

MODELING OF VENTILATION AIR HEAT RECOVERY AND ITS IMPACT IN HIGH-PERFORMANCE GREEN BUILDINGS

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ABSTRACT

A custom-developed spreadsheet tool was used to analyze the performance of a constant volume HVAC system, with options of ventilation air heat recovery (VAHR), air-side economizer (ASE) system, or a combination of VAHR and ASE, for a typical office space and classroom space, in five different climates in the U.S. For all but one of the cases studied, the results show that VAHR used the most energy on an annual basis.

INTRODUCTION

Recent experience with several green building projects indicated that there are times when VAHR systems may not save energy. It is true that VAHR systems reduce the installed capacity of heating and cooling equipment, especially for buildings requiring large amounts of ventilation air, like schools. This can significantly reduce the first cost of the heating and cooling systems by reducing the heating and cooling load due to outside air. In very hot and very cold weather VAHR typically reduces the HVAC energy usage as well.

There are two reasons though, why a VAHR system can increase annual energy usage. One is the substantial fan power required to move the air through the heat recovery heat exchanger. In this study fan power typical of heat recovery wheels was used (1.5 W/cfm). The other is that at times when ventilation air could be used to cool the building (like an air-side economizer), VAHR systems often continue to operate in heat recovery mode. Because green buildings typically have better insulated building skins, they also have relatively low balance temperatures. Balance temperature is the outside temperature at which the space heat gains balance the skin losses. For modern green buildings balance temperatures when occupied can be below 20°F. Therefore, because these green buildings typically need space cooling down to reasonably cold outdoor air conditions, ASE operation can substantially reduce or eliminate the cooling load on the HVAC system.

It is worthwhile to note that ASHRAE Standard 90.1-1999 requires VAHR systems for HVAC systems with a supply flow greater than 5,000 cfm, and with minimum outside air of 70% or greater of the design supply air quantity (ASHRAE 1999). Furthermore, if there is a VAHR system, Standard 90.1-1999 states that "Provision shall be made to bypass or control the heat recovery system to permit air economizer operation as required by 6.3.1.1." Many of the projects in the author's experience did not have this bypass capability, but could have benefited greatly if they had. This study will show the potential energy savings for systems that combine VAHR and ASE.

A literature survey of the last 20 years revealed very little published information on the performance of VAHR systems, especially those with heat recovery wheels. One relevant and recent study focused on ventilation air for ground source heat pump systems (Kavanaugh and Xie, 2000). The authors studied seven different ventilation air treatment options in 3 different climates using a custom spreadsheet, much like that used in this study. One of their conclusions is "Heat recovery units, which may be effective at reducing energy use when outdoor conditions are extreme, can consume more energy on an annual basis than conventional equipment if ancillary equipment (fans, pumps, etc.) energy use is high."

SIMULATION

This study extends the work of Kavanaugh to five climates, with the focus on three HVAC systems. The base case system is a constant volume air-to-air heat pump system for heating and cooling. Two options are considered for ventilation air. One is a heat wheel VAHR system, and the other is an ASE system. A third option is also considered that utilizes both VAHR and ASE, assuming that whichever results in minimum energy use will be utilized at that outdoor condition.

Because the focus of this study is ventilation air, two basic building types were considered. One is an office building with relatively low proportion of ventilation

air. The other is a school building with relatively high proportion of ventilation air. The scope of the modeling effort is to do preliminary analysis with a custom spreadsheet, and then to perform more rigorous analysis with DOE2.2. This paper reports the results of those preliminary analyses. One of the advantages of this spreadsheet is it allows one to see the effect of outside air temperature on system performance, illuminating the reasons for the differences in annual results. For this spreadsheet analysis, only one representative space, equivalent to an elementary school classroom size, was used.

The five different cities used in the analysis were Atlanta, Detroit, Houston, Los Angeles and New York. These cities were defined by an earlier study to represent the different climates in the United States (B. Andersson, W.L. Carroll, M.R. Martin 1982). Weather data for each city, in the form of bin temperature data, with mean coincident wet bulb temperature, was used (Departments of the Air Force, the Army, and the Navy, 1978). Only the data for the eight-hour period from 8 a.m. to 4 p.m. was used because the analysis was only for the occupied period of the building. This assumes that no ventilation air is used when the buildings are unoccupied. Thus the results only represent the HVAC energy use during the occupied period.

Standard assumptions used in the simulation for both buildings can be seen in Table 1. Table 2 represents assumptions made specific to each building type.

Other assumptions made include:

- The density and heat capacity of air and water vapor are constant
- Steady-state conditions
- In sensible load calculations we assume $\Delta h \approx c_p \Delta t$
- Volume flow rate of supply air is constant
- $h_{fg @ t-skin} \cong 1044 \frac{Btu}{lb}$
- Window solar gain is based on average hourly gain for a North-facing window with a shading coefficient of 0.4.

What follows is an explanation of the analysis contained in the spreadsheet. The desired output is the energy use for HVAC. The system modeled here is an all-electric heat pump system where electric usage includes compressor energy for heating, compressor energy for cooling, fan energy, and VAHR fan energy.

The analysis starts by finding the space load at a given outside air condition, determining the outside air load,

and the subsequent cooling or heating coil load. Coil loads are converted to electric usage using outside air temperature-dependent heat pump efficiencies. Both sensible and latent loads are considered.

We start by solving for the occupied space cooling sensible load $\dot{Q}_{c,s}$, essentially by assuming steady-state and summing up the internal and external load components. The analysis is separated into sensible loads and latent loads. Sensible loads are described first; latent loads are described near the end of this section (a positive load reflects a cooling load).

$$\dot{Q}_{c,s} = \dot{Q}_{int,s} + \dot{Q}_{ext,s} \quad (1)$$

where $\dot{Q}_{int,s}$ is the sum of the lighting load, equipment load, and people loads:

$$\dot{Q}_{int,s} = \left[(LPD + EPD)A_{floor} + (N_p) \dot{q}_{p,s} \left(\frac{1W}{3.413 \frac{Btu}{h}} \right) \right] \quad (2)$$

and $\dot{Q}_{ext,s}$ is:

$$\dot{Q}_{ext,s} = \dot{Q}_{sol} + \dot{Q}_{win} + \dot{Q}_{wall} \quad (3)$$

where

$$\dot{Q}_{sol} = \overline{SHGF}(SC)A_{win} \quad (4)$$

and

$$\dot{Q}_{win} + \dot{Q}_{wall} = [(U_{win}A_{win} + U_{wall}A_{wall})(t_o - t_r)] \quad (5)$$

Cooling coil sensible load, $\dot{Q}_{coil,s}$ is the sum of the space load and the outside air load:

$$\dot{Q}_{coil,s} = \dot{Q}_{c,s} + \dot{Q}_{oa,s} \quad (6)$$

$$\dot{Q}_{oa,s} = \dot{m}_{oa} c_{p,oa} (t_{oa} - t_r) \quad (7)$$

Since \dot{V}_{oa} is in cfm, and we assume standard air, then we get the sensible cooling coil load without heat recovery:

$$\dot{Q}_{coil,s} \cong \dot{Q}_{c,s} + 1.10 \dot{V}_{oa} (t_{oa} - t_r) \quad (8)$$

If heat recovery is being used, $\dot{V}_{oa} = \dot{V}_{oa,\min}$ and the net outside air load is reduced by the effectiveness of the heat recovery system.

$$\dot{Q}_{oa,s,VAHR} \cong (1 - \varepsilon_s) \dot{Q}_{oa,s} \quad (9)$$

For the ASE, we need to determine how much outside air is being brought into the system. To do this, we start with the mixing equation is:

$$\dot{m}_{oa} h_{oa} + (\dot{m}_{ma} - \dot{m}_{oa}) h_r \cong \dot{m}_{ma} h_{ma} \quad (10)$$

Because we assume standard air density, we can replace \dot{m} with \dot{V} . For h, we can substitute $c_p t$, reducing equation 10 to:

$$\dot{V}_{oa} t_{oa} + (\dot{V}_{ma} - \dot{V}_{oa}) t_r \cong \dot{V}_{ma} t_{ma} \quad (11)$$

Fraction of outside air, f_{oa} , can be defined as

$$f_{oa} = \frac{\dot{V}_{oa}}{\dot{V}_{ma}} \quad (12)$$

Multiplying Equation (11) by $\frac{1}{\dot{V}_{ma}}$ we get

$$f_{oa} t_{oa} + (1 - f_{oa}) t_r \cong t_{ma} \quad (13)$$

Then solving for f_{oa} we get:

$$f_{oa} = \frac{t_r - t_{ma}}{t_r - t_{oa}} \quad (14)$$

For an ASE, the f_{oa} is controlled as follows. When t_{oa} is greater than a certain temperature that is a few degrees cooler than the room temperature (to prevent high humidity air from being introduced), in this case 65 °F, minimum outside air is used. When t_{oa} is less than 65 °F, and greater than the desired mixed air setpoint, 100% outside air is used. When $t_{oa} \leq t_{ma-set}$ the room air is mixed with outside air to maintain t_{ma-set} .

We modify equation 14 to allow the determination of f_{oa} :

$$f_{oa} = \frac{t_r - t_{ma-set}}{t_r - t_{oa}} \quad (15)$$

If f_{oa} falls below $f_{oa-\min}$ then $f_{oa} = f_{oa-\min}$

Latent loads due to people and ventilation air are also accounted for and used to determine the coil latent load when cooling. This coil latent load is added to the coil sensible load to get the total coil load that is used in equation 16. Moisture calculations were based on algorithms in the 1997 ASHRAE Handbook (ASHRAE 1997).

In order to solve for the moisture generated by people, we start with latent heat given off by a person $\dot{q}_{p,l}$, from the ASHRAE Handbook for a moderately active office worker, 200 Btu/hr. A slightly smaller value was used for the classroom analysis as representative of children. Then,

$$\dot{q}_{p,l} = \dot{m}_p h_{fg@t-skin} \quad (16)$$

Knowing $\dot{q}_{p,l}$ and $h_{fg@t-skin}$, we can then solve for \dot{m}_p .

$$\dot{m}_p = \frac{200 \text{ Btu/hr} - \text{person}}{1044 \text{ Btu/lb}} = 0.19 \frac{\text{lb/hr}}{\text{person}}$$

An H₂O mass balance inside the room gives us:

$$\dot{m}_{sa} W_{sa} + \dot{m}_p N_p = \dot{m}_{sa} W_r \quad (17)$$

where:

$$\dot{m}_{v,sa} = \dot{m}_{sa} W_{sa} \quad (18)$$

The air mass flow rate, \dot{m}_{sa} , is determined by an energy balance on the room at design conditions:

$$\dot{Q}_{c,s} \cong \dot{m}_{sa} c_p (t_r - t_{sa}) \quad (19)$$

Since we assume that when the cooling coil is running and there is a latent load, $W_{SA} = W_{@90\% \phi}$, we can solve Equation (19) for W_r .

Enthalpy wheel latent effectiveness ε_l is defined as:

$$\varepsilon_l = \frac{W_{r,e} - W_r}{W_{oa} - W_r} \quad (20)$$

This allows us to solve for $W_{r,e}$ which is the moisture content of the exhaust air after the enthalpy wheel. If there is no VAHR, $W_{r,e} = W_r$. The cooling coil latent load is found by finding the flow rate of condensate leaving the cooling coil. An H₂O mass balance on the whole system (HVAC plus space) gives:

$$\dot{m}_{v,oa} + \dot{m}_p = \dot{m}_{v,r,e} + \dot{m}_c \quad (21)$$

Assuming that $\dot{m}_{a,oa} = \dot{m}_{a,r,e} \cong 4.5\dot{V}_{oa}$,

$$\dot{m}_c = 4.5\dot{V}_{oa}(W_{oa} - W_{r,e}) + N_p \left(0.20 \frac{lb}{hr \cdot per} \right) \quad (22)$$

From the flow rate of condensate, we can estimate the latent cooling coil load as:

$$\dot{Q}_{coil,l} \cong \dot{m}_c h_{fg @ 55F} = \dot{m}_c \left(1062 \frac{Btu}{lb} \right) \quad (23)$$

This latent cooling coil load is added to the sensible load to get the total cooling coil load.

To determine the electricity use by the heat pump:

$$COP = \frac{\dot{Q}_{coil}}{\dot{E}_c} \quad (24)$$

The variation of COP for cooling and for heating is accounted for in the spreadsheet using the default performance curves in DOE 2.2 (J. Hirsch 2004).

RESULTS

This section will begin with an overview of the results for annual energy usage. Then we will look in some detail at specific cases, utilizing the capability of the spreadsheet to show how each system performs as a function of outside temperature. Finally, we will examine the effect of heat recovery fan power on VAHR annual energy use.

The graphs of both the school's and office's annual electric usage, Figures 1 and 2, show that over a total year in all five climates, VAHR consistently uses the most energy. Out of the three HVAC systems modeled (not including the optimum), the ASE consistently used the least amount of annual energy.

The optimum bar represents a combination of the ASE plus the VAHR. This optimum system uses whichever system demands less electric power at a given outside temperature. For example, in Figure 3, a plot of electricity demand versus outside temperature in a Detroit school, shows that using heat recovery could save a considerable amount of energy over an ASE

during times when outside temperature is below about 40°F. Likewise, above temperatures of 40°F an ASE (or the base case with no VAHR or ASE) uses less energy. The optimum system would use the VAHR for outside temperatures of under 40°F, and the ASE for temperatures above 40°F (or simply shutting down the VAHR and bypassing it). It is interesting to note that in the annual energy use in Figure 1 there was little savings from either the VAHR or the ASE alone. That is because the savings from the VAHR below 40°F is roughly equivalent to the savings of the ASE above 40°F. By selecting the most energy efficient option at any given condition, energy savings is greatest.

One can also observe from these plots, Figures 3 to 6, that an ASE on a high occupancy school has little savings because the system is already bringing in a high proportion of outside ventilation air (around 67%).

Looking at the total annual electric usage for an LA school, shown in Figure 1, it can be seen that there is a large difference between the energy needed for a VAHR system compared to an ASE system. This can be easily explained by Figure 4, which illustrates the electricity demand at a given temperature. Though it is seen that the VAHR saves energy below about 40°F, there are no bin hours in that temperature range. This is also the reason that the ASE is equal to the optimum.

Also noteworthy for the LA school is that for temperatures above about 72 °F, the cooling load is less for the VAHR system than an ASE system, as seen in Figure 5. This does not result in a savings as would be seen on Figure 4, because of the power needed for the fans in the VAHR system. If we reduce the fan use from 1.5W/cfm to 0.75 W/cfm, as seen in Figure 6, a small savings can be seen for temperatures above about 83 °F.

One aspect of a VAHR system that significantly impacts the annual energy use is the fan power. The value of 1.5 W/cfm was based on actual green building projects. It is possible to reduce the fan power, for example, by installing a larger heat recovery wheel. This would reduce the face velocity and therefore reduce the fan power. To examine the effect of reducing fan power we ran the model with VAHR fan power of 0.75 W/cfm. The results show that with one exception the VAHR still consistently used the most energy. The exception was the Detroit school case, as seen in Figure 7. The main reason VAHR saves energy in this Detroit case is the Detroit climate has considerable hours at colder temperatures where the VAHR system uses much less energy than the other systems.

To get some sense of the reasonableness of the VAHR fan power assumptions used here, we compared two other published values. In the work of Kavanaugh (Kavanaugh and Xie, 2000), the power for the VAHR fans and wheel motor was equivalent to 1.40 W/cfm. This is in the range of the 1.5 W/cfm used for the majority of the cases here. In a recent ARI (Air-conditioning and Refrigeration Institute) publication (ARI, 2003), they present an example in Appendix C that is equivalent to 0.61 W/cfm. This value is low compared to the values studied here. It would be useful to survey other actual VAHR installations to determine the range of fan power requirements.

Because the performance of a VAHR system is greatly affected by the fan power associated with the system, it is important for designers to select systems that tend to reduce the fan power, and to carefully consider the impact through detailed energy analysis. As stated earlier, in buildings with large ventilation requirements, like schools, systems with VAHR should also have the ability to bypass the VAHR when an ASE is advantageous.

CONCLUSIONS/RECOMMENDATIONS

These preliminary results show that, for the constant volume HVAC system and climates studied here, VAHR is not the best option from the standpoint of efficient energy use. Because green buildings are well insulated and have low balance temperatures, ASE systems can significantly reduce cooling energy use. If a VAHR system is installed in a building to reduce the installed system capacities, the system should have the option of bypassing the VAHR system and using an ASE when advantageous. Attention should also be given in the VAHR system design to reducing the fan power requirements.

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NOMENCLATURE

- A_{wall} = area of wall
- A_{win} = area of window
- COP = coefficient of performance
- EPD = equipment power density
- \mathcal{E}_s = enthalpy wheel sensible effectiveness
- \mathcal{E}_l = enthalpy wheel latent effectiveness
- $h_{fg @ t-skin}$ = enthalpy of vaporization at skin temperature
- h_{ma} = enthalpy of mixed air
- h_{oa} = enthalpy of outside air
- h_r = enthalpy of room air
- LPD = lighting power density
- $\dot{m}_{a,oa}$ = mass flow rate of outside air entering system
- $\dot{m}_{a,r,e}$ = mass flow rate of air exiting system
- \dot{m}_c = mass flow rate of condensate from coil
- \dot{m}_{ma} = mass flow rate of mixed air
- \dot{m}_p = mass flow rate of moisture from people
- \dot{m}_{sa} = mass flow rate of supply air
- $\dot{m}_{v,r,e}$ = mass flow rate of vapor exiting system
- $\dot{m}_{v,oa}$ = mass flow rate of vapor in outside air
- $\dot{m}_{v,sa}$ = water vapor flowrate in supply air
- N_p = number of people
- $\dot{Q}_{c,s}$ = occupied space sensible cooling load
- \dot{Q}_{ext} = external sensible load
- \dot{Q}_{int} = internal sensible load
- $\dot{q}_{p,l}$ = rate of latent heat gain from occupants
- $\dot{q}_{p,s}$ = rate of sensible heat gain from occupants
- \dot{Q}_{oa} = cooling load from outside air
- \dot{Q}_{coil} = coil cooling load

$\dot{Q}_{coil,l}$ = latent cooling load of coil

\dot{Q}_{sol} = heat gain from sun

\dot{Q}_{win} = heat gain through window

\dot{Q}_{wall} = heat gain through wall

\overline{SHGF} = average solar heat gain factor

t_{oa} = air temperature outside

t_r = air temperature of room

t_{sa} = temperature of supply air

U_{win} = overall heat transfer coefficient of window

U_{wall} = overall heat transfer coefficient of wall

\dot{V}_{oa} = volumetric flow rate of outside air

\dot{V}_{ma} = volumetric flow rate of mixed air

$W_{r,e}$ = humidity ratio of air exiting system

W_{oa} = humidity ratio of air outside

W_{sa} = humidity ratio of supply air at 90% relative
humidity, and at t_{sa}

W_r = humidity ratio of room air

APPENDIX

Table 1 Room and system assumptions

Room Description		
Room Temperature	74	F
Width	28	ft
Depth	24	ft
Wall Height	12	
Floor Area	672	ft ²
Wall Area	336	ft ²
Window Fraction	0.5	
Window Area	168	ft ²
Wall U-factor	0.10	Btu/hr-ft ² -F
Window U-factor	0.50	Btu/hr-ft ² -F
Fresh Air Amount	15	cfm/person
HVAC Parameters		
COP Heating	3	
COP Cooling	3	
Sensible Effectiveness	0.7	
Latent Effectiveness	0.7	
Supply Fan Power	1.0	W/ft ²

Table 2 School and office assumptions

	School	Office
Lighting Power Density (W/ft ²)	1.2	1.0
Equipment Power Density (W/ft ²)	0.5	1.0
Number of People	25	5
People Sensible Gain (Btu/hr-person)	200	250

Table 3 School electric usage

City	Electric Usage			
	School			
	All units in (kWh/ft ²)			
	HR	No HR	Econ	Optimum
Atlanta	9.09	7.11	6.94	6.77
Detroit	7.84	7.63	7.50	5.94
Houston	10.09	8.44	8.31	8.22
LA	8.93	5.76	5.37	5.37
New York	7.85	6.44	6.27	5.55

Table 4 Office electric use

City	Electric Usage			
	Office			
	All units in (kWh/ft ²)			
	HR	No HR	Econ	Optimum
Atlanta	4.80	4.26	3.88	3.87
Detroit	4.01	3.60	3.27	3.11
Houston	5.31	4.95	4.68	4.66
LA	4.73	4.10	3.51	3.51
New York	4.06	3.44	3.02	2.98

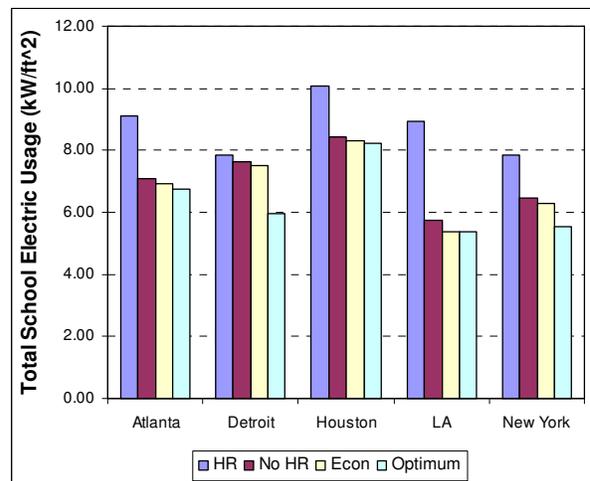


Figure 1 Total school electric usage

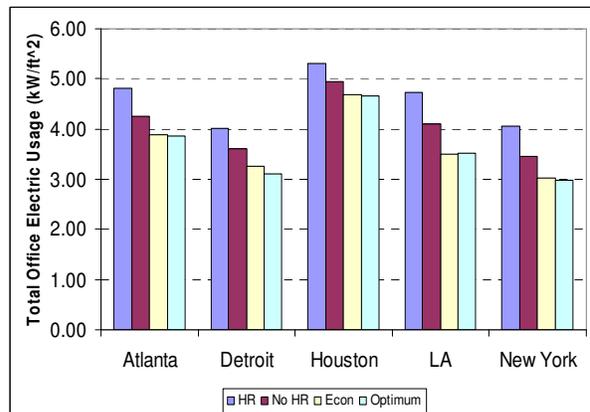


Figure 2 Total office electric usage

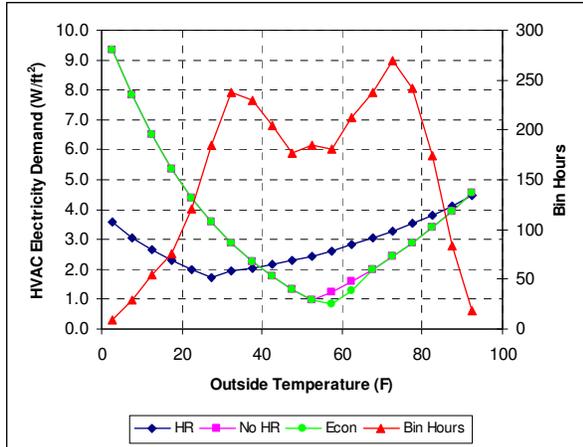


Figure 3 Detroit school electricity demand

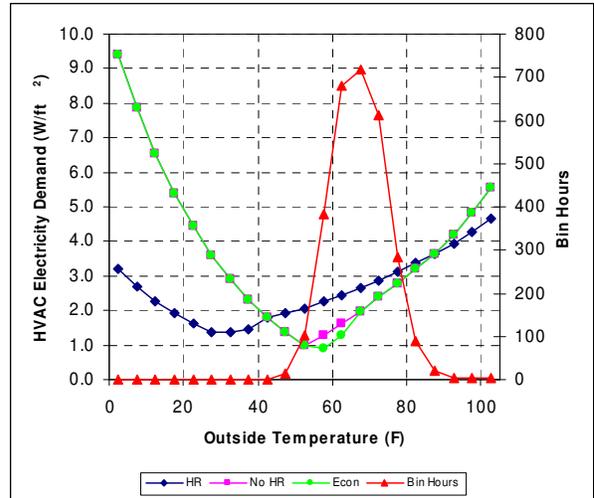


Figure 6 LA school cooling load with half the fan use power

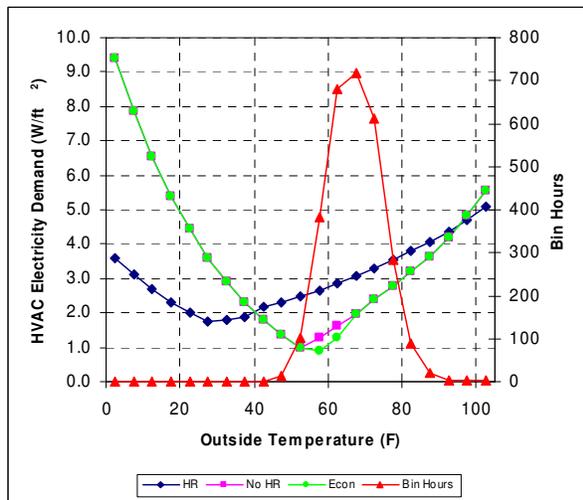


Figure 4 LA school electricity demand

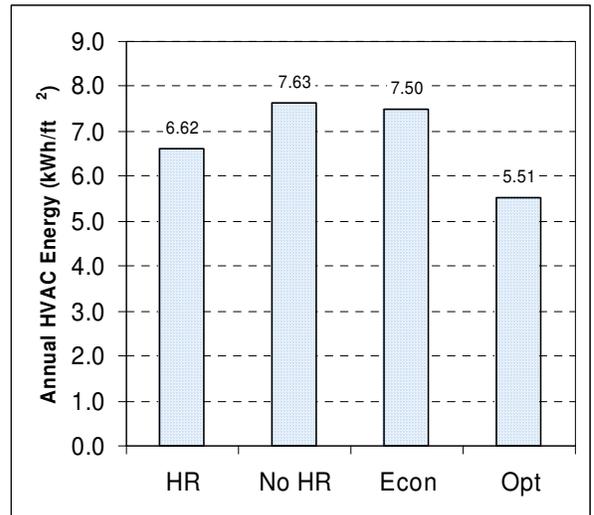


Figure 7 Detroit school energy usage with reduced VAHR fan power

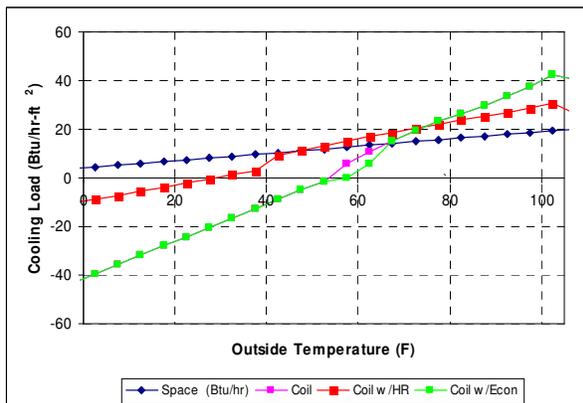


Figure 5 LA school cooling load