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Analysis of the HVAC System at the Willow Branch Intermediate School

Final Report

MEEN 685

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Presented to
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by

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Executive Summary

This report presents an analysis of the HVAC system at the Willow Branch Intermediate School for the MEEN 685 class project. The school is located at College Station, Texas. A portion of the school belonged to Oakwood Intermediate School which was built in 1983. It was recently expanded in 1994 and renamed the Willow Branch Intermediate School. It now has a total floor area of 88,617 square feet. The system under investigation is a water-loop heat pump system which provides the HVAC needs for the new building.

In the first section of this report, the energy consumption of the Willow Branch Intermediate School is compared with other similar schools across Texas. It is found that the school's electricity consumption is average and that gas consumption is low, which is to be expected for a water-loop heat pump system with supplemental gas boilers.

The school has experienced a high humidity problem ever since it was first put into operation. In response to this problem, the school district's HVAC system operational staff closed the fresh air vents to prevent outside moisture from entering the building. This may have improved the high humidity problem but it caused new indoor air quality problems from the reduced fresh air. This was confirmed by CO_2 concentrations as high as 2,000 PPM measured inside the building, which is well above the American Society of Heating, Refrigeration, and Air-Conditioning Engineers(ASHRAE) recommended 1,000 PPM upper limit.

Manual recalculation of the peak cooling and heating loads shows that the system's cooling and heating capacities are adequate to meet the peak loads as designed. In addition, on-site measurements confirm that the room temperatures are being well maintained at the set point. Unfortunately, high relative humidity problem still exists even with the fresh air vents shut during hot and humid ambient conditions. The high relative humidity problem seems to be caused by the thermostatically-controlled cycling of the heat pumps at partial loads, which results in a higher average cooling coil temperature that is not cold enough to drive the moisture out of the air. Although the act of blocking the outside air vents did reduce the moisture attributable to the outside air, it also reduced the total cooling load which increases the cycling of the heat pumps.

A test was conducted to determine how the heat pump performs under reduced air flow. This test was intended to confirm lower average supply air temperature from reduced air flow rates which would reduce humidity levels. Unfortunately, the cooling season was over before the test could be conducted. However, the test did confirm that the heat pump works well in the heating mode under reduced air flow condition.

To remedy the high indoor humidity problem, it is recommended that separate fresh air pre-conditioning (i.e. heat pump) units to be installed to remove the moisture in the ventilation air before it is injected into the return air. Until this can be accomplished, we suggest that the fresh air vents be opened as soon as possible and that all the exhaust fans been turned off during unoccupied periods. Finally, it is also suggested that the cooling tower water be treated to avoid such problems as Legionnaires' disease. We would also like to recommend that the CSISD investigate installing a variable frequency drive (VFD) on the water-loop pumps which currently use 30hp when operating.

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1. Introduction

The purpose of this report is to analyze the HVAC system at the Willow Branch Intermediate School for an MEEN 685 class project. This report compares the energy consumption of Willow Branch Intermediate School with eleven other schools across Texas; discusses the HVAC system designed for this school, and analyzes the existing problems with the current HVAC system. Some common HVAC systems that are found in secondary schools are discussed in the appendix. Finally, some recommendations are made to fix the high humidity problems and reduce pumping energy use.

1.1. School Description

The Willow Branch Intermediate School is located at College Station, Texas. Portions of the school was formerly part of the Oakwood Intermediate School which was built in 1983. It was recently expanded in 1994 and renamed the Willow Branch Intermediate School. It consists of four separate buildings with a total floor area of 88,617 square feet. The single-story main building is 54,000 square feet and houses 502 sixth-grade students and 44 teachers. It is served by a water-loop heat pump system that has two 570,000 Btu/hr boilers to provide supplemental heating and a cooling tower for heat rejection.

Figure 1: *Willow Branch Intermediate School.*

2. Analysis of Willow Branch Intermediate School

2.1. Comparison of Utility Bills with Eleven Other Similar Schools

Two years of monthly electricity bills and one year of natural gas bills for the Willow Branch Intermediate School were gathered in order to compare its energy consumption with other primary and secondary public schools in similar climates. Figures 2 and 4 present the energy consumption pattern of eleven primary and secondary public schools in Texas from a study by Landman and Haberl (1996) conducted for the US Department of Energy (DOE). The schools in that study are labeled as follows:

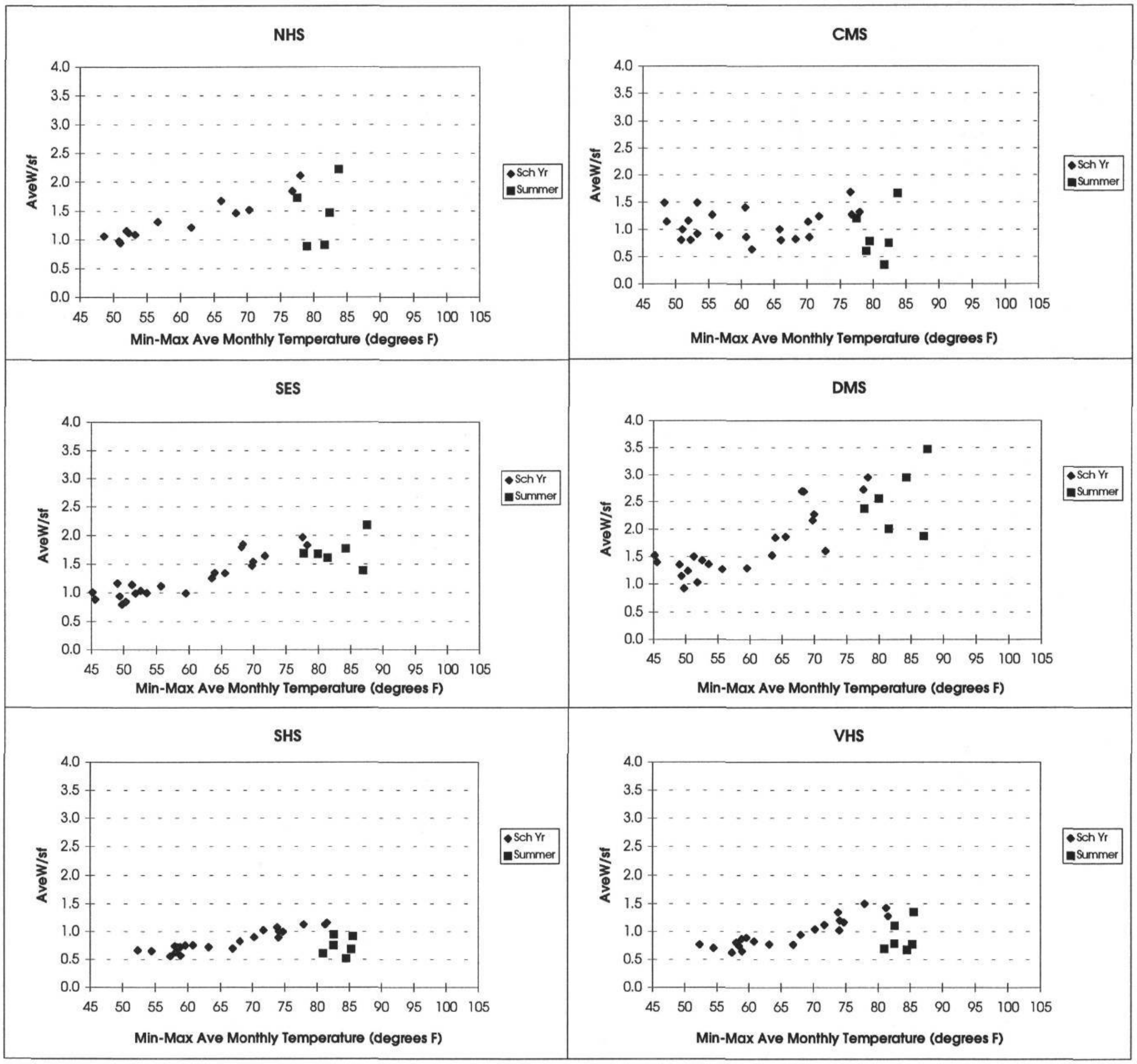
School Initials	School Name	Location	School Dist.	Weather Station
SHS	Stroman High School	Victoria, Texas	VISD	VCT
VHS	Victoria High School	Victoria, Texas	VISD	VCT
SES	Sims Elementary School	Fort Worth, Texas	FWISD	DFW
DMS	Dunbar Middle School	Fort Worth, Texas	FWISD	DFW
NHS	Nacogdoches High School	Nacogdoches, Texas	NISD	LFK
CMS	Chamberlain Middle School	Nacogdoches, Texas	NISD	LFK
OES	Oppe Elementary School	Galveston, Texas	GISD	GLS
WMS	Weis Middle School	Galveston, Texas	GISD	GLS
PES	Parker Elementary School	Galveston, Texas	GISD	GLS
MES	Morgan Elementary School	Galveston, Texas	GISD	GLS
RES	Rosenberg Elementary School	Galveston, Texas	GISD	GLS

Figure 2a and 2b show monthly average electricity consumption in W/sf for consecutive months from September 1991 through December 1993. These are plotted versus the average min-max monthly dry bulb temperature. The diamonds stand for school year and the squares stand for non-school year (summer). Monthly min-max temperature represents the average value calculated from daily temperatures recorded by the National Weather Service (NWS) for each day from a nearby weather station and averaged for each month. The average W/sf is from 0.25 to 3.5 W/sf with the majority of the schools consuming between 0.5 and 2.0 W/sf. DMS has the highest electricity consumption and SHS has the lowest electricity consumption.

Figure 3 shows monthly average electricity consumption in W/sf for Willow Branch Intermediate School for consecutive months from September 1994 through August 1996 with the corresponding 3-

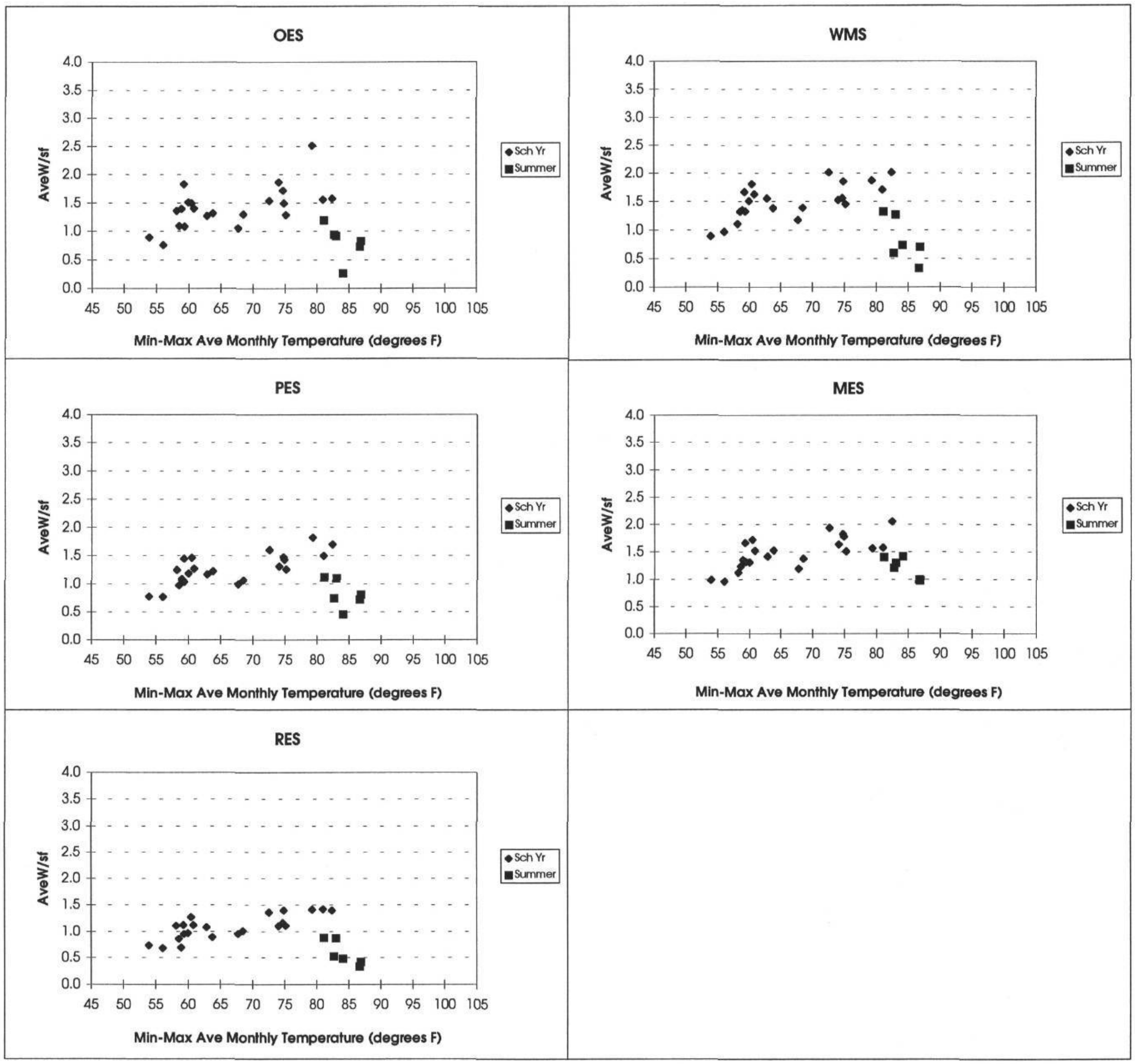
parameter change-point regression model (Kissock 1993) for the school year data. It can be seen that the average electricity consumption for Willow Branch ranges from 0.9 to 1.6 W/sf, which is average when compared to the eleven schools in Figure 2. Summertime electricity use averages about 0.9 W/sf which would seem to indicate that systems are being turned-off to conserve energy.

Figure 2a: *Monthly Ave Consumption: W/sf versus min-max average monthly temperatures for September 1991 through December 1993.*



(*Source:* Landman and Haberl 1996)

Figure 2b: *Monthly Ave Consumption: W/sf min-max average monthly temperatures for 1991 through December 1993.*



(*Source:* Landman and Haberl 1996)

Figure 3: *Monthly Average Consumption: W/sf versus monthly billing period average temperatures for September 1994 through August 1996 for Willow Branch Intermediate School.*

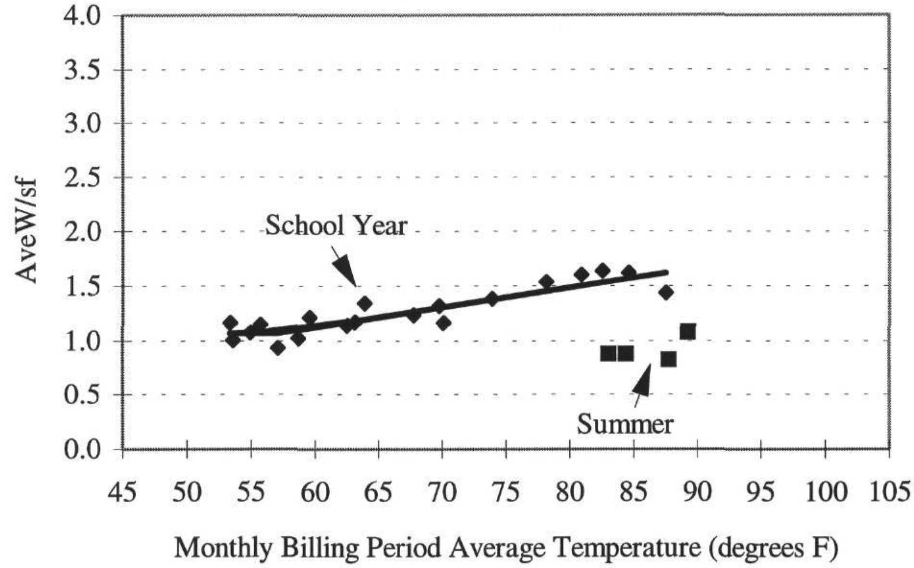
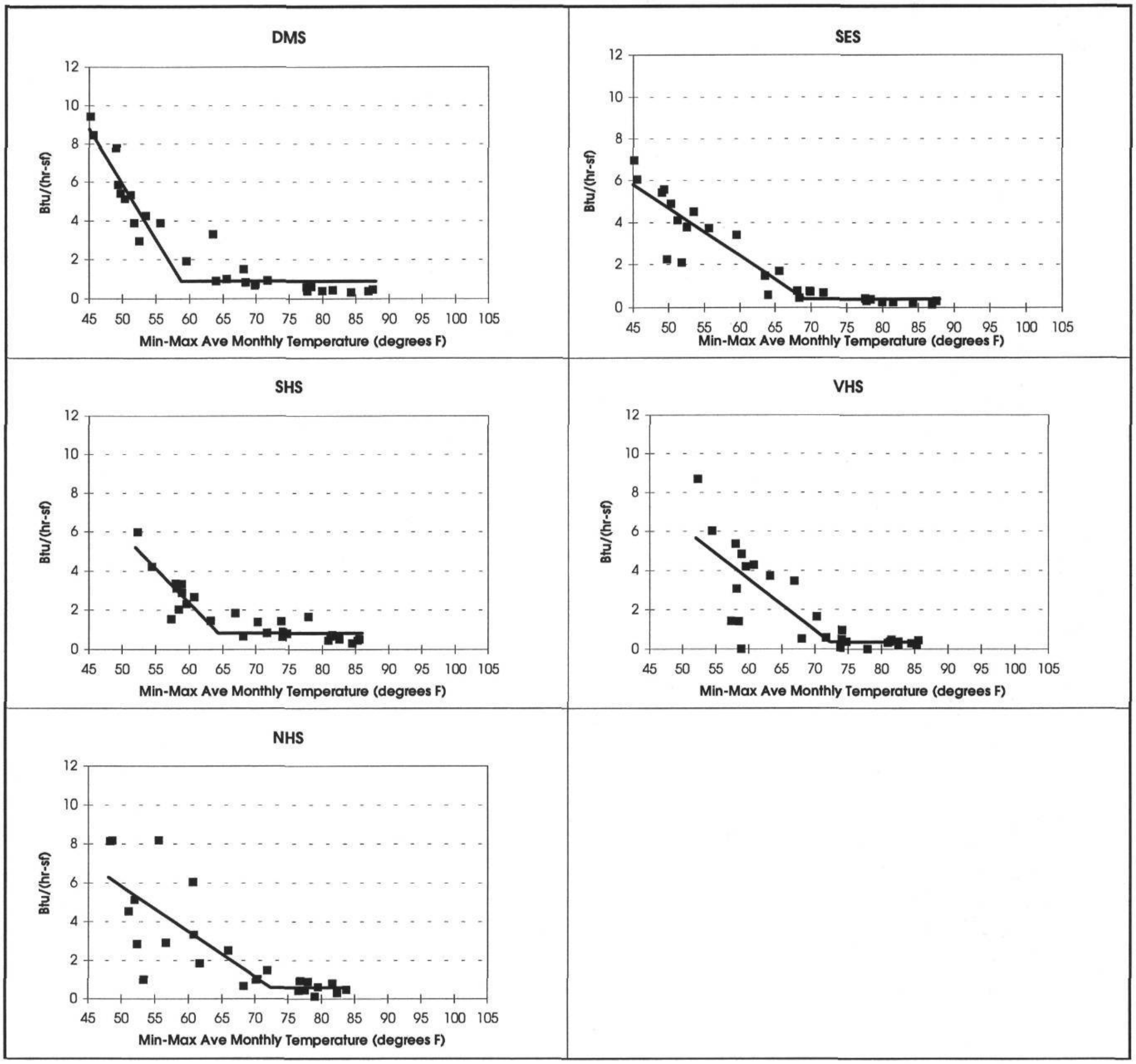


Figure 4a and 4b show monthly average natural gas consumption for the same eleven schools in Btu/(hr-sf) for consecutive months from September 1991 through December 1993 with the corresponding 3-parameter change-point regression model. These are plotted versus the average min-max monthly dry bulb temperature. The average Btu/(hr-sf) is from 0.0 to 12.0 with the majority of the schools consuming under 9.0 Btu/(hr-sf) during the heating season and under 2.0 Btu/(hr-sf) during the summer time. The average natural gas consumption is approximately 3.5 Btu/(hr-sf)

Figure 5 shows monthly average natural gas consumption in Btu/(hr-sf) for the Willow Branch Intermediate School from September 1995 through August 1996 with the corresponding 3-parameter change-point regression model. It can be seen that the natural gas consumption for Willow Branch is low compared with other schools. This is to be expected since a portion of the heating energy is being provided by zonal heat pumps.

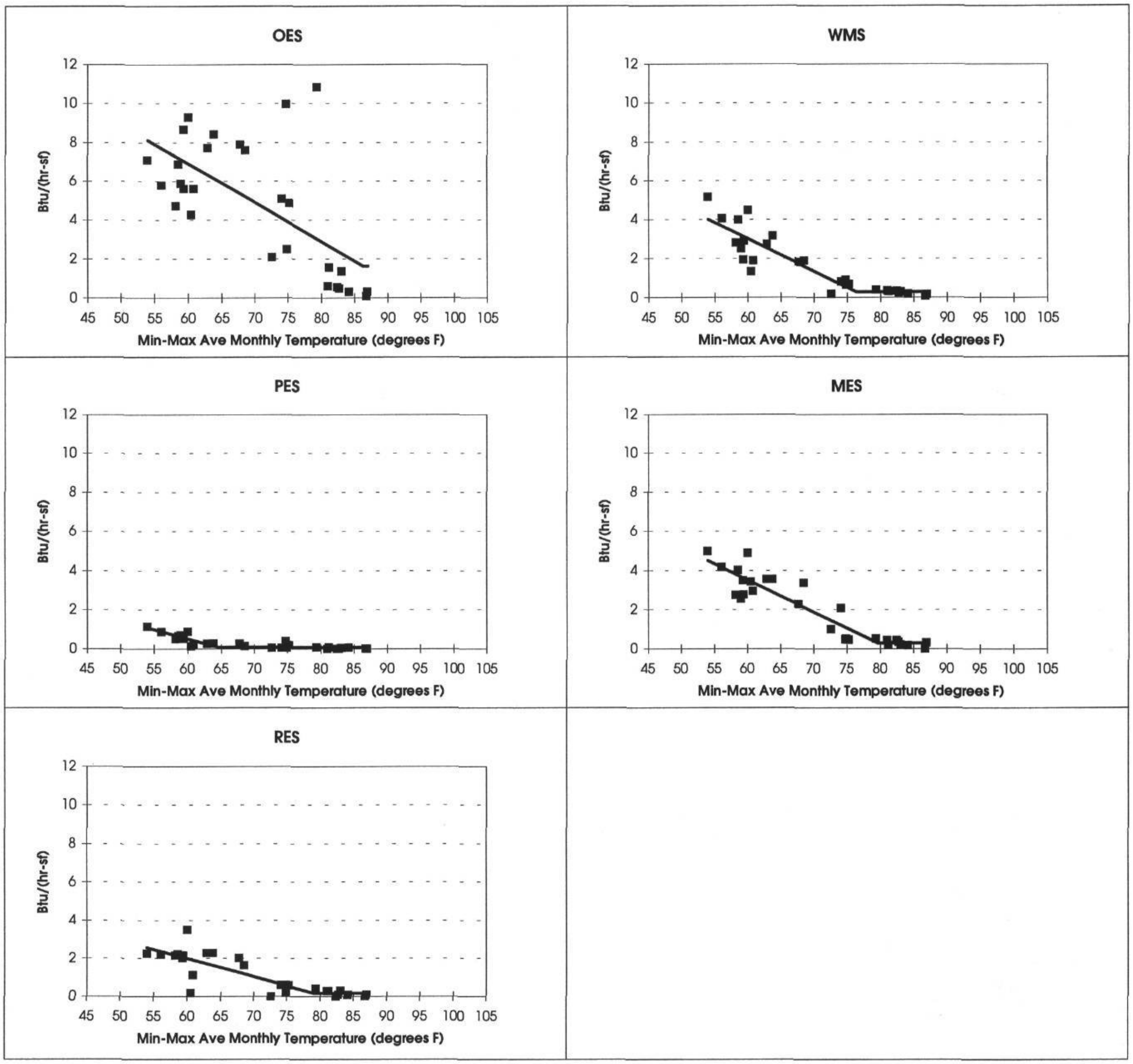
To summarize, the Willow Branch school is an average consumer of electricity and a below average consumer of natural gas when compared to similar schools in Texas. Summertime use seems to indicate that systems are being shut off to conserve energy.

Figure 4a: *Natural Gas Consumption: Btu/(hr-sf) versus min-max average monthly temperatures for September 1991 through December 1993.*



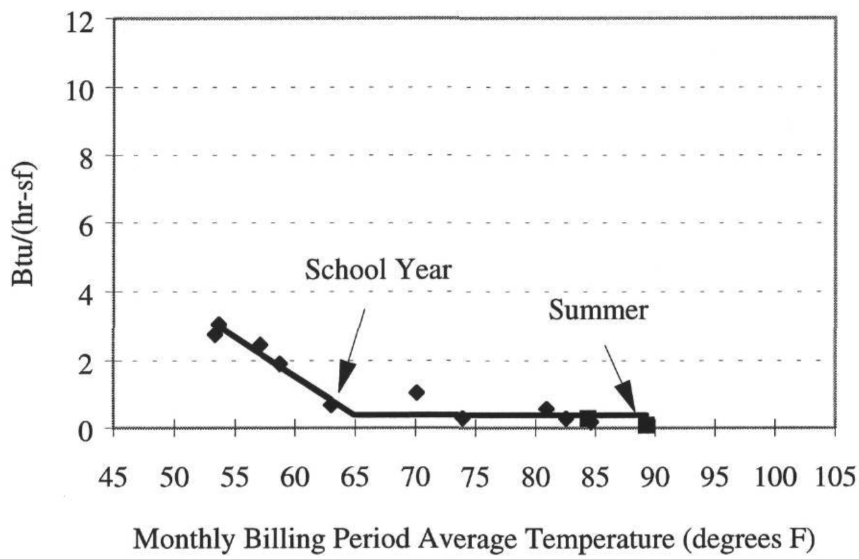
(*Source:* Landman and Haberl 1996)

Figure 4b: *Natural Gas Consumption: Btu/(hr-sf) versus min-max average monthly temperatures for September 1991 through December 1993.*



(*Source:* Landman and Haberl 1996)

Figure 5: *Natural Gas Consumption: Btu/(hr-sf) versus monthly billing period average temperatures for September 1995 through August 1996 for the Willow Branch Intermediate School.*



2.2. Review of the Design of the Willow Branch Intermediate School

The HVAC system for the new building is a water-loop heat pump system. It consists of two supplemental boilers, each with 570,000 Btu/hr input capacity; one cooling tower to reject the heat from the zonal heat pumps with a capacity of 187 tons (2,244,000 Btu/hr). The system also has two 15 hp circulation pumps to circulate water to the 40 zonal heat pump units. Each classroom is accommodated by a zonal heat pump unit. The hallways are not conditioned, except for a small section in front of the principal's office. The HVAC system is remotely controlled by a Solidyne control system from the CSISD administration building. It starts conditioning the school at 7:00 a.m. and is shut down at 5:00 p.m. During the occupied period, the room temperature is maintained at 74°F for the cooling season and 68°F for the heating season. During unoccupied period, the system will turn on whenever a zone temperature rises above 85°F in the summer or falls below 55°F in the winter.

Table 1 is a summary of the as-submitted schedule of the heat pump units. Total cooling capacity is 139 tons (1,668,000 Btu/hr), total heating capacity is 2,460,000 Btu/hr. Column 4 shows the CFM/ton based on the total cooling capacity, which varies from 353 to 432 CFM/ton. In general, the CFM/ton values shown are within the range of 400 CFM/ton that is recommended by ASHRAE for air conditioners and heat pumps. For the total system there is 390 sf/ton, which is also within the ASHRAE recommended range of 250 to 400 sf/ton. This 390 sf/ton value considers the unconditioned hallway areas since the classroom doors are open to the hallway for considerable portions of each day.

Table 1: *Summary of the heat pump units*[1].

Label	No. of Units	CFM	CFM per ton	Cooling Capacity (Btu/hr)		Heating Capacity (Btu/hr)
				Sensible	Total	
HP-A	24	1,400	423	29,532	39,689	59,519
HP-B	4	1,600	411	35,421	46,734	64,653
HP-C	3	2,000	432	42,151	55,590	86,361
HP-D	2	1,400	423	29,532	39,689	59,519
HP-E	1	2,000	421	42,800	57,021	89,893
HP-F	1	630	427	13,440	17,695	26,942
HP-G	1	1,000	353	23,421	34,041	46,109
HP-H	1	1,600	411	35,421	34,041	64,653
HP-I	2	1,600	418	35,076	45,930	63,388
HP-J	1	1000	403	22,037	29,812	44,762
Total	40	58,230			1,662,783	2,464,324

Note: 1. Source: CSISD 1994, Project Record Submittals.

Table 2 is a spreadsheet calculation of the peak cooling demand of the building using the Design Equivalent Temperature Difference method (ASHRAE 1981). It is based on the design dry-bulb and mean coincident wet-bulb temperature of 96/76 °F for Bryan, Texas (ASHRAE 1993). The calculation shows that 132 tons of cooling capacity can meet the peak cooling load, which confirms that the design cooling capacity is adequate. It is interesting to note that during peak cooling loads (55 tons) 42% of the load is latent or cooling that is required to remove excess moisture from the air. Furthermore, 35% (46 tons) of the peak cooling load is used to remove moisture from the outside air that is being brought in to provide fresh air or infiltrates through open doorways, etc.

Also shown in Table 2 is the calculation for the peak heating demand. It is based on the design dry-bulb temperature of 29 °F for Bryan, Texas (ASHRAE 1993). The calculation shows that the building would need 2,346,000 Btu/hr of peak heating capacity to maintain 68°F indoor temperature in the winter, which is very close to the design capacity of 2,464,324 Btu/hr. The unconditioned hallway is included in the calculation.

It should be noted that if the fresh air intakes are blocked, the peak cooling load would reduced to 68 tons, roughly half of the design capacity. Under this condition, the cycling of the heat pumps increases, which lessens their ability to remove humidity because the average supply air temperature rises which reduces the latent heat removal capacity.

Table 2: *Spreadsheet calculations of the design cooling and heating loads.*

							Troom	68
							Tdb	29
	AREA	Wall-area	U-value	DETD	Cooling Load(Btu/hr)	Tons		Heating Load(Btu/hr)
Roof	19000	19000	0.045	36	30,780	2.6		1,365,855
East-Wall	800	600	0.33	16.3	3,227	0.3		7,722
West-Wall	800	600	0.33	16.3	3,227	0.3		7,722
North-Wall	2500	1875	0.33	16.3	10,086	0.8		24,131
South-Wall	2800	2100	0.33	16.3	11,296	0.9		27,027
Northwest-Wall	2600	1950	0.33	16.3	10,489	0.9		25,097
Southeast-Wall	2500	1875	0.33	16.3	10,086	0.8		24,131
Northeast-Wall	800	600	0.33	16.3	3,227	0.3		7,722
Southwest-Wall	1100	825	0.33	16.3	4,438	0.4		10,618
	AREA	Glass-area		DCLF	Cooling			
East-Glass	800	200	1.1	56	11,200	0.9		8,580
West-Glass	800	200	1.1	56	11,200	0.9		8,580
North-Glass	2500	625	1.1	23	14,375	1.2		26,813
South-Glass	2800	700	1.1	31	21,700	1.8		30,030
Northwest-Glass	2600	650	1.1	40	26,000	2.2		27,885
Southeast-Glass	2500	625	1.1	48	30,000	2.5		26,813
Northeast-Glass	800	200	1.1	40	8,000	0.7		8,580
Southwest-Glass	1100	275	1.1	48	13,200	1.1		11,798
						18.5		1,649,102
Internal Load								
	No. of People		Gain/person					
Sensible	546		255		139,230	11.6		
Latent	546		200		109,200	9.1		
	Power(W/SF)	Area(SF)			Btu/hr			
	1.5	35,000			164,850	13.7		
						34.4		
Ventilation & Infiltration								
	CFM		Tdb	Troom	Btu/hr			
Ventilation-sensible	13185		96	74	319,077	26.6		565,637
Infiltrition-sensible	3060		96	74	74,052	6.2		131,274
	CFM		W	Wroom				696,911
Ventilation-latent	13185		0.016	0.009	446,708	37.2		
Infiltrition-latent	3060		0.016	0.009	103,673	8.6		
						78.6		
					Tons-total	**132**		**2,346,013**
					Tons-sensible	77		
					Tons-latent	55		
					Qs/Qt	0.58		

Four fresh air intakes provide the required ventilation air for the building. A fresh air duct is connected to the return duct of each individual heat pump units. Total fresh air supply is 13,185 CFM according to the design drawings. There are also five exhaust fans which operate during the occupied period with a total capacity of 3,060 CFM as shown in Table 3. Thus the maximum outside air supply is 16,245 CFM when all the heat pump units are turned on. This amounts to 30 CFM of fresh air per occupant (if the units ran continuously), which is 50% above the ASHRAE recommendation of 20 CFM per person for schools. However, the fresh air is available only when the heat pump is operational due to the cycling of the blower. An analysis of the measured data indicates that the heat pumps operate approximately 50% of the time, which depends on the cooling or heating load. This would reduce the fresh air to about 9653 CFM or 17.7 CFM/person.

Table 3: *Summary of the exhaust fan capacities.*

Exhaust Fans	EF-1	EF-2	EF-3	EF-4	EF-5	Total
Capacity(CFM)	540	540	520	960	500	3,060

2.3. Problems with the Existing HVAC System

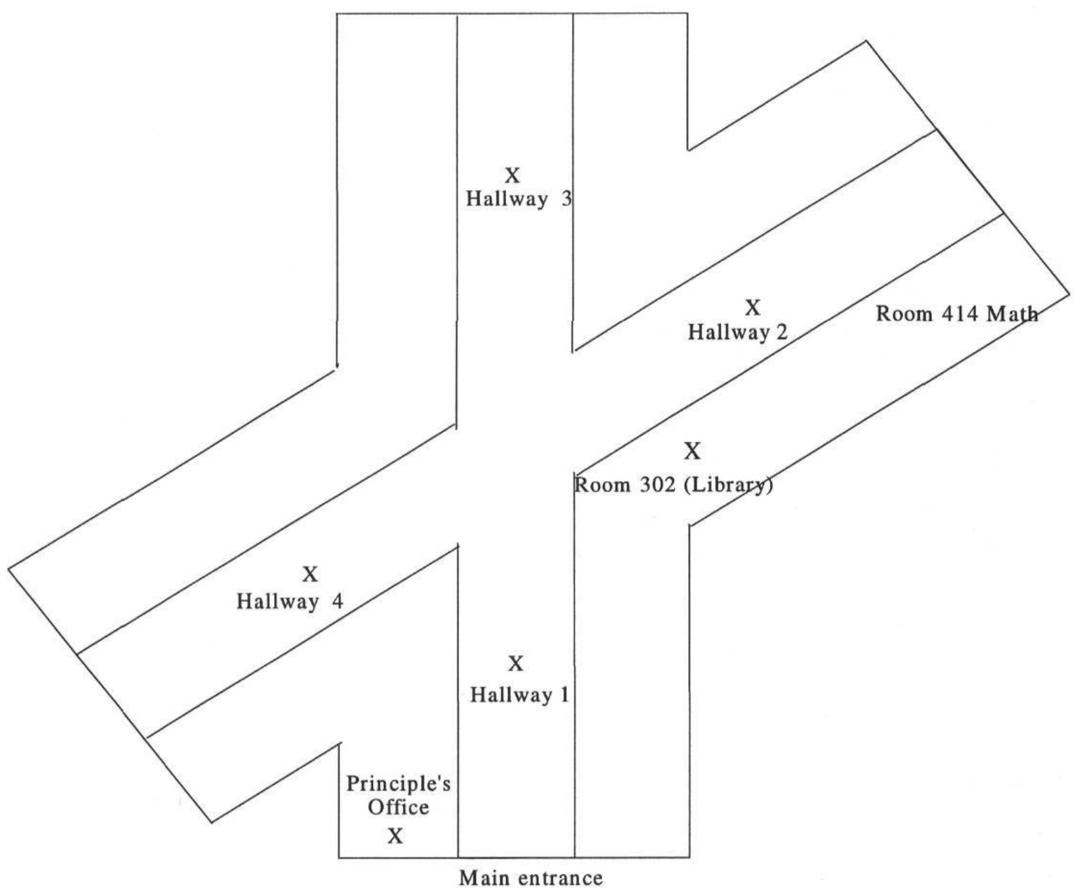
The Willow Branch Intermediate School has experienced a high indoor humidity problem since it was first put into operation in 1994. Indoor relative humidity in excess of 70% was also measured as part of this study. As a temporary solution, the HVAC maintenance personnel decided to reduce the indoor humidity level by sealing off all fresh air intakes. Although this action did reduce the humidity level (according to conversations with the Willow Branch staff), it also reduced the outside air to a level of less than 5 CFM/person which is about 1/4 of what is recommended by ASHRAE (i.e. 20 CFM/person). The 5 CFM/person is currently being provided by infiltration which serves as make-up air for the 3,060 CFM of exhaust air in the bathrooms.

3. Tests and Measurements of the HVAC System

3.1. CO_2 and Room Conditions Measurement

Several tests were carried out to measure the CO_2 levels in various places of the building during different times. Figure 6 shows the location of the sampling sites. A summary of the measurements is presented in Table 4. Figure 7 shows the results of five consecutive days of sampling. All measurements were taken with a hand-held TSI CO_2 meter that has been calibrated at the Energy Systems Laboratory. CO_2 measurements are a reasonable indicator of the freshness of the indoor air when compare against ambient conditions. Very high levels of CO_2 can cause drowsiness and may impede attention levels.

Figure 6: *Floor Plan of Willow Branch Intermediate School.*



("X" is the place where the measurement took place)

The measurements consistently show over 1,500 PPM of CO_2 in the classroom by mid-afternoon which is well in excess of the ASHRAE recommended upper limit of 1,000 PPM for CO_2. The coincident outside measurements were in the 370 to 400 PPM range during this period as indicated in the graph. The sample numbers in the graph refer to the measurements as follows:

SAMPLE 1	SAMPLE 2	SAMPLE 3	SAMPLE 4	SAMPLE 5	SAMPLE 6
Monday	Tuesday	Tuesday	Wednesday	Wednesday	Thursday
10/21/96	10/22/96	10/22/96	10/23/96	10/24/96	10/24/96
2:00 p.m.	7:00 a.m.	2:00 p.m.	7:00 a.m.	2:00 p.m.	7:00 a.m.

Samples were taken in the following locations:

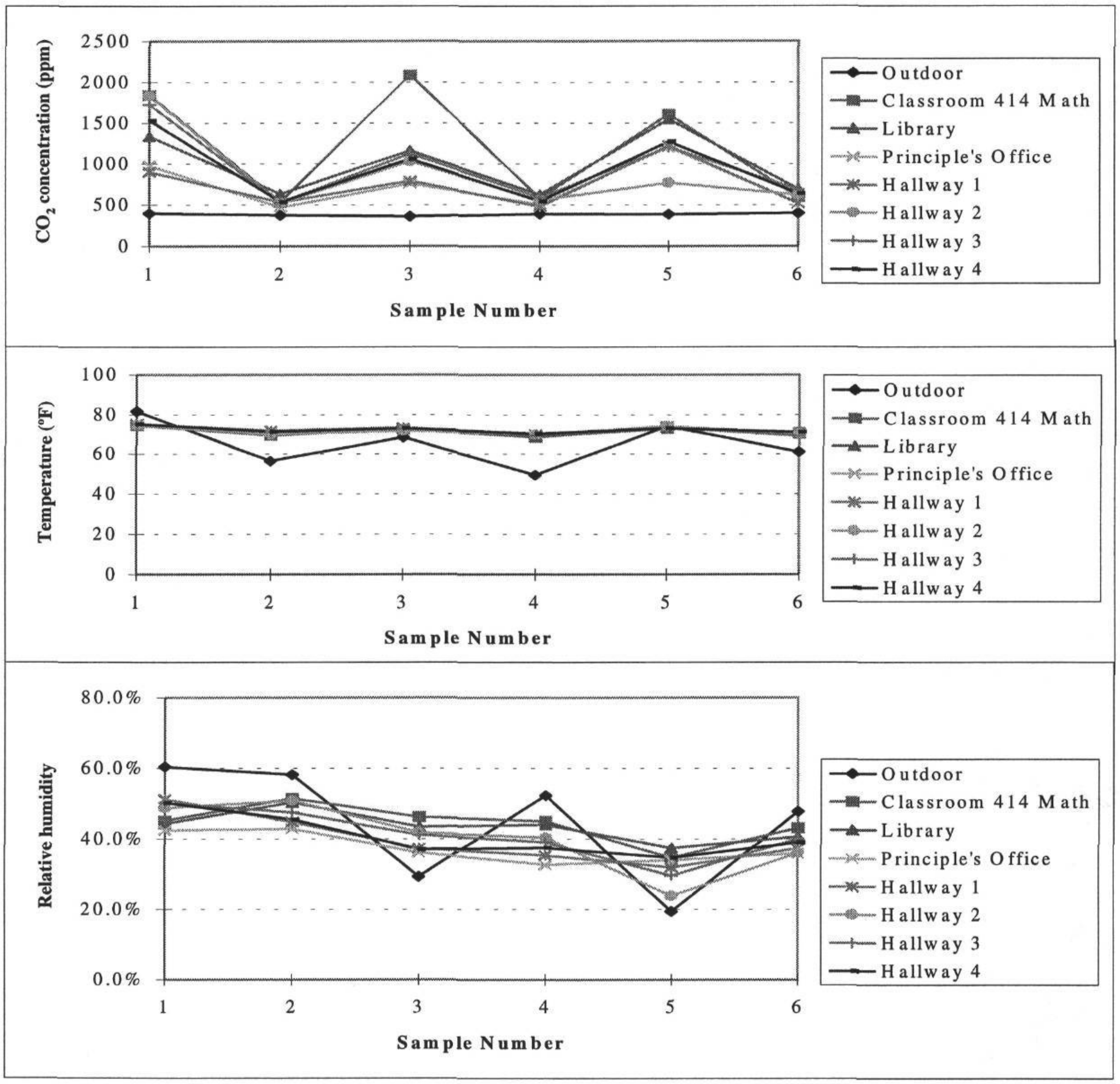
- Outdoors
- Classroom 414 Math
- Library
- Hallway 1 -- the main entrance to the school in the middle of the hallway
- Hallway 2 -- the hallway to the right of hallway 1 in the middle of the hallway
- Hallway 3 -- the hallway to the right of hallway 2 in the middle of the hallway
- Hallway 4 -- the hallway to the right of hallway 3 in the middle of the hallway

Table 4: *CO_2 measurements.*

	Sample number	1	2	3	4	5	6
CO_2 (PPM)	Outdoor	393	372	358	385	383	395
	Classroom 414 Math	1842	535	2090	580	1604	595
	Library	1334	630	1160	622	1542	693
	Principle's Office	974	470	760	503	1209	514
	Hallway 1	900	530	790	480	1223	519
	Hallway 2	1832	520	1033	560	769	621
	Hallway 3	1723	524	1129	591	1208	686
	Hallway 4	1528	534	1060	543	1271	637
Temperature (°F)	Outdoor	82	56.3	68.4	49.3	74	61.2
	Classroom 414 Math	74	69.5	72.5	68.5	73.5	70.8
	Library	76	70.8	72.5	68.9	72.8	70.2
	Principle's Office	75	71.9	73.3	70	73.8	70.6
	Hallway 1	76	71.7	73.3	69.8	74	70.5
	Hallway 2	75	69.8	71.8	69.6	74.2	70.2
	Hallway 3	75	71.1	72.7	69.5	73.4	69.1
	Hallway 4	75	71.4	73.1	70.3	73.2	71

	Outdoor	60.4%	58.1%	29.4%	52.2%	19.4%	47.6%
	Classroom 414 Math	45.0%	51.3%	46.0%	44.9%	34.6%	43.0%
	Library	44.2%	50.2%	43.3%	43.9%	37.4%	40.5%
Relative Humidity	Principle's Office	42.3%	42.8%	36.1%	32.7%	33.9%	35.8%
	Hallway 1	51.1%	44.6%	37.2%	35.4%	32.0%	37.2%
	Hallway 2	48.6%	50.6%	41.7%	40.3%	23.8%	36.0%
	Hallway 3	49.9%	47.3%	41.0%	39.0%	29.6%	40.0%
	Hallway 4	50.1%	45.4%	37.0%	37.5%	34.6%	38.9%

Figure 7: *CO_2 measurements.*



In general, it can be seen that high CO_2 levels seem to be worse in the classrooms and library in the afternoons after they have been occupied. The hallways were noticeably better since they can exchange air directly with the outside every time an exterior door was open. The principle's office seems to fluctuate at levels between the hallways and the classrooms.

It is also interesting to note that the CO_2 levels fall to 500 to 700 PPM in the evenings as evidenced by the 7:00 A.M. readings. This would seem to indicate that even though the fresh air vents in the school are closed, there is still significant infiltration and/or ventilation which is why the levels are dropping to the low levels that were measured. This may also be indicating that one or more exhaust fans remain on 24 hours per day.

Since high CO_2 concentrations can cause sleepiness, headaches and other CO_2 related symptoms. Therefore, it is important to correct this particular problem by reconnecting the fresh air vents. During the winter, it is not anticipated that this will cause excess indoor humidity problems.

3.2. Temperature and Relative Humidity Measurements

In order to further study the high humidity problem, one month of temperature and humidity data were recorded by portable data loggers. The data from the loggers were plotted together with ambient temperature and humidity data in Figure 8 . The ambient temperature and humidity data is recorded as part of the LoanSTAR program (Turner 1990). In Figure 8a, the upper plot shows the temperature and relative humidity in room 414 Math, the middle plot shows the supply air temperature and relative humidity from the zonal heat pump, the bottom plot shows the ambient conditions. In Figures 8b and 8c an additional plot is provided that shows the hallway temperature and relative humidity from a locker that was fitted with a pair of temperature and humidity sensor and a fan to circulate the air from the hallway. The Willow Branch data are 5-minute data and the LoanSTAR data are hourly ambient data recorded on top of the Zachry Engineering Center at Texas A&M University main campus.

The features worth noting are as follows. First, the room seems to be well maintained at 74 °F during the occupied periods. During unoccupied periods the room temperature actually rise above the ambient temperature since the HVAC system is shutoff and the heat trapped in the building's mass slowly conducts into the space. In Figure 8b, the hallway temperature is less controlled and floats at a point that is somewhere between the room temperature and the ambient temperature. The humidity level in the hallway is similar to the room humidity, however, it follows a general trend that is set by outside conditions.

Figure 8a: *Room 414-math, Supply and Ambient conditions. (10/21 - 10/29)*

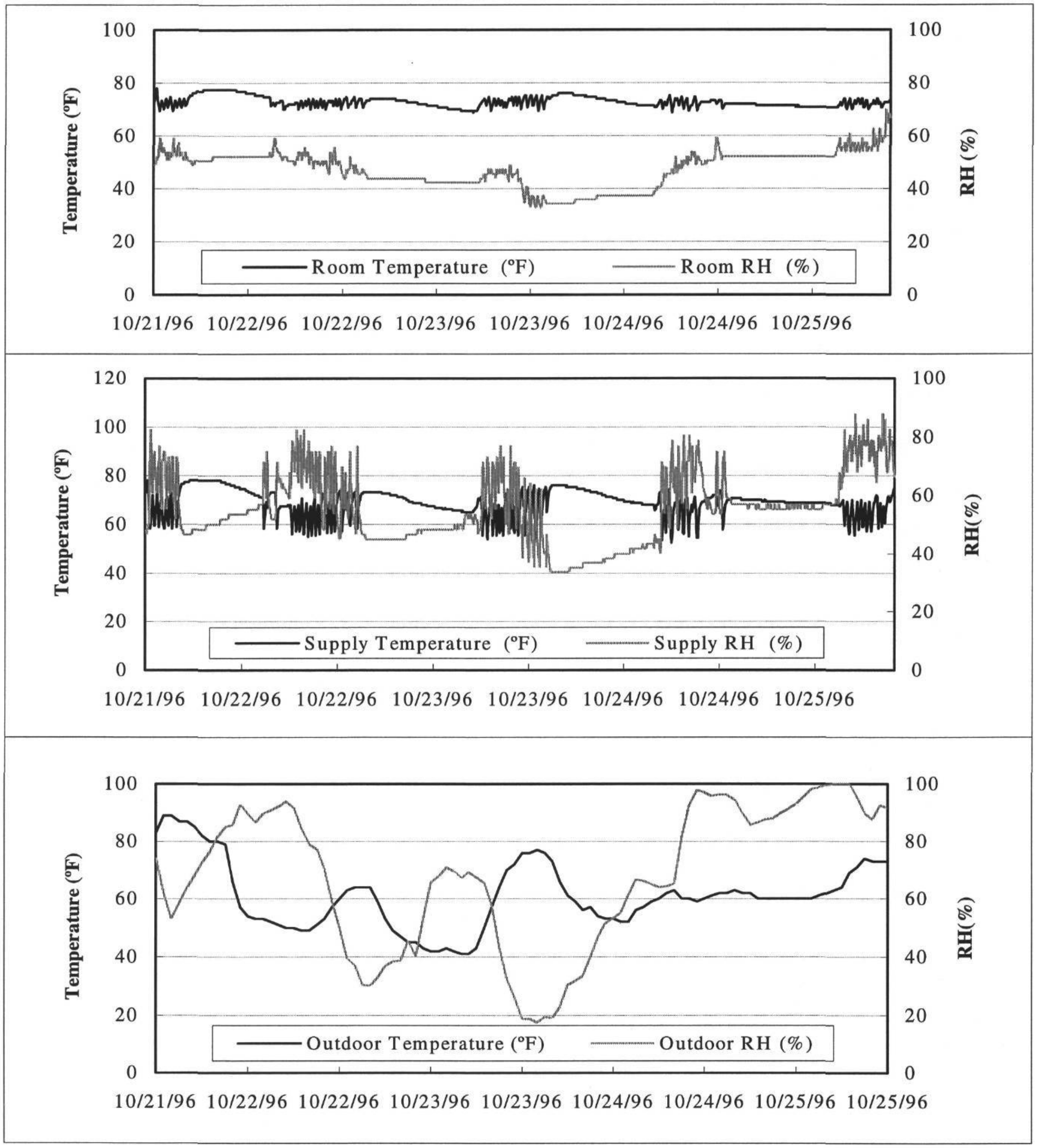


Figure 8b: *Room 414-math, Supply, Hallway and Ambient conditions. (10/25 - 10/31)*

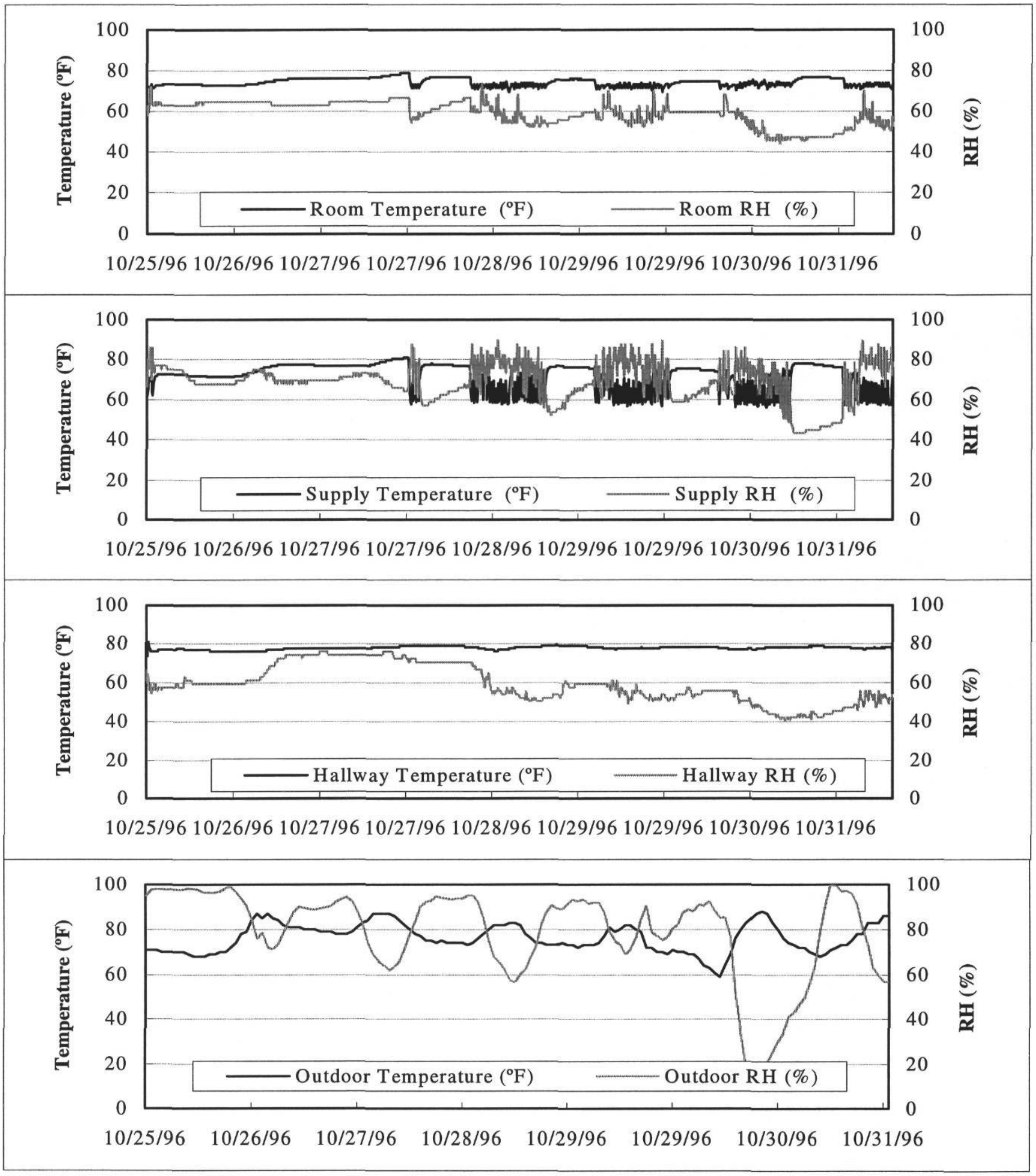
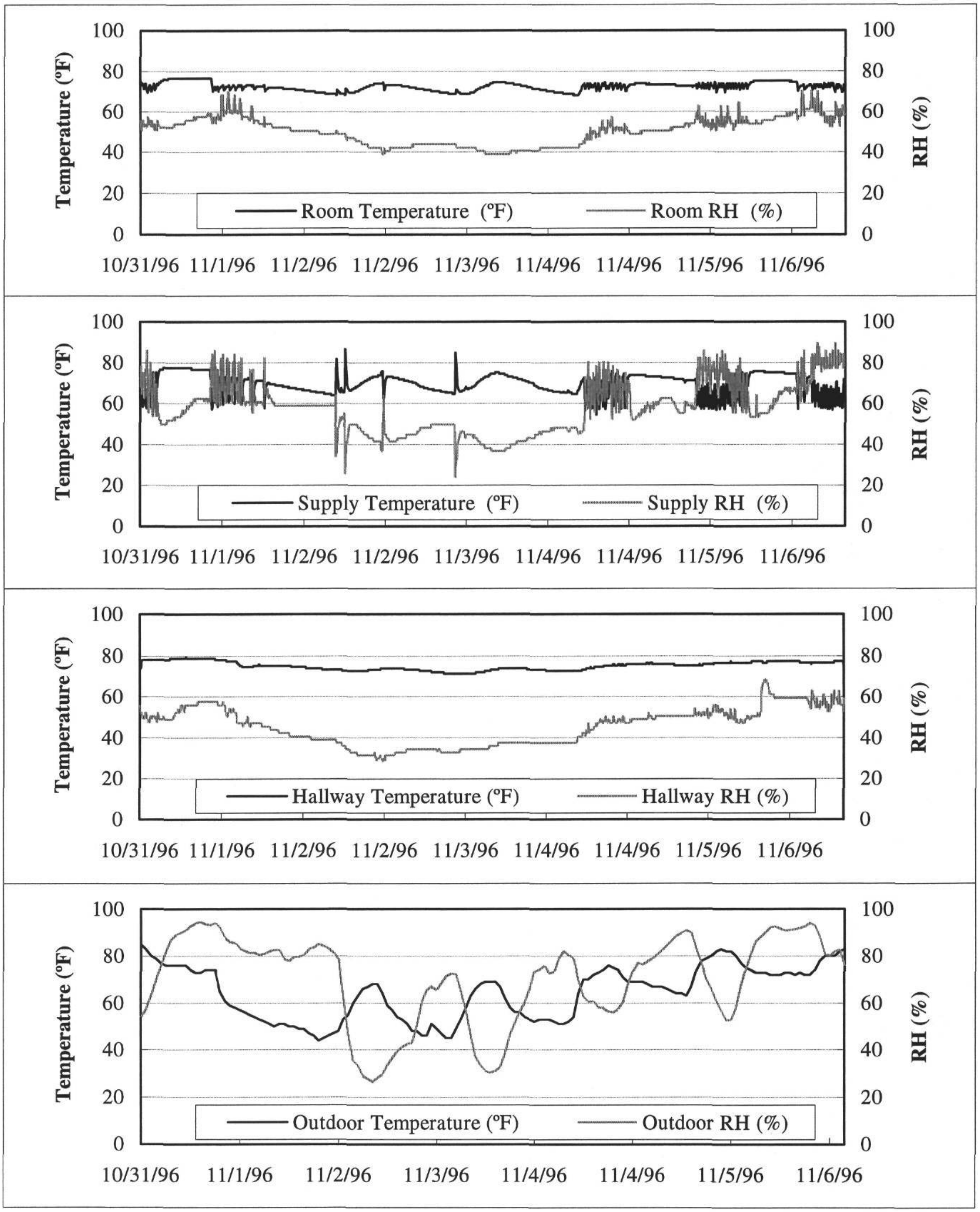


Figure 8c: *Room 414-math, Supply, Hallway and Ambient conditions. (10/31 - 11/6)*



Second, it can be seen that during several periods the humidity in the rooms swelled to 70% which indicates that the current system is just barely maintaining control of the humidity in the rooms when the rooms are loaded with students (Note: This is also with the outside air vents shut). The high humidity is caused by the cycling of the heat pumps which does not allow the average supply air temperature to become cold enough to drive out the moisture.

Third, the effect of the exhaust fans (or excessive infiltration) is accelerating the rise and fall of humidity during the weekends when the building is not occupied. During Saturday 10/26/96 and Sunday 10/27/96 the humidity level over the weekend actually rose in the hallway and slightly in the classroom when the building was unoccupied. This is most likely indicating that either one or more exhaust fans were drawing in moist air from the outside, or that the building's exterior shell is very leaky, or both.

During Saturday 11/2/96 and Sunday 11/3/96 the humidity levels dropped in the hallway corresponding to dry ambient conditions. Both of these conditions support the hypothesis that the exhaust fans are drawing in outside air when the building is unoccupied which can humidify or dehumidify depending upon the difference in moisture levels between the indoor conditions and the ambient conditions. This hypothesis was confirmed during a site visit on December 6, 1996 (Saturday), when it was noticed that one of the exhaust fans was running even though conversations with Mr. Marinik indicated that the Solidyne control system was scheduled to shut-off the fan at 6:00 p.m.

A fourth feature worth mentioning is the fact that the supply temperature remained in the 55-60 ºF range when the heat pump was on in the cooling mode. As indicated in the psychrometric chart in the appendix, the supply air needs to be colder than 55 ºF to begin to dehumidify the moist room air. Therefore, the supply air from the heat pump is removing little, if any, of the moisture in the room air.

To solve the high humidity problem, two options are available. Either install a fresh air pre-conditioner or reduce the cycling of the heat pumps, or both. Installation of a preconditioning unit is the recommended long term solution but it is also quite expensive. A lower cost alternative may be available to reduce the air flow rate across the heat pump units. This will lower the supply air temperature during the cooling season and raise the supply air temperature during the heating season.

Unfortunately, reducing the air flow will reduce the SEER of the units (i.e. reduce the efficiency), and the peak cooling and heating capacity. One must also be careful not to induce icing on the coils (an effect that occurs at 25 - 50% of normal air flow) by reducing air flow too low.

This concept was tested by simply taping the filters to reduce the air flow to the unit. Air flow rates before and after taping were measured using a calibrated air flow meter. The results are shown in Table 5. By taping 75% of the face area of the filter, air flow rate was reduced by 20%. This allowed for an adequate test at the reduced air flow .

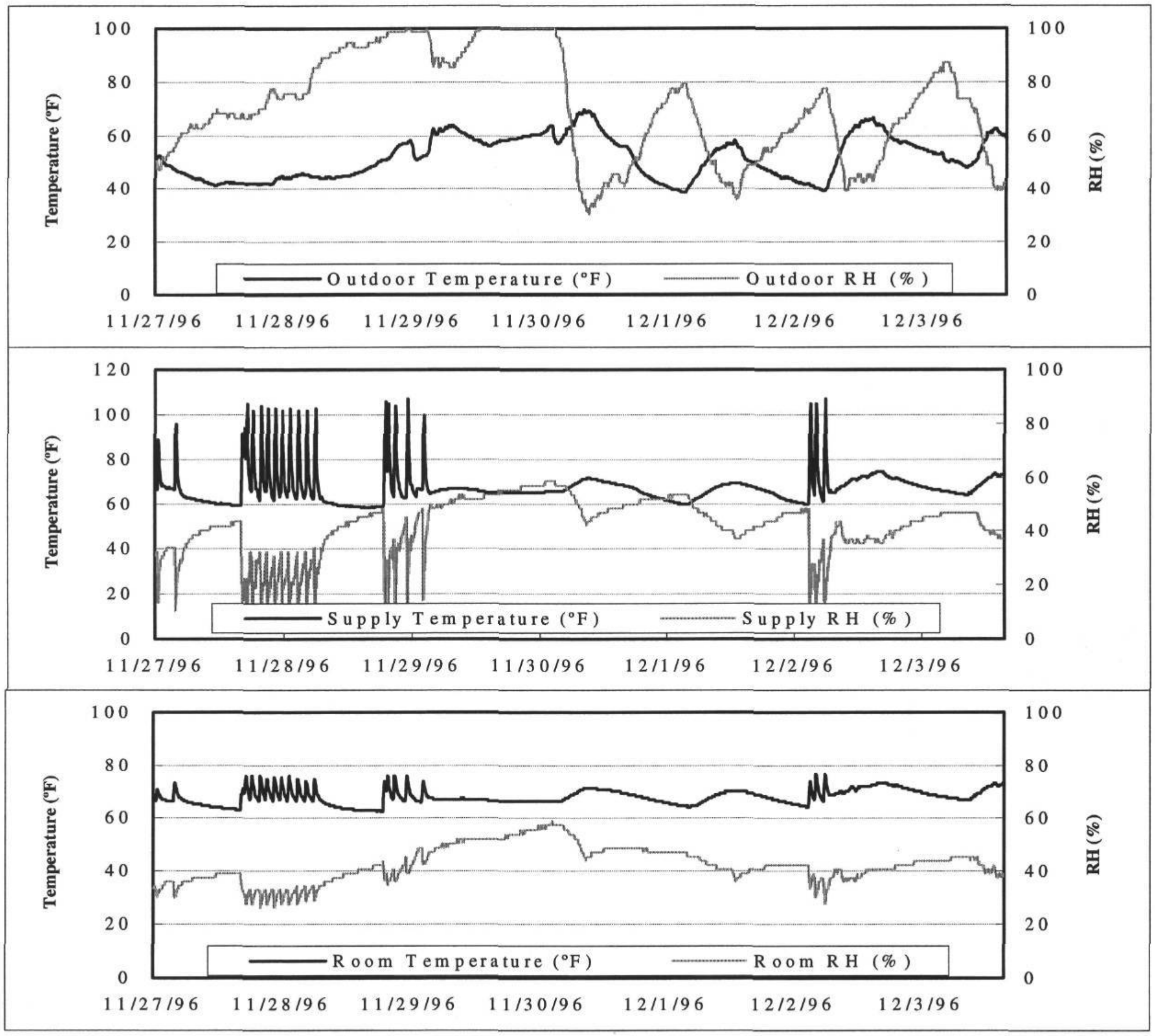
Table 5: *Air flow measurement.*

	Diffuser #1	Diffuser #2	Diffuser #3	Diffuser #4	Total
Normal Flow (CFM)	280	290	270	330	1,200
Reduced Flow (CFM)	230	240	230	270	970
Reduction (%)					20%

One week of supply air and room temperature and humidity were recorded by the loggers after the taping. This is plotted in Figure 9. Unfortunately, the cooling season was over and we were not able to see the intended effect of reducing the air flow on the average supply air temperature. However, we did notice that the heat pump was able to meet the heating load under reduced air flow since room temperature were easily maintained at above 70 °F.

While reviewing the operation manual of the heat pump units, we also noticed that the fan blower inside the heat pump unit has two speeds. All the units were set to the high speed when shipped from the factory. To switch the fan blower to low speed, the wiring of each unit has to be modified. Therefore, it may also be possible to modify the HVAC controls to automatically switch from high to low depending upon indoor humidity levels. We would like to recommend that this option be further investigated.

Figure 9: *Room, Supply and Ambient Conditions under heating mode. (11/27 - 12/3)*



4. Conclusions and Recommendations

The Willow Branch HVAC system seems to be properly designed for temperature control, however, it needs additional equipment and/or controls for proper humidity control. Our suggestions for the HVAC system are as follows:

1) Open the fresh air vents until the high humidity returns next Spring;
2) Install fresh air pre-conditioner or investigate the use of two-speed fans;
3) Shut off exhaust fans during unoccupied periods;
4) Treat the cooling tower water and reduce the rate of over flow that is currently used (i.e. 250,000 gal/mo).

Finally, we would like to recommend that monitoring and/or testing continue in the Spring. This can include setting the Math 414 unit to low speed with the fresh air vent reconnected and possibly measuring another room at high speed that also has the fresh air vent reconnected. We would also like to suggest that it may be possible to reduce electricity use by installing a variable speed controller for the two water-loop pumps. A quick calculation indicates that if the electricity use of the pumps could be cut in half, this would amount to $2,175 per year in electricity savings. Estimated cost of such controllers would be about $2,500 for each motor.

References

1. ASHRAE Handbook of Fundamentals, 1981. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.
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3. Landman, D., Haberl, J., 1996. "A Study of Diagnostic Pre-screening Methods for Analyzing Energy Use of Public Schools". Energy Systems Laboratory, Report number ESL-TR-96/11-02. Sponsored by the USDOE. through Oak Ridge National Lab.
4. Kissock, J. K., 1993. "A Methodology to Measure Retrofit Energy Savings in Commercial Buildings". Doctoral Dissertation, Department of Mechanical Engineering, ESL-TH-93/12-03, Texas A&M University.
5. Turner, W. D. 1990. "Overview of the Texas LoanSTAR Monitoring Program", 7th Annual Symposium on Improving Building Systems in Hot and Humid Climates 1990, October 9-10, 1990, Fort Worth, Texas, pp. 28-34. ESL-PA-90/10-01.

Appendix A

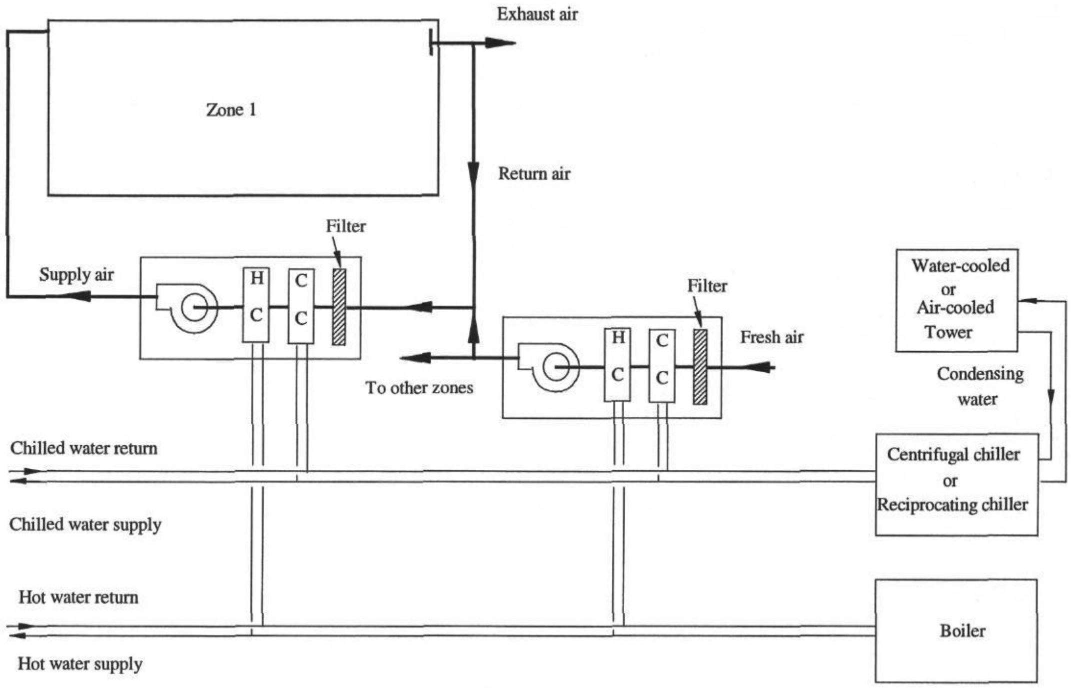
Typical HVAC System for Texas Primary and Secondary Schools

In this appendix, five typical HVAC systems suitable for Texas primary and secondary schools are presented, along with a schematic drawing and a description of the pros and cons of each system.

Five typical HVAC systems for Texas primary and secondary schools are presented in the following pages. They are:

1) Four-pipe, single zone air handling unit;
2) Package split system with individual heat pump;
3) Two-pipe, water-loop heat pump with a packaged split system for pre-conditioning fresh air and ground-coupled water-loop heat pump with a packaged split system for pre-conditioning fresh air;
4) Four-pipe Texas multizone system with cold deck bypass;
5) Four-pipe fan coil units with fresh air pre-conditioning. The schematic of each system is shown in Figures A1 to A6, along with a summary of their respective advantages and disadvantages. A fresh air pre-conditioning unit is included for each type of system to remove moisture from the fresh air in the summer and preheat the fresh air in the winter.

Figure A1: *Four-pipe, single zone air handling unit.*



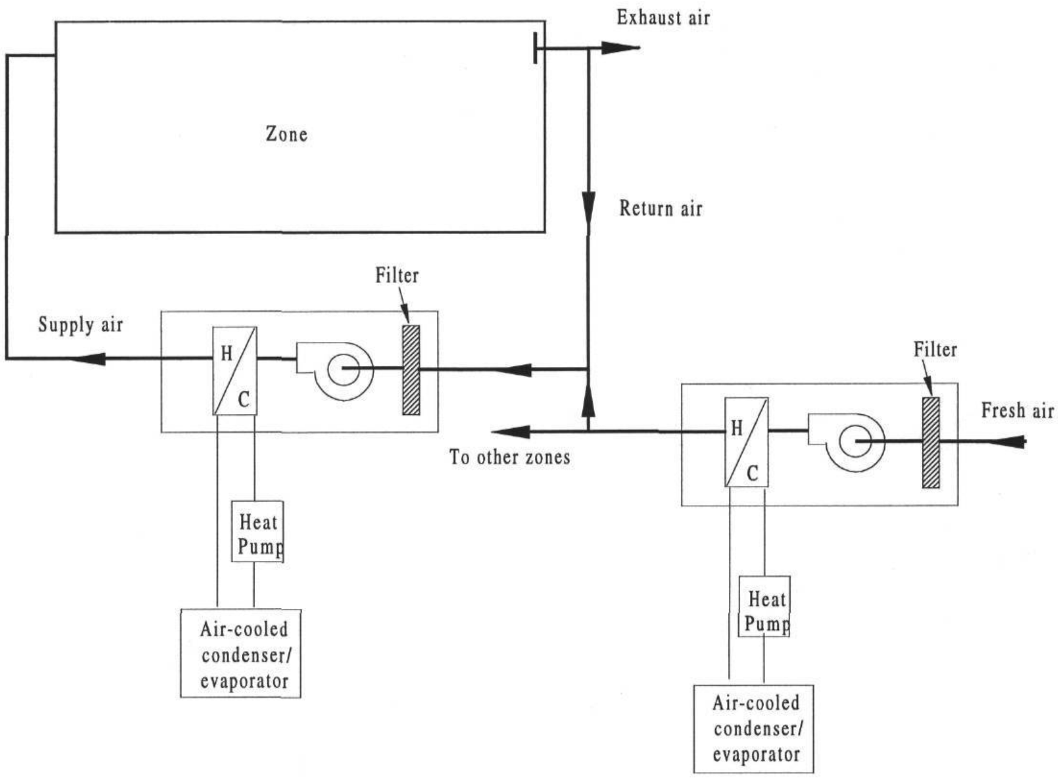
Advantages:

- Can provide heating and cooling as needed.
- Good humidity control.
- Boiler and chiller are installed at central location.
- May operate with or without air distribution ductwork(minimum ductwork).
- Zonal air handling unit can be shut down without affecting adjacent areas.
- Chiller efficiencies are higher than that of individual heat pump.
- Long equipment life (i.e. 25 years).

Disadvantages:

- Initial cost of four-pipe system is slightly more expensive than two-pipe system.
- Hot water and chiller water-loop must run when only one zone needs heating or cooling.
- Entire system is shut down when either loop fails.
- Chiller and boiler need servicing by special repairman.
- Water-cooled condenser tower needs frequent service and water quality check.
- Air-cooled condenser temperature is higher, therefore, chiller efficiency is lower than water-cooled condenser.
- Central system may have more energy use due to operation of four-pipe loop.

Figure A2: *Packaged split system with individual heat pump.*



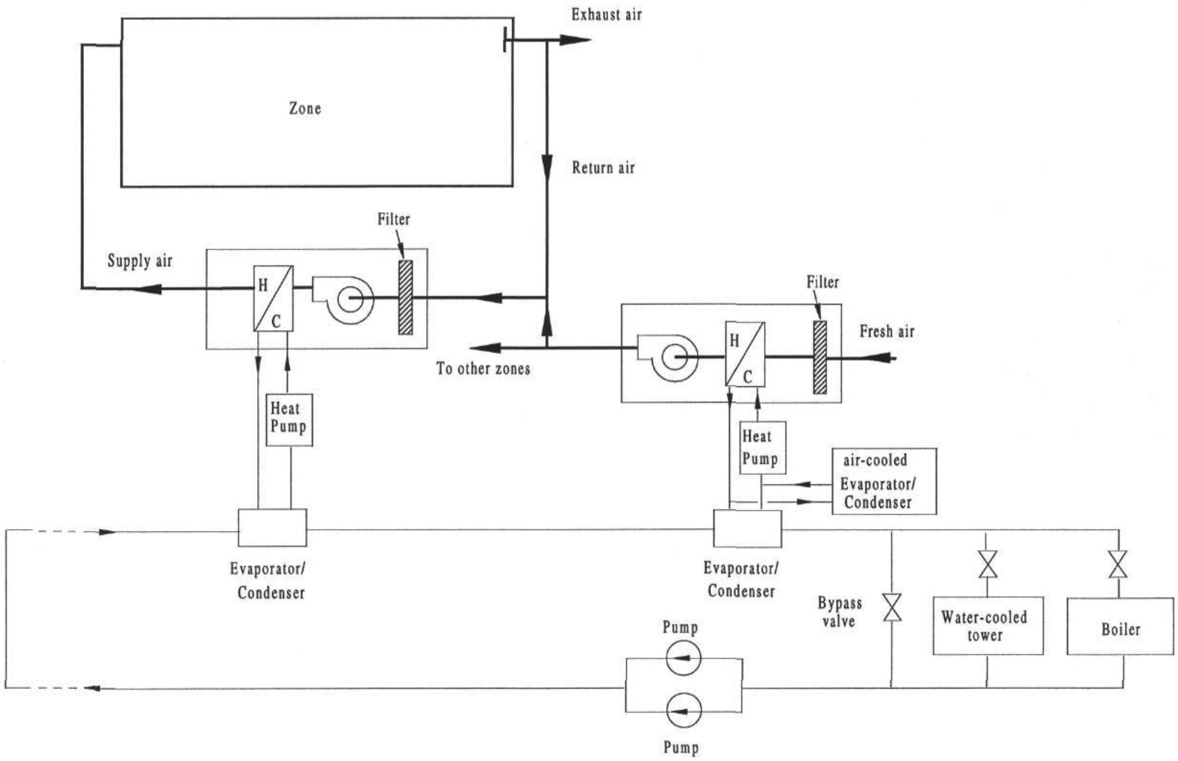
Advantages:

- Can provide heating and cooling as needed.
- Installation is simple.
- No mechanical room is required.
- Ductwork only required for fresh air.
- Initial cost is low.
- Well suited to spaces requiring many zones with individual temperature control.
- System operation and maintenance are simple and can possibly be maintained by local repair person.

Disadvantages:

- Combined noise level of individual condensers is generally high.
- Maintenance costs are high.
- Overall efficiencies of individual heat pump less than that of one large chiller.
- Needs electric resistance heating when outside air temperature is below 35 °F.
- Humidity control can be problematic if there is no pre-conditioning unit and zonal heat pumps are oversized.
- Wall penetration required for refrigerant lines to/from condenser/evaporator.
- Overall appearance of individual condensers can be unappealing.
- Equipment life may be relatively short (typically 10 years).

Figure A3*: Two-pipe, water-loop heat pump with packaged split system.*



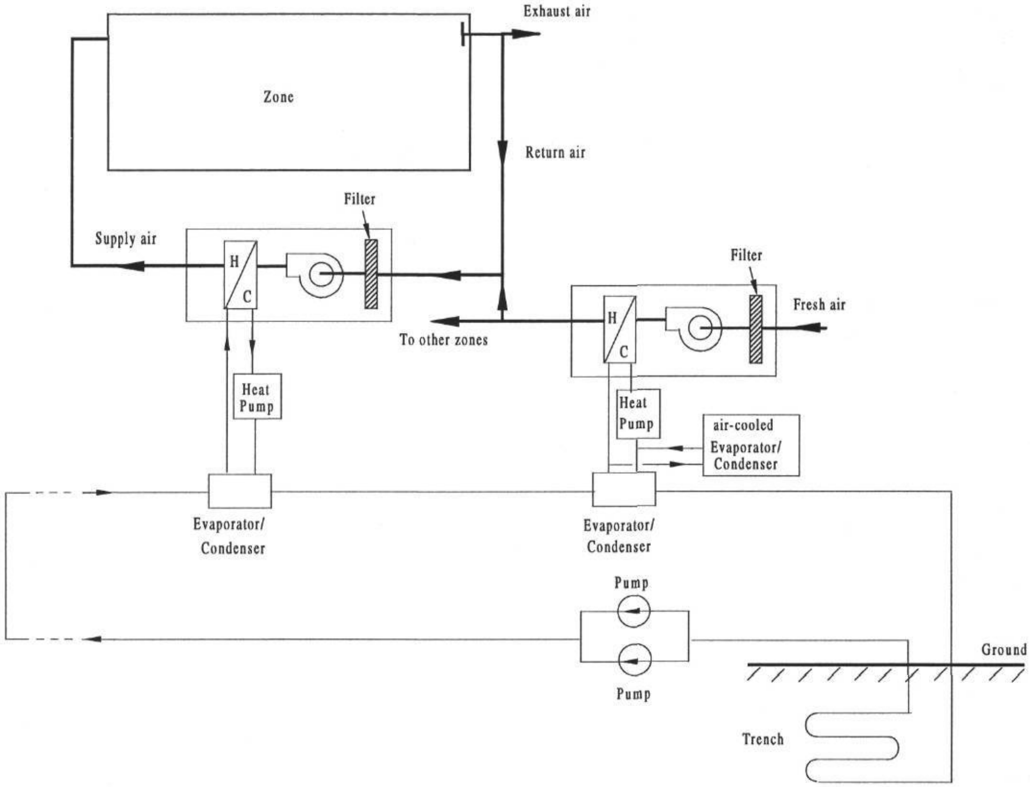
Advantages:

- Provides heating and cooling as needed.
- Allows recovering heat from interior zones and/or waste heat.
- Packaged split-system for pre-conditioning allows for fresh air to be supplied without running main loop.
- Can be maintained locally, no special chiller repairman is needed for heat pumps.
- Units have longer service life than air-cooled heat pumps.
- The entire system is not shut down when a zonal unit fails.
- Total life-cycle cost of this system compares favorably to that of central systems when considering installed cost, operating costs, and system life.

Disadvantages:

- Initial cost may be higher than for systems that use multiple unitary HVAC equipment.
- Special repairman needed for boiler.
- Water-cooled tower needs frequent service and water quality check.
- Cleanliness of the piping loop must be maintained.
- Loop needs to run 24 hours/day when any zone needs cooling or heating.
- Entire system is shut down when loop fails.
- More maintenance will be required since heat pumps together with air handling units are decentralized.

Figure A4: *Two-pipe, ground-coupled water-loop heat pump with packaged split system.*



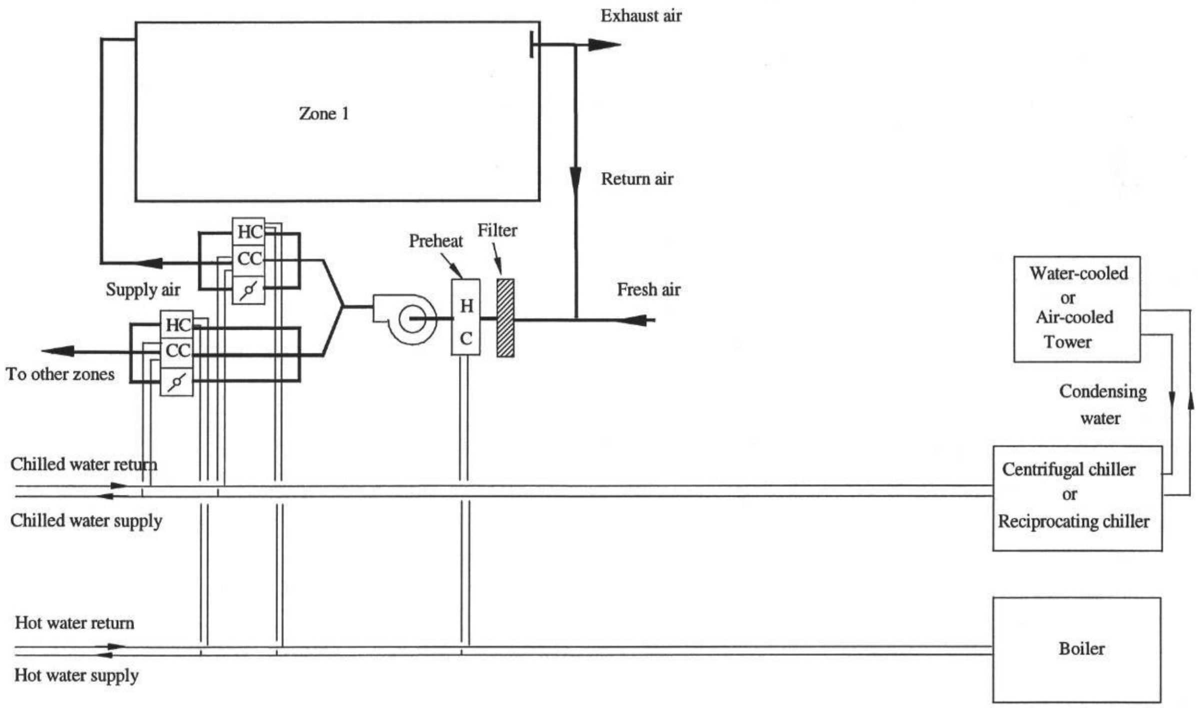
Advantages:

- Can provide heating and cooling as needed.
- Allows recovering heat from interior zones and/or waste heat.
- Can be maintained locally, no special chiller repairman is needed for heat pump.
- Units have longer service life than air-cooled heat pumps.
- The entire system is not shut down when a zonal unit fails.
- Total life-cycle cost of this system compares favorably to that of central systems when considering installed cost, operating costs, and system life.
- Does not normally need a boiler and/or cooling tower.

Disadvantages:

- Initial cost may be higher than for systems that use individual unitary HVAC equipment.
- Cleanliness of the piping loop must be maintained.
- Loop needs to run 24 hours/day when any zone needs cooling and heating.
- Entire system is shut down when loop fails.
- More maintenance will be required since heat pumps together with air handling units are decentralized.
- Soil type, moisture content, composition, density, and uniformity affect the success of this method of heat exchange.
- The material of construction for the pipe and the corrosiveness of the local soil and ground water may affect the heat transfer and service life.
- Large area needed to drill wells.

Figure A5: *Four-pipe Texas multizone system with cold-deck bypass.*



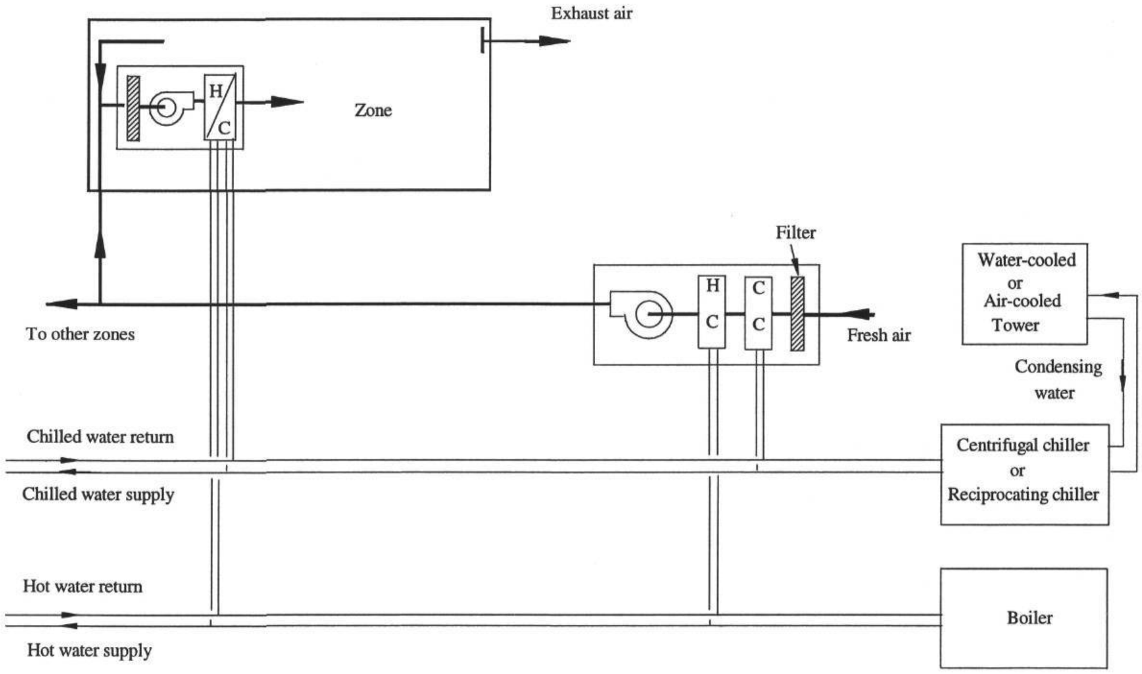
Advantages:

- Can provide heating and cooling as needed.
- Supplies several zones from centrally located air-handling units.
- No pipes required that could leak in occupied areas.
- Chiller efficiency is higher than that of individual heat pump.
- Good humidity control.
- Chiller and boiler typically have longer life than individual units.

Disadvantages:

- Hot water and chilled water-loop must run when only one zone needs heating or cooling.
- Chiller and boiler need servicing by special repairman.
- Water-cooled condenser tower needs frequent service and water quality check.
- Required additional space for distribution ductwork.
- Air-cooled condenser temperature is higher, therefore, chiller efficiency is less than water-cooled condenser.
- Central system may have more energy use due to the running of two loops.

Figure A6: *Four-pipe fan coil units.*



Advantages:

- Provides heating and cooling as needed.
- Zonal fan energy is minimized.
- Minimal air distribution ductwork is needed.
- Chiller efficiency is higher than that of individual heat pumps.
- Zonal fan coil unit can be shutdown without affecting adjacent areas.
- No summer/winter changeover requirement.
- Simpler operation.

Disadvantages:

- Initial cost of four-pipe system is slightly more expensive than two-pipe system.
- Hot water and chilled water-loop must run when only one zone needs heating or cooling.
- Chiller and boiler need servicing by special repairman.
- Noise of individual fan coil units may be a problem.
- Water-cooled condenser tower needs frequent service and water quality check.
- Air-cooled condenser temperature is higher, therefore, chiller efficiency is less than water-cooled condenser.
- Central system may have more energy use due to the running of two loops.
- Decentralized maintenance of zonal units can cause additional maintenance time.

Appendix B

Supporting Material for Table 2

The heat gain /loss in Table 2 is calculated as follows (source: ASHRAE 1981):

Gains through walls: $q = U \times A_{walls} \times DETD$

Gains through roof : $q = U \times A_{roof} \times DETD$

Gains through glass: $q = A_{glass} \times DCLF$

Gains from outdoor air: $q_{sensible} = Q \times (T_{design} - T_{room}) \times 1.08$

Gains from outdoor air: $q_{latent} = Q \times (T_{design} - T_{room}) \times 4840$

Gains from people: q = Number of occupants × Gain per person

Gains from lights: $q = 1.5 \times A_{floor}$

where

DETD = Design equivalent temperature difference, see Table B1

DCLF = Design cooling load factors, see Table B2

A = Area

Q = Ventilation air flow rate

T_{design} = Design temperature

T_{room} = Room temperature

Table B1: *Design Equivalent Temperature Difference.*

29.4°C		32.2			35.0			37.7		40.5	43.3	Outdoor Design Temperature	85 F		90			95			100		105	110
L	M	L	M	H	L	M	H	M	H	H	H	Daily Temperature Range[a]	L	M	L	M	H	L	M	H	M	H	H	H
												Walls and Doors												
9.7	7.5	12.5	10.3	7.5	15.3	13.1	10.3	15.8	13.1	15.8	18.6	1. Frame and veneer-on-frame	17.6	13.6	22.6	18.6	13.6	27.6	23.6	18.6	28.6	23.6	28.6	33.6
5.7	3.5	8.5	6.3	3.5	11.3	9.1	6.3	11.8	9.1	11.8	14.6	2. Masonry walls, 203.2-mm (8-in.) block or brick	10.3	6.3	15.3	11.3	6.3	20.3	[illegible]	11.3	21.3	16.3	21.3	26.3
5.0	2.7	7.7	5.5	2.7	10.5	8.3	5.5	11.1	8.3	11.1	13.8	3. Partitions, frame	9.0	5.0	14.0	10.0	5.0	19.0	15.0	10.0	20.0	15.0	20.0	25.0
1.4	0.0	4.2	1.9	0.0	6.9	4.7	1.9	7.5	4.7	7.5	10.3	masonry	2.5	0	7.5	3.5	0	12.5	8.5	3.5	13.5	8.5	13.5	18.5
7.7	7.5	12.5	10.3	7.5	15.3	13.1	10.3	15.8	13.1	15.8	18.6	4. Wood doors	17.6	13.6	22.6	18.6	13.6	27.6	23.6	18.6	28.6	23.6	28.6	33.6
												Ceilings and Roofs[c]												
21.1	18.8	23.8	21.6	18.8	26.6	24.4	21.6	27.2	24.4	27.2	30.0	1. Ceilings under naturally vented attic or vented flat roof—dark	38.0	34.0	43.0	39.0	34.0	48.0	44.0	39.0	49.0	44.0	49.0	54.0
16.6	14.9	19.4	17.2	14.4	22.2	20.0	17.2	22.7	20.0	22.7	25.5	—light	30.0	26.0	35.0	31.0	26.0	40.0	36.0	31.0	41.0	36.0	41.0	46.0
21.1	18.8	23.3	21.6	18.8	26.6	24.4	21.6	27.2	24.4	27.2	30.0	2. Built-up roof, no ceiling—dark	38.0	34.0	43.0	39.0	34.0	48.0	44.0	39.0	49.0	44.0	49.0	54.0
16.6	14.4	19.4	17.2	14.4	22.2	20.0	17.2	22.7	20.0	22.7	25.5	—light	30.0	26.0	35.0	31.0	26.0	40.0	36.0	31.0	41.0	36.0	41.0	46.0
5.0	2.7	7.7	5.5	2.7	10.5	8.3	5.5	11.1	8.3	11.1	13.8	3. Ceilings under unconditioned rooms	9.0	5.0	14.0	10.0	5.0	19.0	15.0	10.0	20.0	15.0	20.0	25.0
												Floors												
5.0	2.7	7.7	5.5	2.7	10.5	8.3	5.5	11.1	8.3	11.1	13.8	1. Over unconditioned rooms	9.0	5.0	14.0	10.0	5.0	19.0	15.0	10.0	20.0	15.0	20.0	25.0
0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	2. Over basement, enclosed crawl space or concrete slab on ground	0	0	0	0	0	0	0	0	0	0	0	0
5.0	2.7	7.7	5.5	2.7	10.5	8.3	5.5	11.1	8.3	11.1	13.8	3. Over open crawl space	9.0	5.0	14.0	10.0	5.0	19.0	15.0	10.0	20.0	15.0	20.0	25.0

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[a]Daily temperature range: L (Low) calculation value: 6.7C° (12F°) Applicable range: less than 8.3C° (15F°) | M (Medium) calculation value: 11.1C° (20F°) Applicable range: 8.3 to 13.8C° (15 to 25F°) | H (High) calculation value: 16.7C° (30F°) Applicable range: more than 13.8C° (25F°)

[c]For roofs in shade, 18-h average = 6.1C° (11F°) temperature differential. At 32.2°C (90F) design and medium daily range, equivalent temperature differential for light-colored roof equals 6.1 − (0.71)(21.6 − 6.1) = 17.1C° [11 − (0.71)(39 − 11) = 31F°].

Table B2: *Design Cooling Load Factors.*

Part B. Btu/h ft²

Outdoor Design Temp.[a]	Regular Single Glass						Regular Double Glass						Heat-Absorbing Double Glass						Clear Triple Glass		
	85	90	95	100	105	110	85	90	95	100	105	110	85	90	95	100	105	110	85	90	95
No Awnings or Inside Shading																					
North	23	27	31	35	39	44	19	21	24	26	28	30	12	14	17	19	21	23	17	19	20
NE and NW	56	60	64	68	72	77	46	48	51	53	55	57	27	29	32	34	36	38	42	43	44
East and west	81	85	89	93	97	102	68	70	73	75	77	79	42	44	47	49	51	53	62	63	64
SE and SW	70	74	78	82	86	91	59	61	64	66	68	70	35	37	40	42	44	46	53	55	56
South	40	44	48	52	56	61	33	35	38	40	42	44	19	21	24	26	28	30	30	31	33
Horiz. skylight	160	164	168	172	176	181	139	141	144	146	148	150	89	91	94	96	98	100	126	127	129
Draperies or Venetian Blinds																					
North	15	19	23	27	31	36	12	14	17	19	21	23	9	11	14	16	18	20	11	12	14
NE and NW	32	36	40	44	48	53	27	29	32	34	36	38	20	22	25	27	29	31	24	26	27
East and west	48	52	56	60	64	69	42	44	47	49	51	53	30	32	35	37	39	41	38	39	41
SE and SW	40	44	48	52	56	61	35	37	40	42	44	46	24	26	29	31	33	35	32	33	34
South	23	27	31	35	39	44	20	22	25	27	29	31	15	17	20	22	24	26	18	19	21
Roller Shades Half-Drawn																					
North	18	22	26	30	34	39	15	17	20	22	24	26	10	12	15	17	19	21	13	14	15
NE and NW	40	44	48	52	56	61	38	40	43	45	47	49	24	26	29	31	33	35	34	35	35
East and west	61	65	69	73	77	82	54	56	59	61	63	65	35	37	40	42	44	46	49	49	50
SE and SW	52	56	60	64	68	73	46	48	51	53	55	57	30	32	35	37	39	41	41	42	43
South	29	33	37	41	45	50	27	29	32	34	36	38	18	20	23	25	27	29	25	26	26
Awnings[b]																					
North	20	24	28	32	36	41	13	15	18	20	22	24	10	12	15	17	19	21	11	12	13
NE and NW	21	25	29	33	37	42	14	16	19	21	23	25	11	13	16	18	20	22	12	13	14
East and west	22	26	30	34	38	43	14	16	19	21	23	25	12	14	17	19	21	23	12	13	14
SE and SW	21	25	29	33	37	42	14	16	19	21	23	25	11	13	16	18	20	22	12	13	14
South	21	24	28	32	36	41	13	15	18	20	22	24	11	13	16	18	20	22	11	12	13

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[a]Based upon indoor design temperature of 23.8°C (75 F) and outdoor design temperatures as indicated. Interpolate to obtain factors for outdoor design temperatures other than those given.

[b]For other external shading devices which completely shade the glass at any orientation, use the values for "Awnings, north".

Appendix C

Cooling Process in the Psychrometric Chart

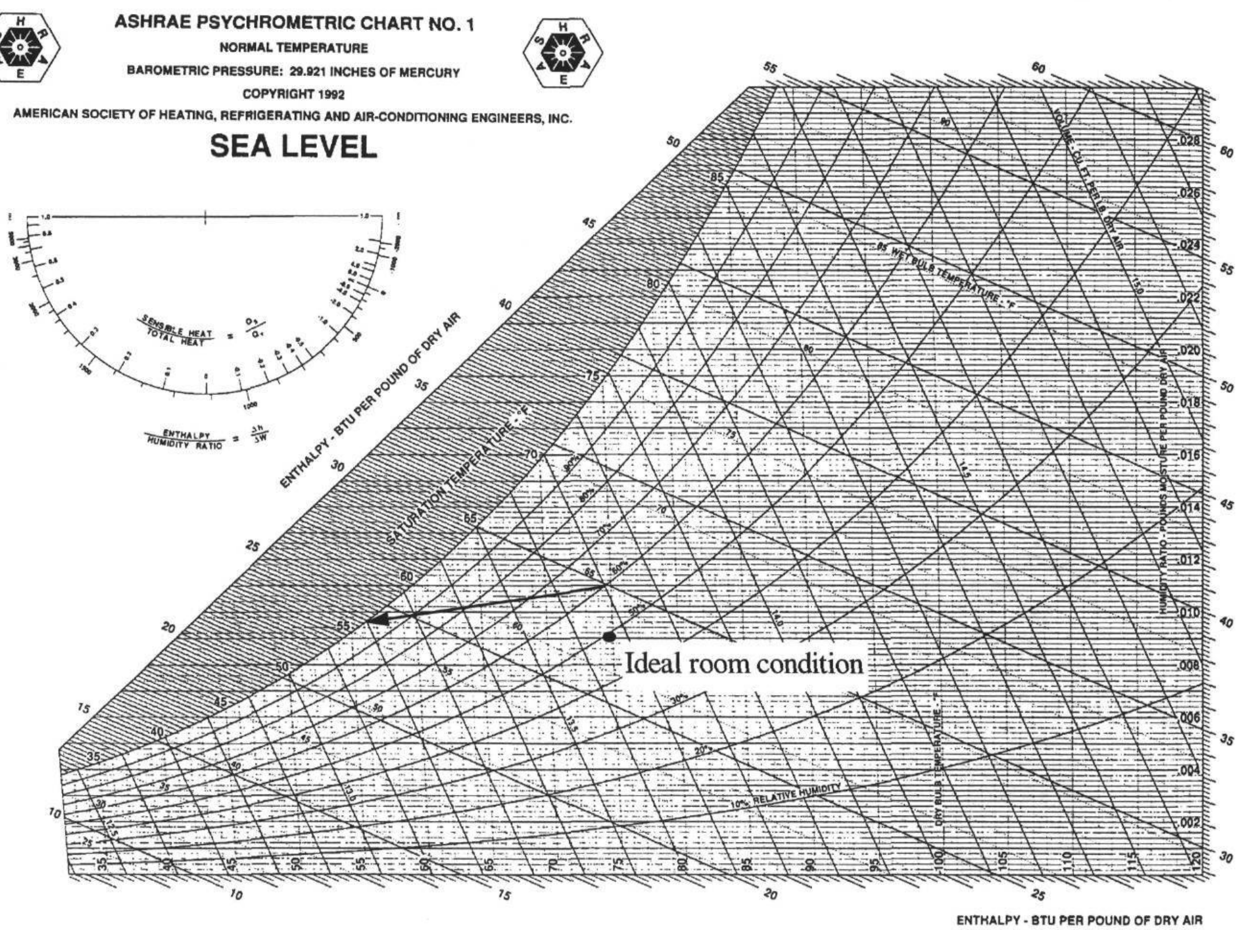

Fig. 1 ASHRAE Psychrometric Chart No. 1

Table 1. Summary of the heat pump units

Mark	Unit Numbers	CFM	CFM /ton	OA CFM	Fan HP	Cooling Capacity Sensible	Cooling Capacity Total	Tower Wtr GPM	Heating Btuh	EER	Model
HP-A	24	1,350	410	360	1/2	28,000	39,500	11	53,500	11.2	HE42
HP-B	4	1,600	393	360	1/2	35,000	48,900	13	65,000	11.6	HE52
HP-C	3	1,800	424	360	1/2	38,000	51,000	13	65,000	11.6	HE52
HP-D	2	1,400	425	375	1/2	30,700	39,500	11	53,500	10.7	HE42
HP-E	1	2,000	407	300	3/4	45,000	59,000	16.5	84,000	10.2	HE62
HP-F	1	585	425	90	1/6	13,100	16,500	4.8	23,500	10.8	HE19
HP-G	1	1,080	405	135	1/3	23,500	32,000	8.8	44,000	11.1	HE34
HP-H	1	1,530	391	300	1/2	34,800	46,900	13	65,000	10.7	HE52
HP-I	2	1,675	509	225	1/2	30,700	39,500	11	53,500	10.7	HE42
HP-J	1	800	376		1/4	18,300	25,500	6.8	3,400	12.0	HE27
Total	40	56,345		13,185							

General design criteria:

- 450 ± 100 ft^2/ton
- 1.5 cfm/ft^2 for exterior spaces
- 0.6 cfm/ft^2 for interior spaces
- 400 cfm/ton

Design condition for Willow Branch School

- 396 ft^2/ton
- 1.0 cfm/ft^2 fot entire space
- 376 ~ 509 cfm/ton, typical classroom is 410 cfm/ton

Exhaust fans:

- EF-1: 540 CFM
- EF-2: 540 CFM
- EF-3: 520 CFM
- EF-4: 960 CFM
- EF-5: 500 CFM
- Total: 3,060 CFM