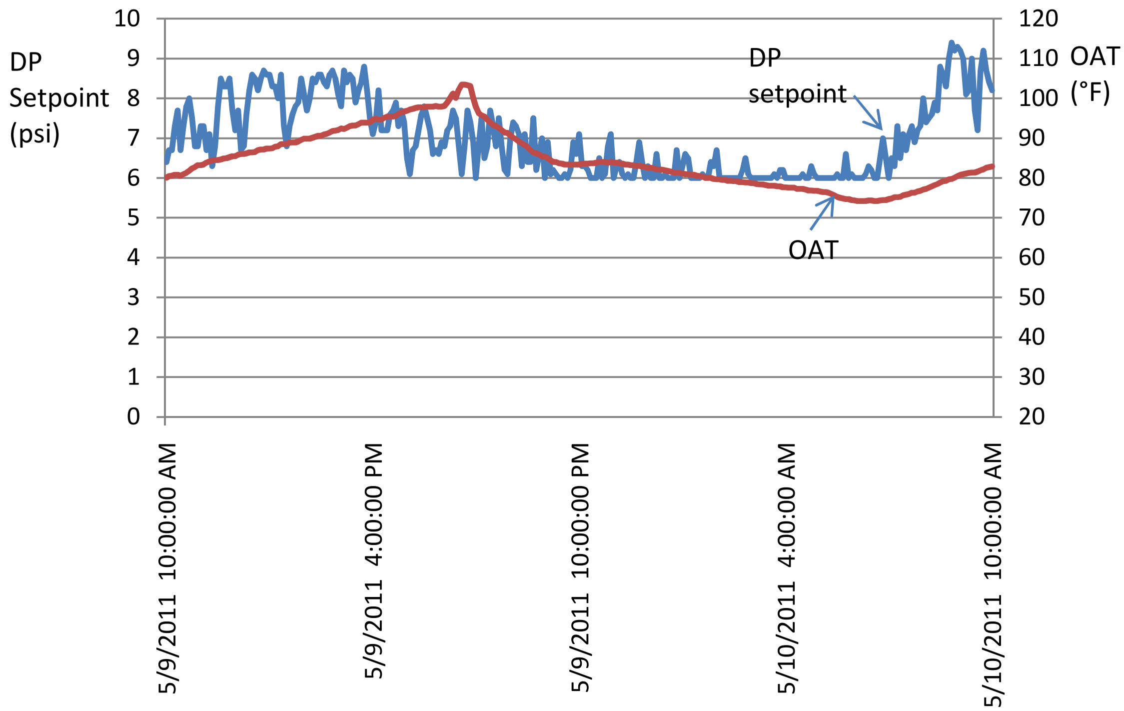

Fig. 3-8 Reset DP Setpoint vs. Outside Air Temperature on a typical school day

In order to find out if other methods generate better pump operation, another pump control method based on loop temperature difference was tested on the same water loop system. The pump speed was adjusted to maintain a constant Delta-T between supply water and return water. The temperature difference setpoint was set at 10 °F based on the design temperature difference for the heat pump units and the design temperature range of the fluid cooler. However, it was found that during most of time the loop Delta-T was less than 10 °F. Although in the winter of 2010, the loop pump was once operated to maintain a 10 °F Delta-T without causing any heat pump failures, in the summer some heat pump units tripped with a 10 °F Delta-T. The reason is that the heat pumps farthest from the loop circulation pump are not provided with enough flow using this method.

In conclusion, the DP reset pump control method was found to be more efficient than the constant DP pump control method for the WLHP system. For this specific middle school building, there can be 10% to 25% pump speed difference between these two control methods. As a result, 40% to 70% pump KW savings can be expected by using the new DP reset control strategy without comprising the COP or EER for the heat pump units.

Chapter 4 FLUID COOLER CONTROL AND LOOP WATER TEMPERATURE CONTROL OPTIMIZATION

The heat pump unit efficiency is closely related with the loop water temperature in a WLHP system. This chapter will discuss in detail how to operate the fluid cooler and how to control the loop water temperature to improve the heat pump efficiency and the overall WLHP system efficiency through a case study. In addition, a study will be conducted in this chapter to analyze how much energy can be saved by increasing the EER of heat pumps by lowering the loop water temperature in the summer and how much extra energy is consumed on the fluid cooler operation.

4.1 Overview of the Existing Loop Temperature Control Strategy in the Case Study

The same middle school building is used again for a comparison of the existing loop temperature control method and the improved loop temperature control method. The system (as shown in **Fig. 3-1**) is a very typical WLHP system in which the water loop is connected with two gas-fired boilers that add heat in the heating season and one closed-circuit fluid cooler that rejects heat in the cooling season. The data of the boiler and fluid cooler can be found in **Appendix A** Table A-3 and Table A-4. The fluid cooler is similar with the one shown in **Fig. 2-8**.

The fluid cooler is connected to the loop through a three-way valve. There are only two positions available for this valve. The valve can be controlled to either direct water through the cooler or let water bypass the cooler. The fluid cooler is induced-draft counter-flow type with two fans on top. One VFD is connected with both fans to modulate the fan speed. One make-up valve and one drain valve are connected with the fluid cooler reservoir. The drain valve opens and the make-up valve closes if the outside air temperature is below 35 °F, so that water can be removed from the reservoir to prevent tower freezing. The drain valve closes and the make-up valve opens if the outside air temperature is above 45 °F, so that water can be kept in the reservoir and recirculated by the spray pump when heat extraction is required. The two boilers are parallel with each other and can be both cycled on and off to maintain a certain loop temperature.

The existing control method is listed in **Table 4-1**. In the cooling mode, the loop supply water temperature (SWT) was maintained at 80 °F or higher based on a linear relationship between supply water temperature and cooling signal. The cooling signal directly controlled the fluid cooler fan and spray pump on/off and the fan speed. In the heating mode, the boilers cycled on if the loop return water temperature (RWT) drops below 63 °F and cycled off if the return water temperature rises above 73 °F. The boiler fire rate was controlled to maintain the loop supply water temperature at 85 °F.

This loop control method is easy for controller programming and causes no serious or obvious problems such as heat pump tripping or malfunctioning. However, this method is not energy efficient. Quite similar with the 60 °F/90 °F loop temperature control method, it forced most heat pumps to run with low COP or EER, especially in part load seasons. It is certainly not a good choice to operate the heat pumps at the 80 °F loop temperature when it is possible to drop the heat sink temperature to a much lower level in some mild temperature days.

4.2 Loop Temperature Optimization Theories

The COP or EER of the heat pump units is a function of the loop water temperature. In the cooling mode, the higher the loop water temperature, the lower the cooling EER for the heat pump units. On the contrary, in the heating mode, the higher the loop water temperature, the higher the heating COP for the heat pump units. Palahanska-Mavrov, Wang and Liu's paper (2005) presented a diagram of the relationship between EER and loop water temperature as shown in **Fig. 2-10**. Former researchers have also put forward the idea that the loop temperature can be as low as possible in the cooling season and as high as possible in the heating season to improve the efficiency of the heat pump units.

The fluid cooler outlet temperature setpoint can be reset based on the enthalpy of the outside air, since the physical process inside the cooling tower is ideally an isentropic process. *ASHRAE*

2008 Chapter 39 provides four diagrams (two of them shown below) of the relationship between outside air enthalpy and the supply water from the cooling tower. **Fig. 4-1** and **Fig. 4-2** demonstrate the curves of cooling tower outlet water temperature with different dry bulb temperatures and wet bulb temperatures under 100% tower design flow rate and 67% tower design flow rate.

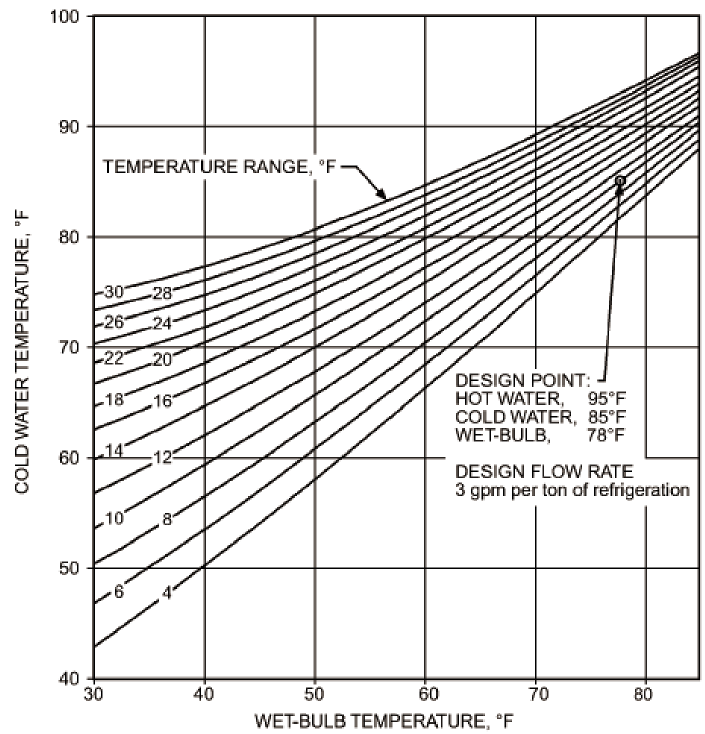

Fig. 4-1 Cooling Tower Performance—100% Design Flow from 2008 ASHRAE Handbook Chapter 39

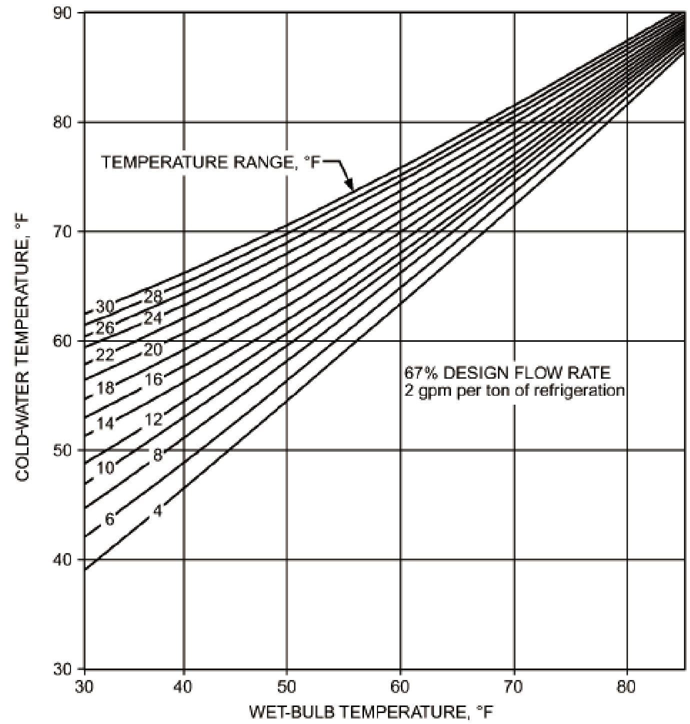

Fig. 4-2 Cooling Tower Performance—67% Design Flow from 2008 ASHRAE Handbook Chapter 39

However, the water supply temperature reset by outside air enthalpy is not applicable or at least difficult to be used in reality, simply because:

(1) Relative humidity or wet bulb temperature in addition to the dry bulb temperature must be known to calculate the value of enthalpy. Usually, the readings from the outside wet bulb temperature sensors are not accurate because they are very sensitive to too many interfering factors such as surrounding air velocity, direct sunshine, locations, etc. The large error of the wet bulb temperature may create large error of the enthalpy especially when the part of latent heat dominates in the total enthalpy.

(2) Even wet bulb temperature is known, it is difficult to follow the procedure given by *ASHRAE 2008* in a real project, since the curves vary by too many parameters such as leaving tower water temperature, wet bulb temperature, flow rate, the range of the cooling tower, etc. Even all these parameters can be measured and determined, and a universal equation is fitted to satisfy all the conditions, how to incorporate the equation to the on-site controller programming is still another problem.

Therefore, an easy and applicable method is strongly recommended in resetting the fluid cooler outlet water setpoint. The goal of this setpoint is to drop the loop supply water temperature to a reasonably low range without affecting system safety. From the Psychrometric Chart, it can be seen that on a hot and humid summer day when the outside air dry bulb temperature is between 65 °F and 100 °F with the relative humidity above 50%, the wet bulb temperature is 12 °F to 18 °F below the dry bulb temperature. For simplicity, use the medium 15 °F in the model. Thus,

$$DBT - WBT = 15 \qquad \text{(Eq. 4-1)}$$

Where,

DBT = Outside air dry bulb temperature, °F;

WBT = Outside air wet bulb temperature, °F.

Assume the fluid cooler approach to the wet bulb temperature is 5 °F. Then,

$$SWT - WBT = 5 \qquad \text{(Eq. 4-2)}$$

Where,

SWT = Loop supply water temperature, also the fluid cooler outlet water temperature, °F.

The temperature difference between supply water and outside air dry bulb temperature can be deducted from the above two equations, as in Eq. 4-3.

$$SWT = DBT - 10 \quad \text{(Eq. 4-3)}$$

According to Eq. 4-3, the fluid cooler outlet temperature setpoint should be 10 °F lower than the outside air temperature. **Fig. 4-3** shows a scatter point diagram of the loop supply water temperature and the corresponding outside air temperature in the cooling mode from May 4, 2011 to June 7, 2011 when the improved control method that resets supply water temperature setpoint was utilized. The supply water setpoint was limited within a 65 °F to 85 °F range. The temperature data were collected every five minutes. It can be seen that the OAT- 10 °F supply setpoint is reasonable especially when the outside air temperature is above 80 °F. The reason why the supply temperature does not quite follow the setpoint at the low outside air temperatures is that the control sequence makes the loop cooling starts at a temperature between 77 °F to 82 °F in order to create a dead band between heating shut-down and cooling start-up. Therefore, the fluid cooler may not be started until the loop temperature rises above 77 °F.

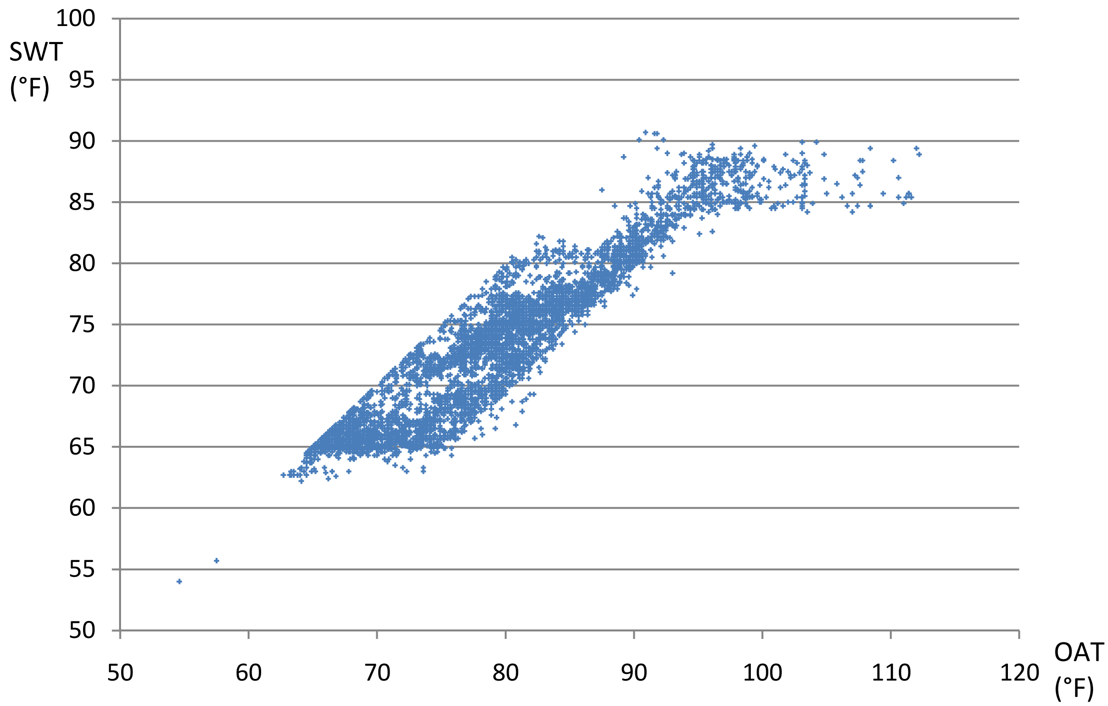

Fig. 4-3 Loop Supply Water Temperature (SWT) vs. Outside Air Temperature (OAT) in the Cooling Mode

One inevitable fact is that lowering the loop temperature in the summer will increase the fluid cooler energy consumption. There is a tradeoff between the heat pump cooling EER and the fluid cooler fan and spray pump power. However, since the energy consumption of the fluid cooler is usually less than 20% of the entire WLHP system energy consumption, it is still possible to achieve large amount of energy savings through the loop temperature optimization.

On the other hand, the temperature difference between the loop water and the heat pump coil on the water side is increased by this optimization. As a result, the loop water flow rate is decreased and less loop pumping power is required. For example, when the fluid cooler is operating, for a lower loop temperature, less water flow is needed to meet the load requirement, since the temperature difference between the heat pump condenser and the loop water becomes greater. In this case, the pumping power will decrease.

4.3 Loop Temperature Optimization Case Study

A series of loop temperature optimization procedures listed in **Table 4-1** were conducted on site by reprogramming of the controllers of the fluid cooler fans, the spray pump, the boilers, and the three-way valve, etc. A few of the key features of the control optimization are discussed in the following.

4.3.1 Cooling/Heating Mode Redefining

The new control strategy provides four modes including heating, non-heating, cooling and non-cooling. Under the cooling mode, the fluid cooler will be enabled and the three-way valve will be direct water to the fluid cooler. Under the non-cooling mode, the fluid cooler will be disabled and the three-way valve bypass water from the fluid cooler. Both of the boilers and the boiler pump will be enabled in the heating mode and disabled in the non-heating mode.

Note that in **Fig. 3-1** the return water temperature sensor is located between the boilers and the fluid cooler. This installation is different from the design mostly seen. In the most common WLHP system design, the supply water temperature sensor and the return water temperature sensor are installed on both sides of the boilers and the fluid cooler. The building load is a function of the difference between the supply and return water temperature. In the primary heating season, the return water temperature is lower than the supply water temperature; while in the primary cooling season, the return water temperature is higher than the supply water temperature. Therefore, the cooling/heating mode can be defined based on whether or not the return water temperature is higher than the supply water temperature.

	Existing Loop Temperature Control Method	Optimized Loop Temperature Control Method
Cooling/Heating mode definition	Cooling mode: SWT is above 80 °F. Heating mode: RWT is below 63 °F.	Cooling mode: RWT is above 82 °F or if RWT is above 77 °F and increasing very fast. Non-cooling mode: RWT is below 65 °F or if RWT is 1 °F below SWT. Heating mode: RWT is below 58 °F or if RWT is below 63 °F and decreasing very fast. Non-heating mode: RWT is above 73 °F
	The fluid cooler is enabled if system turns into cooling mode and the boiler is enabled if system turns into heating mode.	The fluid cooler is enabled if system turns into cooling mode and is disabled if system turns into non-cooling mode. The boiler is enabled if system turns into heating mode and is disabled if system turns into non-heating mode.

Table 4-1 Comparison between the Existing Loop Temperature Control Strategy and the Optimized Loop Temperature Control strategy

	Existing Loop Temperature Control Method	Optimized Loop Temperature Control Method
SWT setpoint in the cooling mode	80 °F. If SWT is above 80 °F, the setpoint is a temperature that corresponds to a pre-decided linear relationship of SWT and cooling signal which controls the fluid fan speed and spray pump on/off.	Reset based on OAT. Normally, this setpoint is (OAT-10) °F, and 65 °F ≤ SWT setpoint ≤ 85 °F. If the fluid cooler is drained in the cold weather, in which case the spray pump is disabled, the SWT setpoint is 77 °F if RWT is less than 90 °F, or 82 °F if RWT is over 90 °F.
Boiler operation in the heating mode	The boiler kicks on if RWT is below 63 °F, and kicks off if RWT is above 73 °F.	The boiler operating temperature range is enlarged. The boiler kicks on if RWT is below 58 °F or if the RWT is below 63 °F and decreasing very fast. The boiler is stopped if RWT is above 73 °F.
3-way valve operation	The 3-way valve changes position and let water run through fluid cooler if RWT is above 80 °F.	The 3-way valve changes position and let water run through fluid cooler if RWT is above 82 °F or if RWT is above 77 °F and increasing very fast.
Fluid cooler fan and spray pump sequencing	3-way valve, fluid cooler spray pump and fluid cooler fan control sequencing are based on a linear coordination between RWT and cooling signal. A certain range of RWT corresponds linearly to a range of signal from 0 to 100. Then, use the signal value to determine the sequencing of the equipments. Fan speed is also based on the same linear coordination table.	If SWT is above setpoint, start the fluid cooler fan. The minimum fan speed is 20%. For the first three minutes after the fan starts, keep the fan speed at 20%. After that, modulate fan speed to maintain SWT setpoint. The desired fan speed is calculated by a PID controller. If the fan is on for over five minutes and SWT is above SWT set point for over three minutes, then start the spray pump. If the spray pump is on for over five minutes and SWT is 2 °F below SWT setpoint for over three minutes, then stop the spray pump. If the spray pump is off for over five minutes and SWT is below SWT setpoint, then stop the fan.
Fluid cooler maximum fan speed	The maximum fluid cooler fan speed is always 100%.	The maximum fluid cooler fan speed is 80% if RWT is below 90 °F or 100% if RWT is over 90 °F.

Table 4-1 (Cont.) Comparison between the Existing Loop Temperature Control Strategy and the Optimized Loop Temperature Control strategy

Due to the special location of the return water temperature sensor in the case-study building, the heating/cooling mode for the water loop is defined mainly based on the change of the return water temperature, since the return temperature is also a good indicator of the building load change. If the return water temperature rises above a high limit or moves up very fast above a certain temperature, the cooling mode will be started. Likewise, if the return water temperature drops below a low limit or moves down very fast below a certain temperature, the heating mode will be initiated.

4.3.2 Reset Supply Water Temperature in the Cooling mode

The new control sequence reset the fluid cooler supply water temperature setpoint 10 °F below the outside air temperature. The building does have a wet bulb temperature sensor. But its readings were abnormal more often than not. The actual wet bulb temperature was measured outside the school building. It was found that sometimes the error could be as much as 15 °F. Therefore, only the outside air dry bulb temperature sensor was utilized in this study after it was verified.

A setpoint high limit of 85 °F was defined in order to ensure a relatively high cooling EER at the high outside air temperature. In another word, when the outside air temperature is over 95 °F, the fluid cooler will run as much as possible to achieve a low loop temperature. On the other side, in mild weather conditions when the outside air temperature is between 45 °F to 65 °F, it is possible that some heat pumps require heating. Thus, a setpoint low limit of 65 °F was defined to make sure that the water loop is not too cold at a low outside air temperature.

In the cold weather, the fluid cooler sump is drained to prevent freezing and the spray pump is disabled. However, it is still possible that the building requires cooling. In this case, a winter fluid cooler operation mode was set up. When cooling is desired, the fluid cooler fan will be modulated to maintain the loop supply water temperature at 77 °F if the return water temperature is below 90 °F; and the supply temperature setpoint will be 82 °F if the return water temperature is above 90 °F.

From June 15, 2011 to June 17, 2011, an experiment was performed to compare the electricity consumption with different supply water temperature setpoint. The OAT-10 °F setpoint was used in the first day and the setpoint of 80 °F was used in the second day. The two days were chosen because the daily outside air temperatures of both days were similar. The temperature ranged from 67.8 °F to 99 °F on June 15 and from 68.5 °F to 99.5 °F on June 16. The loop temperature was trended every five minutes. **Fig. 4-4** shows the loop supply temperature and return temperature during the two days. It can be seen that during the first day the loop temperature was between 65 °F and 75 °F in about 2/3 of the day; while during the second day, the loop temperature was between 80 °F and 90 °F most of the time.

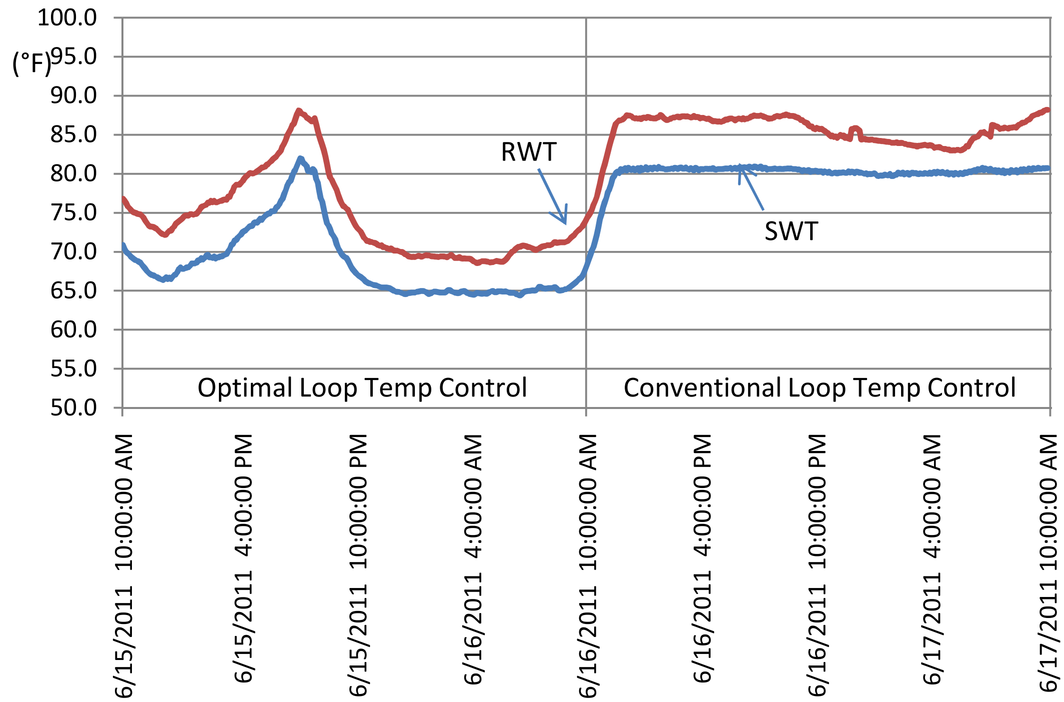

Fig. 4-4 Comparison of Loop Supply Water Temperature (SWT) and Loop Return Water Temperature (RWT) with the Optimal Loop Temperature Control and with the Conventional Loop Temperature Control

Fig. 4-5 shows the loop water flow rate in both days of experiment. The loop DP reset method covered in Chapter 3 was applied for the loop pump control for both days. The loop water flow rate GPM was calculated using the previously mentioned flow station algorithm, and was also trended every five minutes along with the temperature difference data in those two days. It can be seen that the conventional loop temperature control method required more water flow and cost more pumping energy than the optimal loop temperature reset method. The real-time loop pump KW data is illustrated in **Fig. 4-10**.

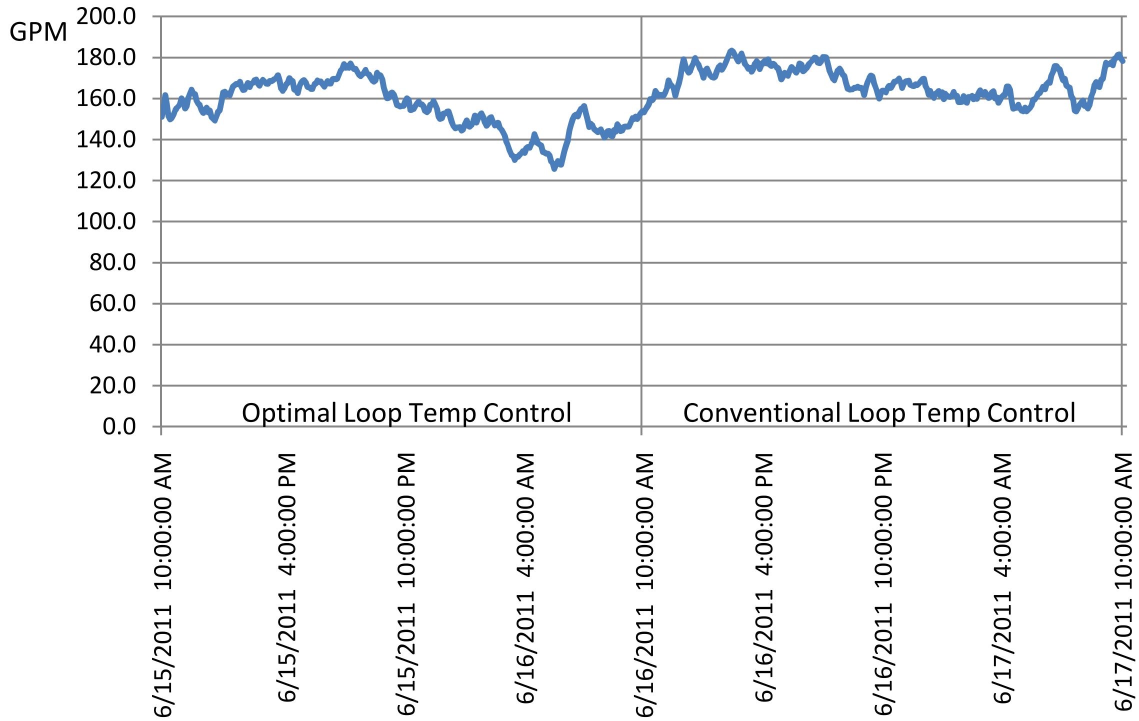

Fig. 4-5 Comparison of Loop Water Flow Rate with the Optimal Loop Temperature Control and with the Conventional Loop Temperature Control

Since there is no KW trending for each heat pump, the heat pump electricity cannot be measured one by one. In this case, an alternative method using water flow rate, loop temperature difference and EER was utilized to calculate the actual building load and electricity consumption. The EERs from the manufacturer's catalog were weighed by the number and the capacity of the heat pump units in the building. **Fig. 4-6** illustrates the weighted average EERs at different heat pump entering water temperature. Since the entering water temperature value for each heat pump is not available, assume the average of the supply temperature and return temperature as the universal entering water temperature. Thus, in any single time interval of five minutes, all the heat pump units share the same entering water temperature and EER until the next five minutes.

The EER against loop average temperature relationship in the two day period is shown in **Fig. 4-7**.

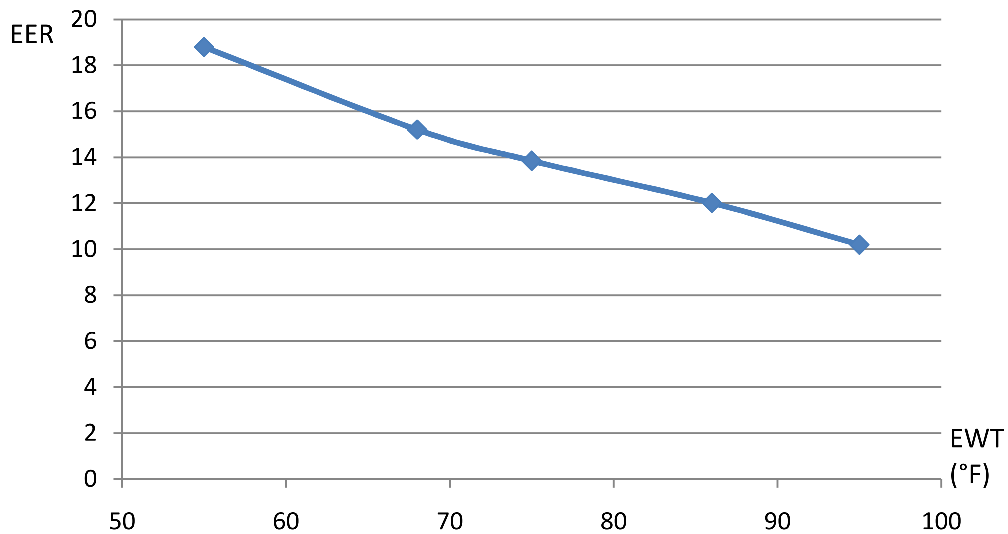

Fig. 4-6 Relationship between Heat Pump Cooling EER and Heat Pump Entering Water Temperature (EWT)

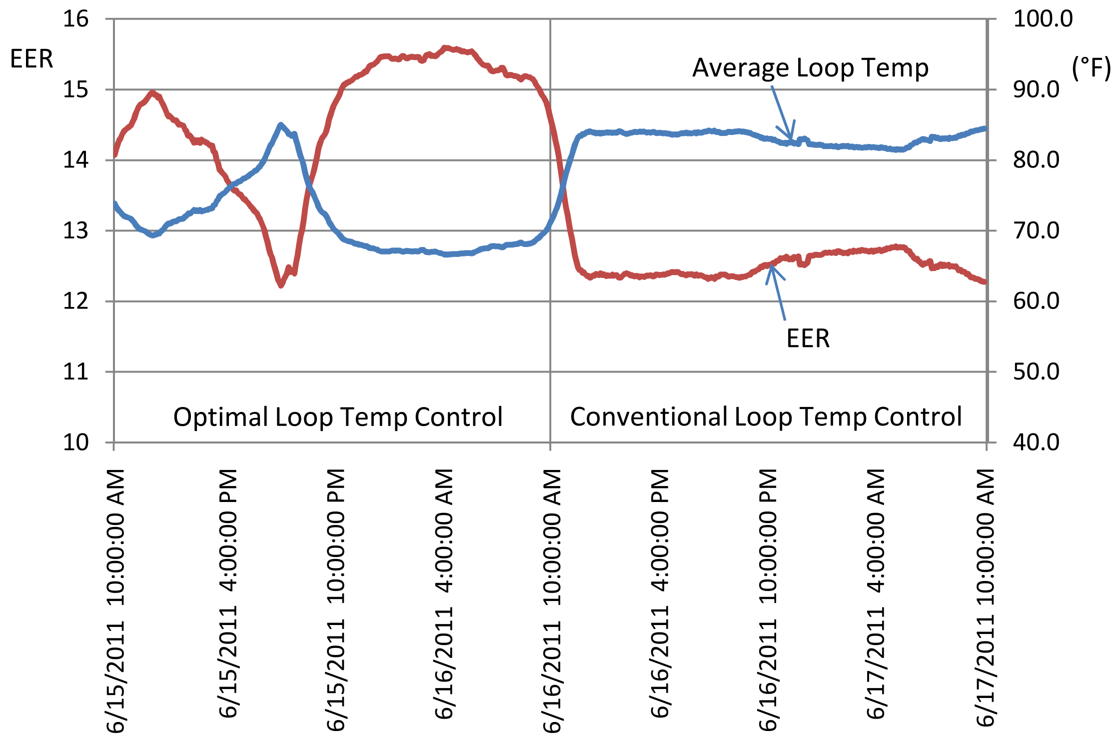

Fig. 4-7 Comparison of Average Loop Temperature and EER with the Optimal Loop Temperature Control and with the Conventional Loop Temperature Control

Throughout the two day period, the return water temperature was found to be higher than the supply water temperature based on the trending data. Therefore, the cooling load can be calculated based on Eq. 2-12 from Chapter 2 as in Eq. 4-4. Note that the ΔT term is always positive during the experiment.

$$Btuh = 500 \times \Delta T \times GPM \qquad \text{(Eq. 4-4)}$$

Where,

GPM = The flow rate to meet the heating or cooling demand, in gallons per minute;

$Btuh$ = The heating or cooling load of a particular zone, in British thermal units per hour;

ΔT = The temperature difference between supply and return water, °F.

ΔT = RWT – SWT;

500 = The specific heat of water, in Btu-min/gallon-hr-°F. 8.33 b/gal$\times$60 min/hr$\times$1 Btu/lb-°F = 500 Btu-min/gallon-hr-°F.

Fig. 4-8 illustrates the calculated cooling load in MBH (1000 Btu/h). It can be found that the load profiles for the two days are very similar, although there are still slight differences.

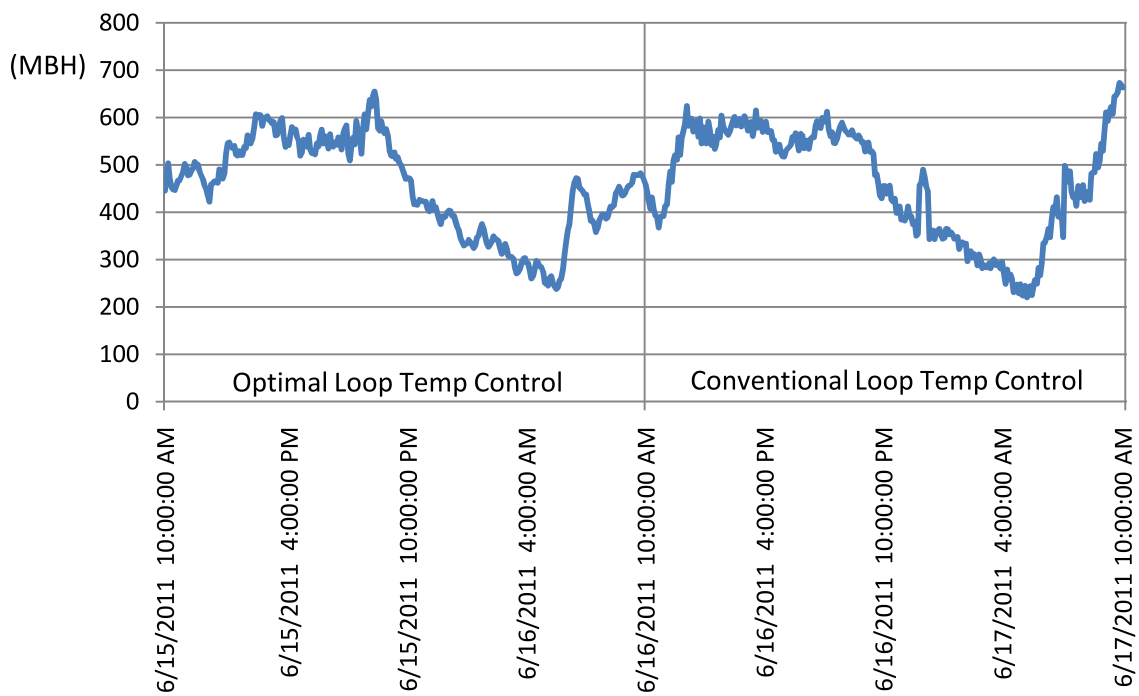

Fig. 4-8 Cooling Load Profile in the Loop Temperature Control Case Study

The heat pump compressor power can be calculated based on the heat pump theory, as in Eq. 4-5.

$$KW = \frac{Btuh}{EER} \tag{Eq. 4-5}$$

Where,

KW = Heat pump compressor power.

Fig. 4-9 gives the profile of electricity consumption of the heat pump units during the two days. The vertical axis represents the KW and the area enclosed by the KW values and the horizontal axis is the electricity consumption in KWh, i.e., the KWh value is the integration of KW by time.

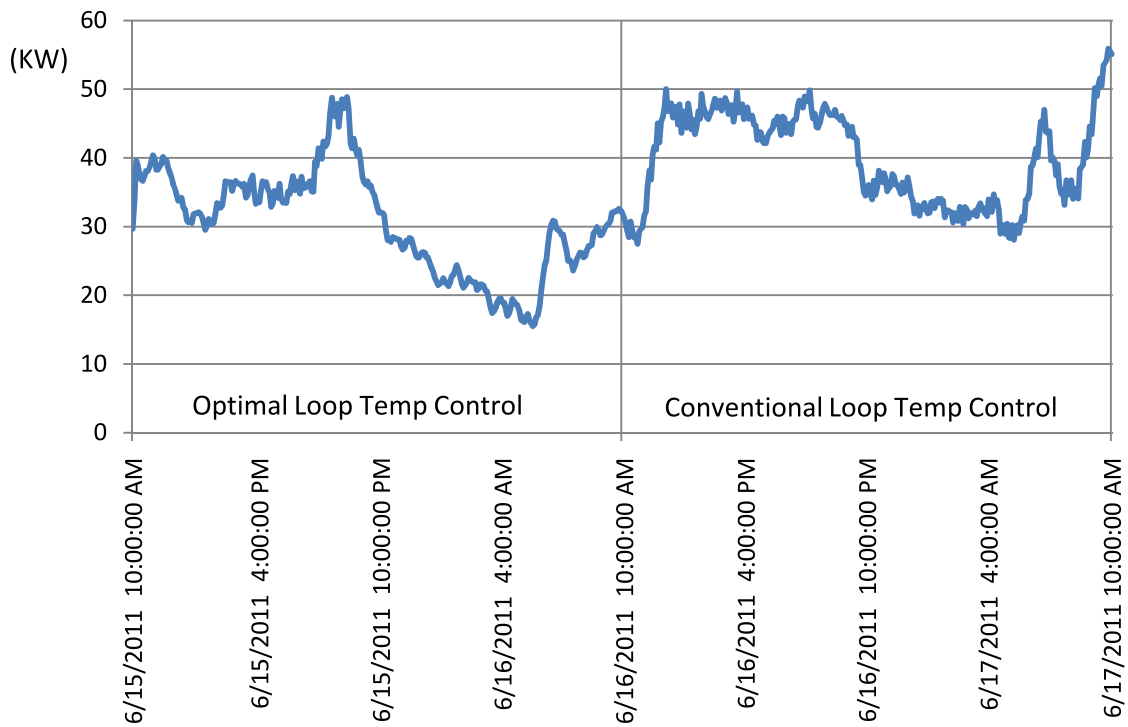

Fig. 4-9 Heat Pump Unit Electricity Consumption Comparison in the Loop Temperature Control Case Study

The KW values were also trended directly for the loop pump and the fluid cooler fan as shown in **Fig. 4-10** and **Fig. 4-11** respectively. The fluid cooler spray pump is constant speed with a 20 horsepower motor. Thus, its electricity consumption was calculated simply using the 20 horsepower multiplied by the time when the spray pump was running.

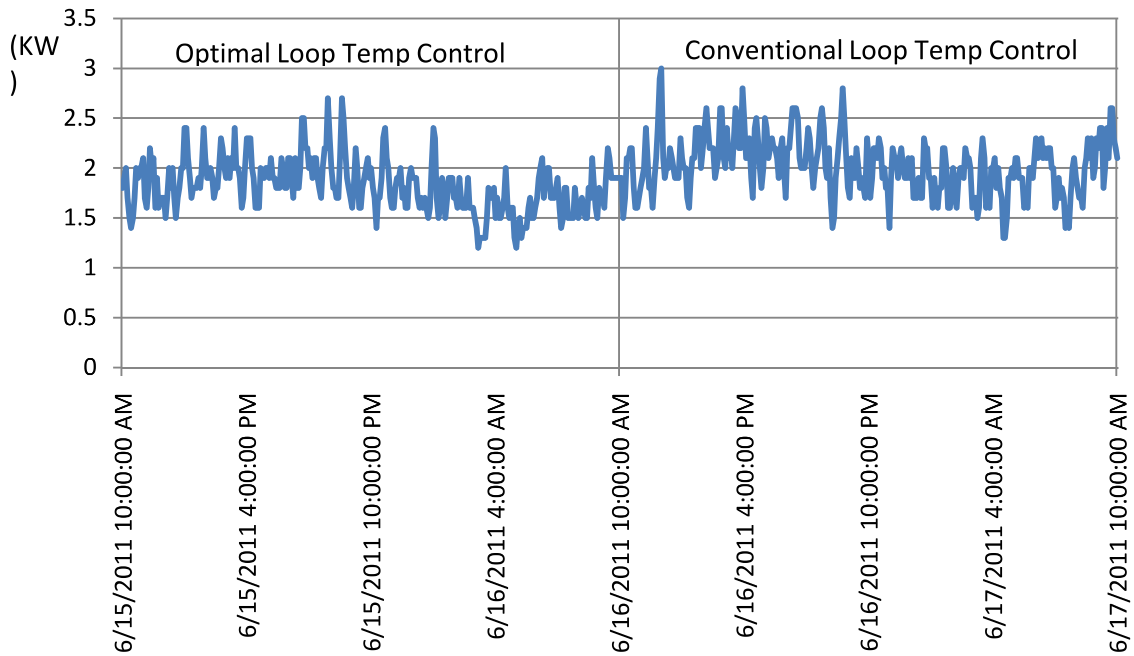

Fig. 4-10 Loop Pump Power Consumption Comparison in the Loop Temperature Control Case Study

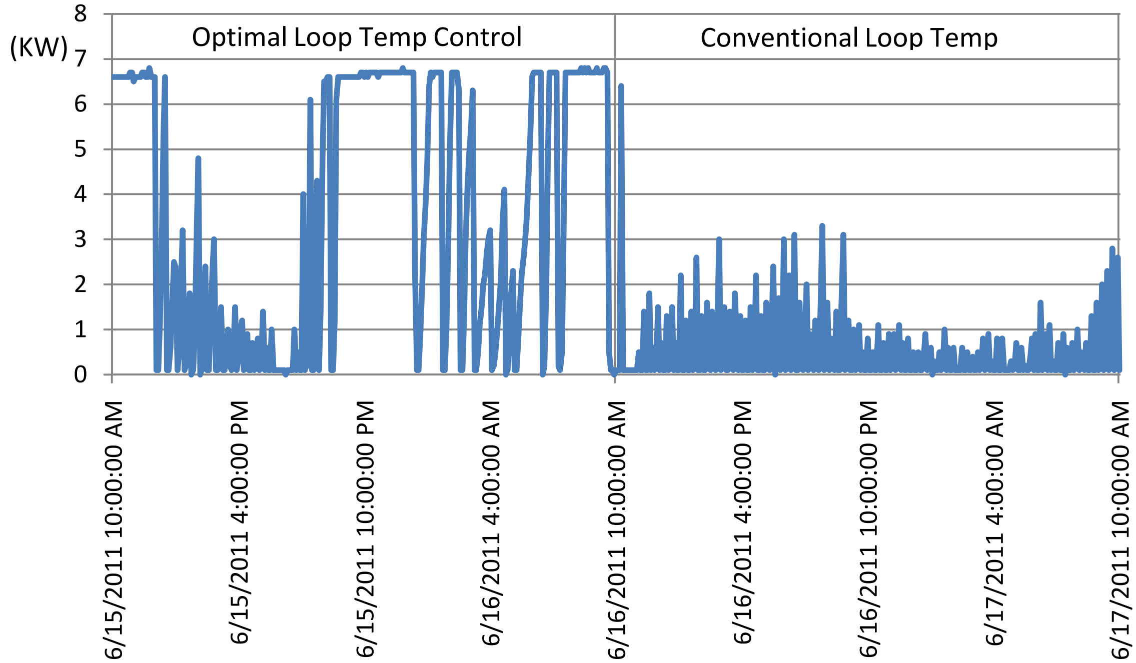

Fig. 4-11 Fluid Cooler Fan Energy Consumption Comparison in the Loop Temperature Control Case Study

The exact electricity consumption for the heat pumps, loop circulation pump, fluid cooler fan, spray pump and the total is listed in **Table 4-2.** The optimal loop temperature control strategy generated 21.8% electricity savings for the heat pump units only. After the combination with the KWh consumption for the fluid cooler and the loop pump, still a total of 12.8% electricity can be saved. In the part load seasons, the savings could be even more than that.

Date	Heat Pump Units (KWh)	Loop Pump (KWh)	Fluid Cooler Fan (KWh)	Fluid Cooler Spray Pump (KWh)	Total (KWh)
June 15—June 16	740.1	43.6	86.4	85.8	955.9
June 16—June 17	946.5	48.8	13.6	87.6	1096.5
Percentage of Savings	21.8%	10.7%	--	2.1%	12.8%

Table 4-2 Electricity Consumption Comparison of the Optimal Loop Temperature Control and the Conventional Loop Temperature Control

4.3.3 Improve the 3-way valve, Fluid Cooler Fan and Spray Pump Control Sequence

The old control sequence of the fluid cooler was based on a linear relationship between the supply water temperature and the cooling signal which determines the fan/spray pump sequencing and the fan speed. For example, if the supply water temperature is 100 °F, then the fan speed is 100%; and if the supply water temperature is 80 °F, then the fan speed is 0%. If the supply water temperature is between 80 °F and 100 °F, the fan speed values are simply interpolated linearly between 0% and 100%. This method is hardly accurate because the actual temperature-fan speed relationship is more complicated than a linear relationship. Thus, a PID loop is applied in the new control sequence to modulate the fan speed to maintain the fluid cooler supply temperature at setpoint. In addition, the new control method makes the sequencing of the 3-way valve, the spray pump and the fans in the fluid cooler more continuous without causing great temperature fluctuation in the water loop.

In the new control sequence, the maximum fluid cooler fan speed was set at 80 % if the return water temperature is below 90 °F and 100% if the return water temperature is above 90 °F. The water cooling effects when the fan is running at 80% and 100% respectively are nearly the same, especially when the outside air temperature is low. However, the power consumption when the fan is running at 80% is almost half of the one when it is at 100% according to the fan affinity law.

Chapter 5 OPTIMIZATION OF THE BUILDING OPERATING SCHEDULE AND THE HEAT PUMP FAN MODE

In this chapter, a case study will be presented to demonstrate the advantage of an optimal building operating schedule and heat pump fan mode. An improved building operating schedule and heat pump fan mode will be given and compared with the existing schedule and fan mode. The actual electricity consumption will also be compared between the new and the old schedules and modes. Finally, the benefit of the new operation method will be discussed.

5. 1 Overview of the Existing Operation Schedule and Mode

The same building used in the previous chapters will be used for the case study again in this chapter. The building was experienced with problems that are related with the building operating schedule and heat pump fan operating mode. The existing schedule allowed too many unnecessary hours to operate the heat pumps. In addition, the heat pump fan was set to run continuously no matter the compressor is activated or not. This can waste much energy especially in the part load seasons when most heat pump compressors work less than two hours in a day. In order to study in detail on these problems, improved operating schedule and fan operating mode will be presented and compared with the existing schedule and mode in terms of daily electricity consumption.

5. 2 Schedule and Mode Optimization Case Study

From Feb. 21, 2011 to Mar. 11, 2011, an experiment was conducted at the Mission Middle School building to analyze the impact of improved building schedule and heat pump fan mode on the electricity consumption. In the experiment, two parameters were modified on a daily basis.

(1) Heat pump supply fan operating mode

There are two modes for the heat pump fan. The fan can be set to either run continuously or cycle along with the heat pump compressor.

(2) Heat Pump Operating Schedule

In the occupied mode, the room set point in the school building varies from 68 °F to 74 °F, with 1 °F of heating/cooling dead band. In the unoccupied mode, the room set point is 65 °F for heating and 78 °F for cooling. The detailed schedule before and after the optimization is listed in **Table 5-1**.

	Existing Schedule	Optimized Schedule
Gym & Auditorium RTUs	24/7	6:00 am--5:00 pm Mon. thru Fri.
Classrooms & Offices on the 2nd Floor HPs	5:30 am--10:00 pm every day	5:30 am--5:00 pm Mon. thru Fri
Classrooms & Offices on the 1st Floor HPs	5:00 am--10:00 pm every day	6:00 am--5:00 pm Mon. thru Fri.
Cafeteria & Kitchen	5:00 am--10:00 pm every day	5:30 am--3:30 pm Mon. thru Fri.

Main Office, Staff Office & Lobby HPs	5:00 am--10:00 pm every day	5:30 am--9:00 pm Mon. thru Fri.
Entrance, Corridor & Stairway EUHs	24/7	5:30 am--9:00 pm Mon. thru Fri.
Outside Air RTUs	8:00 am--4:00 pm Mon. thru Fri.	8:00 am--4:00 pm Mon. thru Fri.

Table 5-1 Schedule Optimization

To minimize the impact of outside air temperature change on the results, the schedule and the fan operating mode were alternated day by day during the experiment. For example, if the system follows the existing schedule and the continuous fan mode today, then in the next day, the system will follow the optimized schedule and the fan will be cycled. The KWh readings on the substation transformer outside the school building have been collected for more than three weeks. A typical day begins at 8:00 am of the first day and ends at 8:00 am of the next day. Outside air temperature was also trended. The detailed results are presented in **Table 5-2**.

The utility data gave a 22.3% saving in electricity for the three weeks in total. But after comparing data from two days with similar weather condition and temperature range (Feb.21 and Feb.24), 23.1% of electricity can be saved by using a different schedule and fan mode. **Fig. 5-1** shows a diagram of daily electricity consumption versus daily average outside air temperature.

Given the possible fact that the outside air served from the five fresh air RTUs may be supplied or “leaked” from the return grille if the heat pump fan stops running, the author did an investigation at the school. After the experiment, the school began to adopt the new schedule and fan mode. There has been no complaint from the students and faculty about the new schedule and fan mode change so far.

Start Time	End Time	Start Point OAT (°F)	OAT Range (°F)	Schedule Type	Heat Pump Fan Cycled/Continuous	Power Consumption (KWh)
Feb.21, 8:00am	Feb.22, 8:00am	22.9	17.6--31.3	Old	Continuous	3900
Feb.22, 8:00am	Feb.23, 8:00am	19.7	19.7--38.5	New	Cycled	2700
Feb.23, 8:00am	Feb.24, 8:00am	33.5	28.1--49.8	Old	Continuous	3000
Feb.24, 8:00am	Feb.25, 8:00am	28.8	16.2--31.1	New	Cycled	3000
Feb.25, 8:00am	Feb.26, 8:00am	18.3	17.6--30.0	Old	Continuous	9000
Feb.26, 8:00am	Feb.27, 8:00am	19.0	19.0--32.7			
Feb.27, 8:00am	Feb.28, 8:00am	32.5	21.8--40.6			
Feb.28, 8:00am	Mar.1, 8:00am	22.4	22.4--38.1	New	Cycled	2700
Mar.1, 8:00am	Mar.2, 8:00am	31.5	18.3--53.5	Old	Continuous	3000
Mar.2, 8:00am	Mar.3, 8:00am	18.5	18.5--39.7	New	Cycled	2700
Mar.3, 8:00am	Mar.4, 8:00am	38.7	35.5--59.1	Old	Continuous	2700
Mar.4, 8:00am	Mar.5, 8:00am	36.5	18.3--36.6	New	Cycled	6000
Mar.5, 8:00am	Mar.6, 8:00am	18.8	18.8--35.9			
Mar.6, 8:00am	Mar.7, 8:00am	29.2	29.2--45.5			
Mar.7, 8:00am	Mar.8, 8:00am	36.8	36.8--53.3	Old	Continuous	3300
Mar.8, 8:00am	Mar.9, 8:00am	38.3	35.3--41.7	New	Cycled	2400
Mar.9, 8:00am	Mar.10, 8:00am	35.3	23.9--40.6	Old	Continuous	3300
Mar.10, 8:00am	Mar.11, 8:00am	26.3	26.3--51.8	New	Cycled	3200

Table 5-2 Daily Electricity Consumption Data with Both Schedule and Fan Mode Change

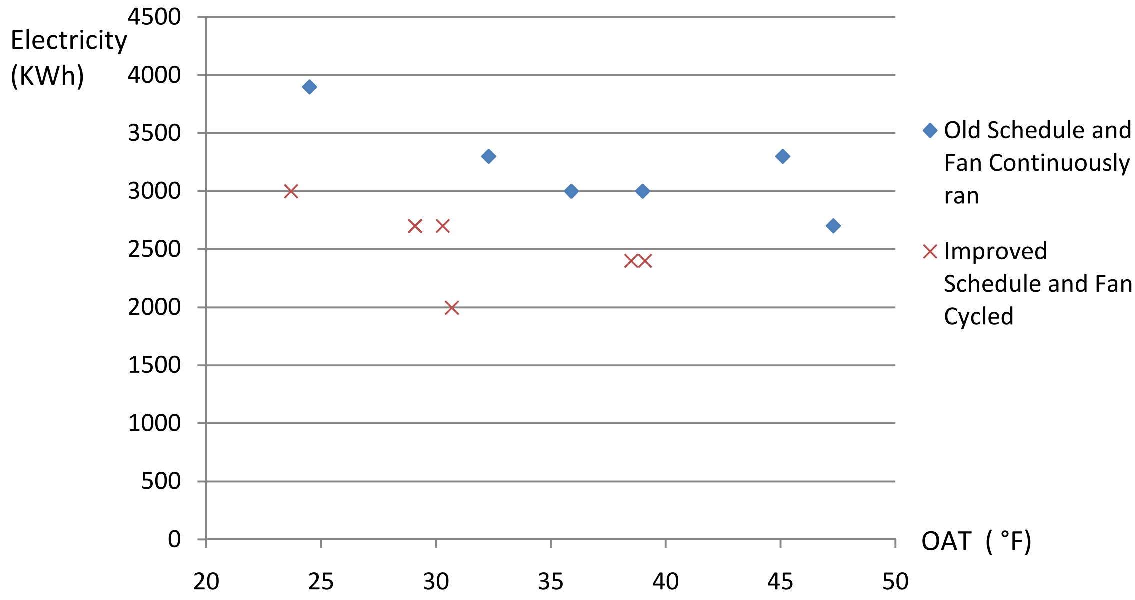

Fig. 5-1 Electricity Consumption Comparison with both Schedule and Fan Mode Change

From May 5, 2011 to May 24, 2011, another experiment was performed to quantify the electricity savings of the new fan cycled mode compared with the old fan continuous mode. The new building operating schedule was applied throughout the entire experiment. The detailed results are shown in **Table 5-3.** The daily electricity consumption diagram can also be seen in **Fig. 5-2**.

18.5% of electricity savings were achieved using the cycled fan mode compared with the continuous fan mode during the experiment, although the outside air temperature varied greatly during May. From May 16 to May 17 when the outside air temperature ranged between 47.7 °F and 80 °F, the author measured the actual heat pump compressor operating hours as shown in **Fig.**

5-3. The diagram indicates that most heat pump compressors worked less than 2 hours in this outside air temperature range.

Start Time	End Time	Start Point OAT (°F)	OAT Range (°F)	HP Fan Mode Cycled/Continuous	Power Consumption (KWh)
May. 5, 8:00am	May. 6, 8:00am	54.9	47.0--75.8	Cycled	2100
May. 6, 8:00am	May. 7, 8:00am	51.3	51.3--82.1	Continuous	6900
May. 7, 8:00am	May. 8, 8:00am	65.2	61.9--90.2		
May. 8, 8:00am	May. 9, 8:00am	63.8	63.8--93.2		
May. 9, 8:00am	May. 10, 8:00am	73.4	73.6--103.5	Continuous	3900
May. 10, 8:00am	May. 11, 8:00am	77.1	68.1--100.5	Cycled	3600
May. 11, 8:00am	May. 12, 8:00am	69.2	64.1--84.5	Cycled	3000
May. 12, 8:00am	May. 13, 8:00am	66.0	49.7--87.0	Continuous	3300
May. 13, 8:00am	May. 14, 8:00am	49.8	46.8--73.0	Cycled	3900
May. 14, 8:00am	May, 15, 8:00am	48.0	46.0--55.9		
May, 15, 8:00am	May. 16, 8:00am	49.0	48.5--73.6		
May. 16, 8:00am	May. 17, 8:00am	49.5	47.7--80.0	Continuous	3000
May. 17, 8:00am	May. 18, 8:00am	52.8	52.8--83.1	Cycled	2700
May. 18, 8:00am	May. 19, 8:00am	56.7	60.0--77.3	Continuous	3300
May. 19, 8:00am	May. 20, 8:00am	61.5	61.5--77.6	Cycled	2800
May. 20, 8:00am	May. 21, 8:00am	63.7	63.7--70.6	Cycled	5100
May. 21, 8:00am	May. 22, 8:00am	67.0	62.7--90.4		
May. 22, 8:00am	May. 23, 8:00am	66.7	62.7--92.3		
May. 23, 8:00am	May. 24, 8:00am	65.1	65.1--91.4	Continuous	3600

Table 5-3 Daily Electricity Consumption Data with Fan Mode Change Alone

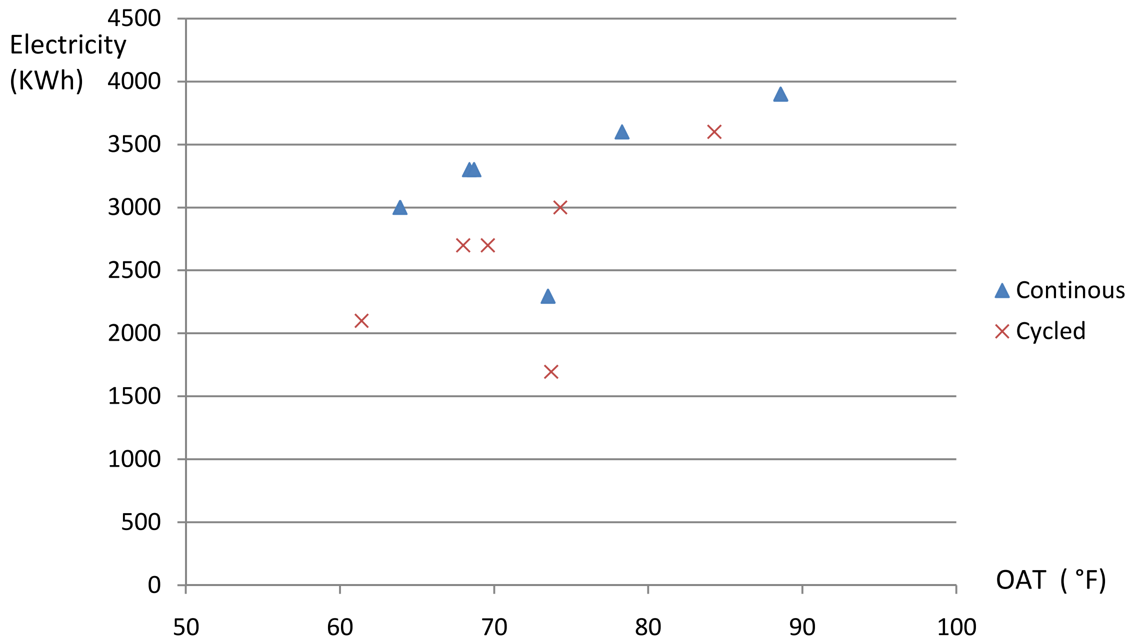

Fig. 5-2 Electricity Consumption Comparison with Fan Mode Change Alone

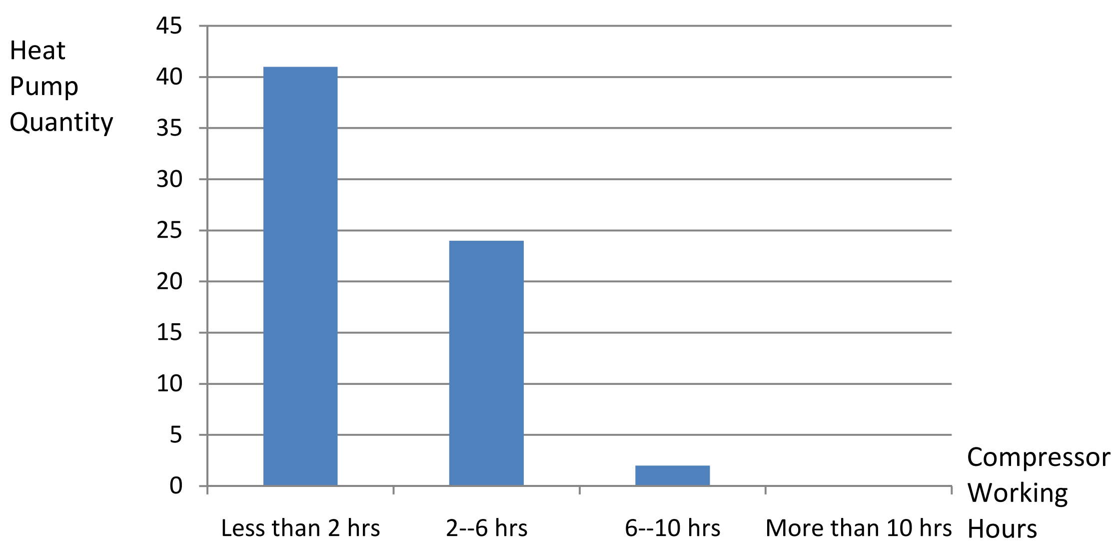

Fig. 5-3 Heat Pump Compressor Working Hours in a Typical School Day

The schedule and fan mode optimization was carried out in February 2011 and has been in effect since then. Accordingly, the building utility data from February to April of 2010 and 2011 provided from Omaha Public Power District (OPPD) are listed in **Table 5-4** and illustrated in **Fig. 5-4**. It can be seen that there is a 24% difference of electricity consumption before and after the schedule and fan mode optimization.

	February	March	April	3 Months Total
2010	122,800 KWh	110,800 KWh	96,800 KWh	330,400 KWh
2011	98,800 KWh	80,800 KWh	71,600 KWh	251,200 KWh
Percentage Savings	19.5%	27.1%	26.0%	24.0%

Table 5-4 Electricity Consumption Comparison before and after the Schedule and Fan Mode Optimization

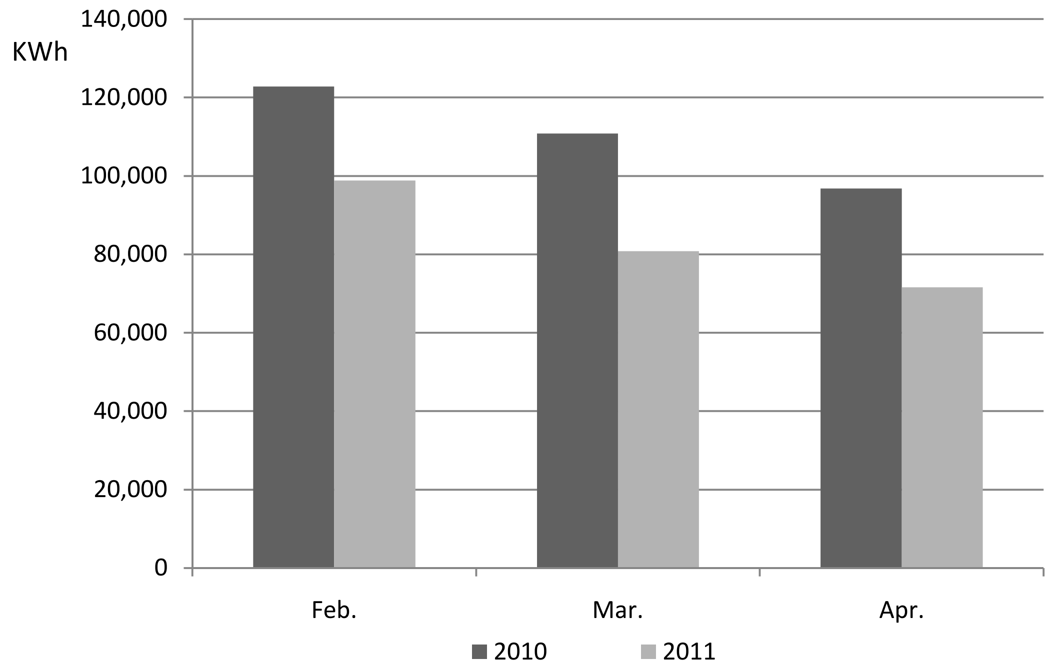

Fig. 5-4 Electricity Consumption Comparison before and after the Schedule and Fan Mode Optimization

For a better prediction of the annual electricity savings, A Carrier HAP Model is created for both the new and the old schedules and fan modes. The detailed simulation process and results can be found in **Appendix B**.

Although there are some activities after class or on weekends, most of classrooms and offices are not in use before 7 a.m. and after 4 p.m. and on weekends. The building operating schedule can modified directly on the energy management and control system (EMCS) to meet the load of the particular zone that is in use while making other zones at “unoccupied” status. More importantly, this is achieved simply by a few mouse clicks and drags on the EMCS computer interface.

The experiment and the HAP model results agreed very well in terms of the percentage of electricity savings. The both experiments were conducted during the school times. However, during school breaks, more energy savings can be predicted since the schedule and the fan mode can both be adjusted according to the building occupancy. Given the results from both the energy simulation and the actual electricity measurement, around 20% to 30% of electricity savings can be expected from the optimization of the system operating schedule and the heat pump fan mode.

Chapter 6 CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

The thesis mainly focused on how to control the water loop heat pump system to make it operate in an energy-efficient way. Several studies and experiments were performed on a school building to testify the advantages of several optimal control strategies including the DP reset pump control method, the fluid cooler supply temperature reset method, the improved building operation schedule heat pump fan mode.

- By using the DP reset pump control method, the pump speed was reduced by over 15% and 60.9% of pumping KWh energy was saved. Depending on the actual building load, 40% to 70% of pump energy savings can be expected compared with the constant DP control method. Moreover, these savings are achieved without sacrificing the heat pump efficiency. On the other hand, the outside air temperature and the occupancy level can influence the reset DP setpoint and the pump speed. The higher the OAT or the occupancy level, the greater the DP setpoint and the pump speed.
- The water loop temperature optimization and new fluid cooler control strategy that resets the loop supply water temperature offered 12.8% of electricity savings on the WLHP system alone compared with the existing loop temperature control method using an 80 °F supply water

setpoint. The experiment was conducted during a summer day when the outside air temperature ranged from 68 °F to 99 °F. More heat pump power can be saved if the outside air temperature is lower.

- The schedule and heat pump fan operation improvement can generate a total of 20% to 30% electricity savings. There are about 15% to 20% of electricity savings when the fan mode is changed from continuously running to cycling with the heat pump compressor. The amount of savings can be even more in the lightly-occupied or unoccupied periods such as school breaks.

6.2 Recommendations for Future Research

Given the short time range, it is impossible to collect sufficient utility data to quantify the actual annual energy savings. However, the short time in-situ experiment data and the energy simulation results were good enough to prove the benefits using these optimized control strategies.

Future research may include the following aspects:

- Besides the reset DP pump control method, the loop pump speed can also be modulated based on a certain reset loop temperature difference. This variable temperature difference setpoint should be a function of the actual building load and the outside weather condition. Research in specifying the proper Delta-T setpoint to operate the pump will be an interesting and practical topic, as long as enough on site temperature data can be gathered. The reset Delta-T method can be combined with the DP reset method for an even better pump control.

- The OAT-10 °F supply temperature reset is an easy and effective method for controlling the fluid cooler/cooling tower. However, to obtain a more accurate supply water setpoint, more outside air temperature and supply water temperature data should be collected. In this case, an OAT-SWT model can be built up for a better operation sequence of the fluid cooler/cooling tower.

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APPENDIX A

Pumps									
Mark	Serves	Pump Type	GPM	Head (ft)	Minimum Efficiency	Motor Data			
						HP	V	PH	RPM
P-1	HP loop	centrifugal	400	130	77	25	208	3	1750
P-2	HP loop	centrifugal	400	130	77	25	208	3	1750
P-3	Boilers	centrifugal	130	20	60	1.5	208	1	1750

Table A-1 Pump Data for the Case Study Building

Variable Frequency Drive					
Mark	Serves	Type	Motor Data		
			HP	V	PH
VFD-1	P-2	variable torque AC	25	208	3
VFD-2	P-3	variable torque AC	25	208	3
VFD-3	CFC-1	variable torque AC	10	208	3

Table A-2 VFD Data in the WLHP system for the Case Study Building

Closed Circuit Fluid Cooler						
Mark	Coil Data				Fan Data	
	GPM	EWT (°F)	LWT (°F)	Pressure Diff (psi)	Flow (cfm)	Type
CFC-1	400	101	85	15.6	83180	Prop

Ambient WB Temp (°F)	Motor Data				Spray Pump Data		
	NO	HP each	V	PH	HP	V	PH
78	2	10	208	3	5	208	3

Table A-3 Fluid Cooler Data for the Case Study Building

Boiler					
Mark	GPM	EMT (°F)	LMT (°F)	Input (MBH)	Output (MBH)
B-1	70	65	120	1500	1290—1409
B-2	70	65	120	1500	1290—1409

Table A-4 Boiler Data for the Case Study Building

APPENDIX B

To find out how the heat pump operating schedule can influence the energy consumption of a school building, an energy simulation study using Carrier Hourly Analysis Program (HAP) was conducted. The simulation covered all 68 heat pump units and nine RTUs (including five with heat recovery wheels) in the building.

1. Data Input

Similar with Trane Trace 700 or other building energy simulation software, there are several levels in Carrier HAP for design and simulation data input such as weather data, spaces, systems, plants and buildings. The spaces are assigned to a system. In the same manner, the systems are assigned to a plant and finally the plants are assigned to a building. A simple step by step procedure for data input is shown below.

1.1 Weather

Since there is no weather data for Bellevue, NE in the HAP program, the weather data for Omaha WSO, NE was applied. The weather conditions for the two places are almost the same, since Bellevue is very close to Omaha, NE. **Fig. B-1** shows the weather properties of Omaha, NE.

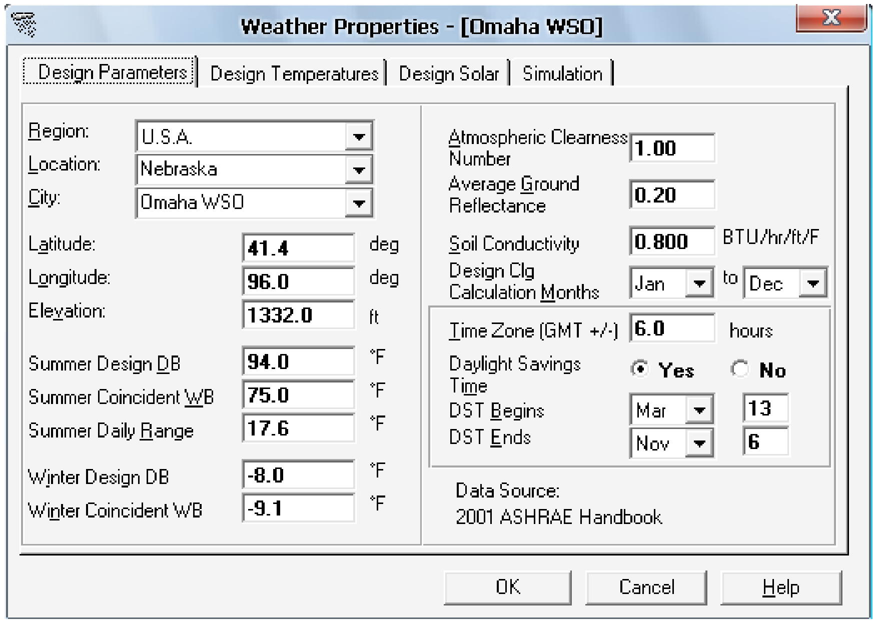

Fig. B-1 Weather Properties for Omaha, NE

1.2 Spaces

(1) Dimension & Orientation

Define rooms that are served by the 68 heat pumps. Room dimensions and area were measured and calculated. Ceiling height is 10 ft. Plenum height is 2 ft. Use *ASHARE Standard 62.1-2004 Ventilation of Acceptable Indoor Air Quality* which considers both cfm per person and cfm per sq ft to calculate the design outside air volume flow rate.

(2) Envelope

The U values and R values of the envelope components are shown in **Table B-1** and **Fig. B-2**. Since the building drawings and architectural schedules are inadequate, the U values for the windows, doors, slab on grade and wall below grade were assumed according to the baseline values from *ASHARE 90.1-2007 Energy Standard for Buildings except Low-rise Residential Building*s.

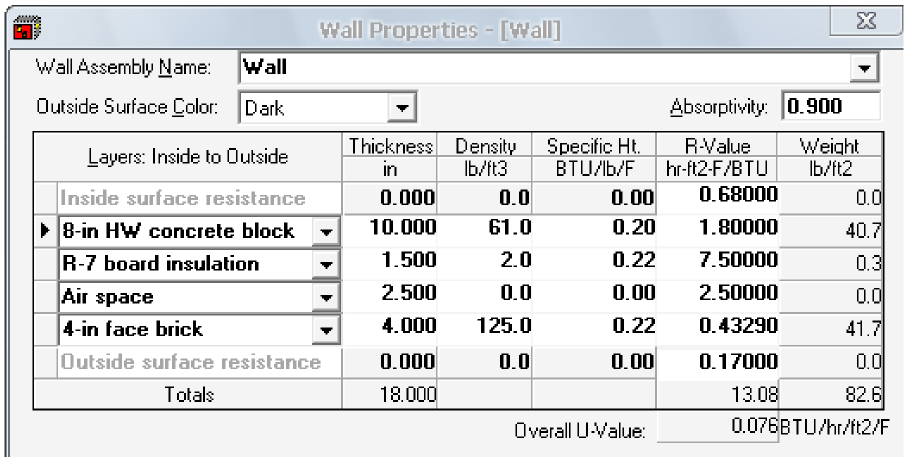

Fig. B-2 Wall Properties

Envelope Thermal Parameters	R Value	U Value	R Value from ASHRAE 90.1	U Value from ASHARE 90.1
	h*ft^2*F/Btu	Btu/(h*ft^2*F)	h*ft^2*F/Btu	Btu/(h*ft^2*F)
Wall	13.08	0.0765	R-11	--
Slab on grade	14	0.0714	R-15 for 24 in	--
Roof	20.42	0.049	R-20	--
Window	--	0.4	--	U-0.45 (Metal)
Door	--	0.5	--	U-0.7 (Swinging)
Wall below grade	10	--	R-7.5 (c.i.)	--

Table B-1 Envelope Thermal Parameters

(3) People and Schedule

Define the density of people and the occupancy schedule based on the function of the particular zone. For instance, use 20 sq ft/person as people density for general classrooms. For general offices, use 143 sq ft/person as people density; while for the cafeteria, use 10 sq ft/person as people density. Select "office work" as people activity level for the majority of the zones in which the sensible heat load is 245 Btu/(hr*person) and the latent heat load is 205 Btu/(hr*person). **Fig. B-3** demonstrates a typical classroom occupancy schedule.

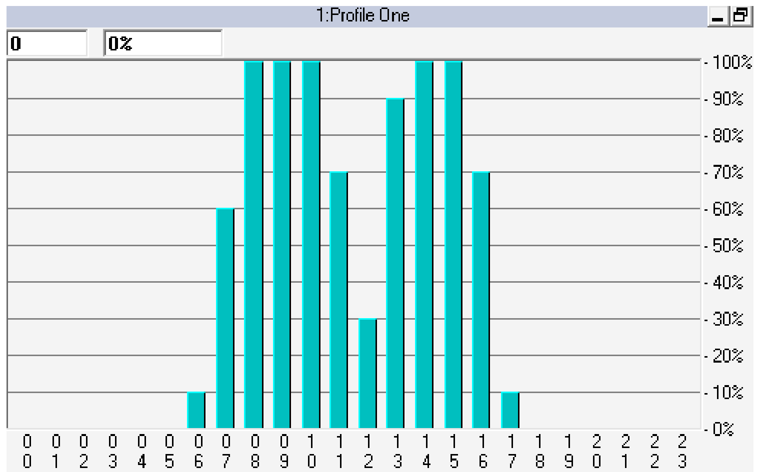

Fig. B-3 Classroom Occupancy Schedule

(4) Lighting & Equipment

Use no-vented recessed fluorescent with 80% load to space. The lighting load is 1 W/sq ft. For classrooms, use standard school equipment which is 0.22 W/sq ft. For offices, use standard office equipment which is 0.5 W/sq ft. For others, no equipment load is applied.

(5) Miscellaneous

No miscellaneous load was applied in the calculating process.

1.3 Systems and Plants

Define five systems (RTU-5, RTU-6, RTU-7, RTU-8 and RTU-9) which are five outdoor air roof top units with heat recovery wheels. Each RTU is linked to the heat pump sub-systems. See **Fig. B-4**, **Fig. B-5** and **Fig. B-6** for the system configuration setup window. The heat pump and RTU equipment data along with the system and plant data were obtained from Mission Middle School mechanical specifications and manufacturer's catalog.

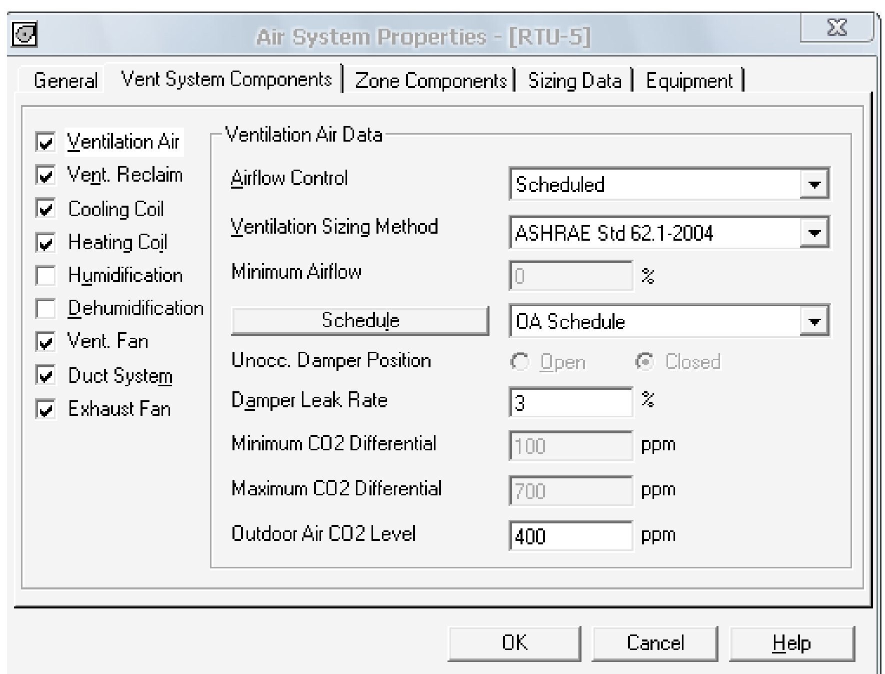

Fig. B-4 System Configuration

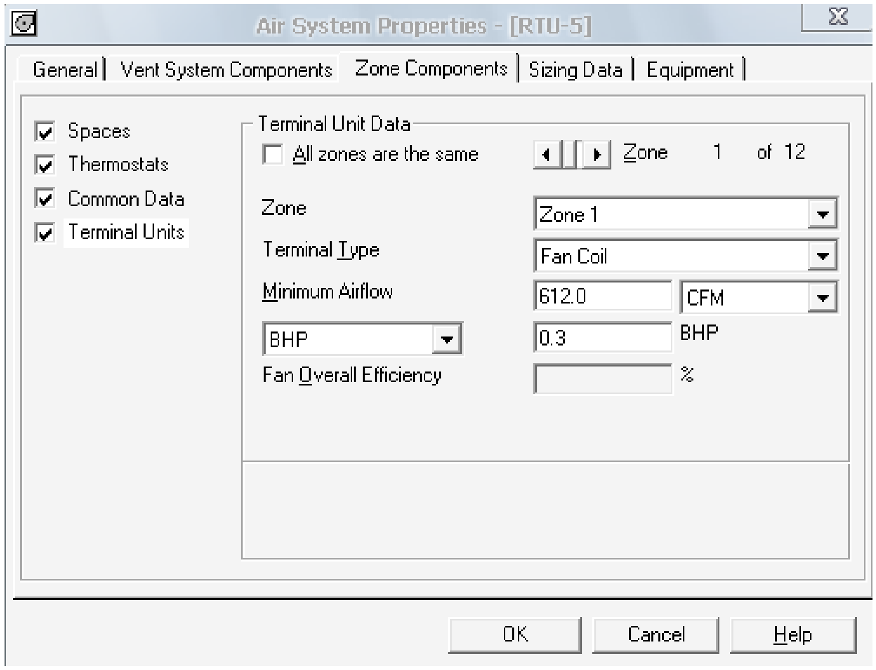

Fig. B-5 Zone Components

For the thermostat setup, set the cooling occupied setpoint at 71 °F and cooling unoccupied setpoint at 78 °F; set the heating occupied setpoint at 69 °F and heating unoccupied setpoint at 65 °F. Thermostat throttling range is 1 °F. Put the occupancy schedule to match the actual schedule of each room.

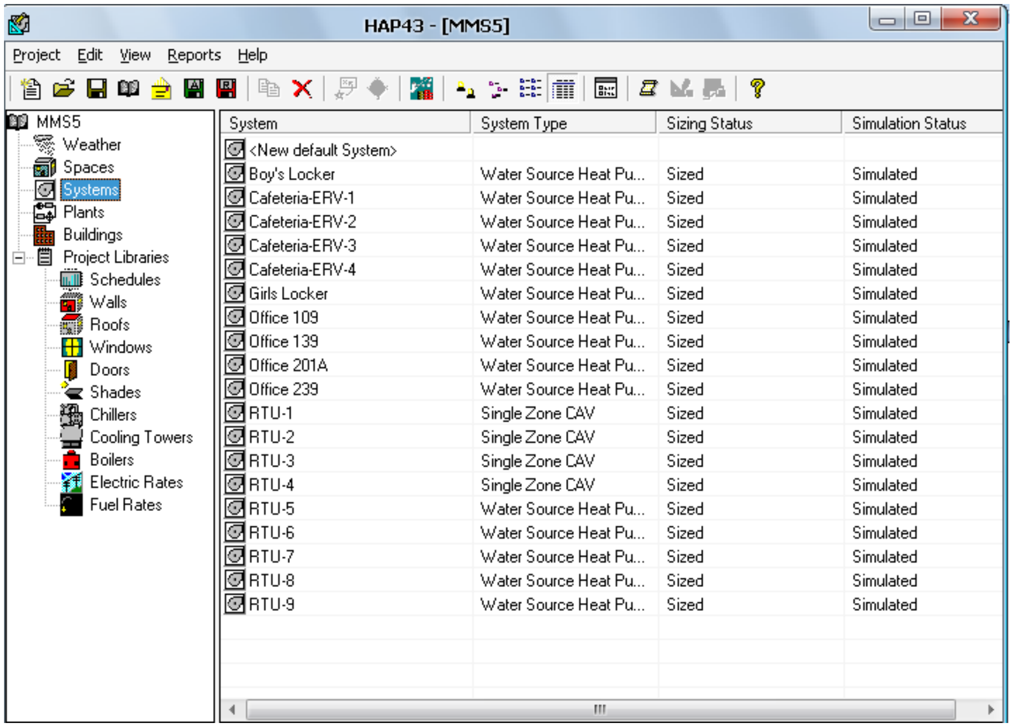

Fig. B-6 System List

1.4 Buildings

Add all the systems to the building. Assume the local electricity charge is $0.06 per KW, with 7% tax rate. Use "Standard Charge" instead of "Compound Charge". Assume the local natural gas charge is $0.7 per therm. Also use "Standard Charge" instead of "Compound Charge".

2. Data Analysis and Simulation Results

The daily operating schedule change and the heat pump fan mode change were both applied during the simulation. The electricity consumption before and after the change were compared.

(1) Daily Schedule Change

Change the daily schedule of all heat pumps units as shown on **Fig. B-7**. For example, before the change, the heat pumps on the first floor follow the occupied heating and cooling setpoint from 5:00 a.m. to 10:00 p.m. After the change, the occupied time was shortened by six hours from 6:00 a.m. to 5:00 p.m. Schedules of other HVAC equipments were also modified in the simulation based on Table B-2.

Fig. B-7 Heat Pump Daily Schedule Change

(2) Heat Pump Fan Control Mode Change

Change all heat pump fan control mode on from “Fan On” to “Fan Cycled” as shown on **Fig. B-8** to see the electricity consumption when the fans are running continuously and when the fans are cycling with the compressor.

Fig. B-8 Fan Control Mode Change

The HAP simulation results are shown in **Table B-2** for both the schedule and the fan mode optimization and **Table B-3** for the fan mode optimization alone. The results show that around 28% electricity savings can be achieved if the improved operating schedule and fan mode are used, and 17% of electricity can be saved by simply changing only the fan operating mode.

	Old Schedule & Continuous Fan Mode	New Schedule & Cycled Fan Mode	Percentage of Savings
Electricity (KWh)	973,025	699,285	28.1%
Grand Total ($)	95,416	72,993	23.5%

Table B-2 HAP Simulation for both Fan Mode and Schedule Optimization

	Continuous Fan Mode	Cycled Fan Mode	Percentage of Savings
Electricity (KWh)	973,025	80,566	17.2%
Grand Total ($)	95,416	81,581	14.5%

Table B-3 HAP Simulation for Fan Mode Optimization Only