ANALYSIS OF DEDICATED OUTDOOR AIR SYSTEMS FOR DIFFERENT CLIMATES

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ABSTRACT

Dedicated outdoor air systems (DOAS) have been proposed to treat the outdoor air before it enters the building and thus reduce the load on the zone equipment. This paper presents a study of different DOAS configurations and their energy and power consumptions for multiple climates of the United States. Based on the simulation results, the DOAS showed promise in reducing energy consumption relative to the baseline system in the office building in all climates studied. A more complex DOAS did not show significant improvement over a simple DOAS consisting of only a preheat coil and enthalpy wheel.

INTRODUCTION

A number of different approaches to introducing and controlling outdoor air intake and ventilation air distribution have been employed in commercial buildings, with potentially significant impacts on both indoor air quality (IAQ) and the operating costs associated with energy consumption. Typical ventilation systems in commercial buildings employ equipment intended to simultaneously meet both outdoor air ventilation and space conditioning requirements, using air distribution approaches intended to provide supply air to a conditioned space that is a mixture of outdoor air and recirculated air. More recently, ventilation approaches have been proposed, and in some cases installed, that separate the outdoor air ventilation and space conditioning functions (dedicated outdoor air systems or DOAS). DOAS systems have been described recently in numerous journal articles (Coad 1999, Gatley 2000a, Gatley 2000b, Gatley 2000c, Khattar and Brandemuehl 2002, Morris 2003, and Mumma 2001). A few detailed analyses of specific DOAS systems have also been published (Jeong et al. 2003, Khattar and Brandemuehl 1996).

The U.S. EPA's Indoor Environment Division (IED) has recently completed a software package that allows building designers and engineers to evaluate the potential financial payback and humidity control benefits of energy recovery ventilation (ERV) systems for school applications (available at http://www.epa.gov/iaq/schooldesign/saves.html). In the first phase of their Advanced Ventilation Systems effort, EPA determined that there are first cost, operating cost, and IAQ advantages to bringing outdoor air into schools using readily available ERV technologies, as compared to the conventional approaches used during most of the latter half of the 20th century. Additional work is needed to determine if there are first and operating cost, and IAQ advantages to using dedicated outdoor air systems in lieu of conventional mixing ventilation in commercial buildings. At the same time, there is also a need to identify any potential limitations of DOAS systems.

The objective of this simulation study is to perform an initial evaluation of the potential benefits and limitations of DOAS in commercial buildings.

METHOD

A combined airflow-building energy modeling tool linking TRNSYS (Klein, et al 2000) and CONTAM (Dols, et al 2002) (McDowell, et al 2003) was used to study the energy impact of DOAS on a modern office building in multiple U.S. climate types. Simple HVAC systems representative of system types used in typical buildings are included to model the energy requirements of the buildings along with DOAS models to assess their impact on energy usage. Building model parameters are chosen such that the buildings would be considered typical new construction and meet current ASHRAE Standard 90.1 (ASHRAE 2001) requirements. Simulations of annual energy employ Typical Meteorological Year, TMY2, files (Marion and Urban 1995) for five different cities representing different climates of the U. S. (Miami, FL; Phoenix, AZ; St. Louis, MO; Bismarck, ND; and Minneapolis, MN).

SIMULATIONS

This section describes in detail the building and systems modeled in the study.

Thermal and Structural

The building modeled in this study is a two story office building with a total floor area of 2250 m^2 and a floorplan as shown in Figure 1. The building has a window-to-wall ratio of 0.2 with a floor-to-floor height of 3.66 m, divided between a 2.74 m wall height for the occupied space and a 0.92 m plenum height per floor. The building includes a single elevator shaft.

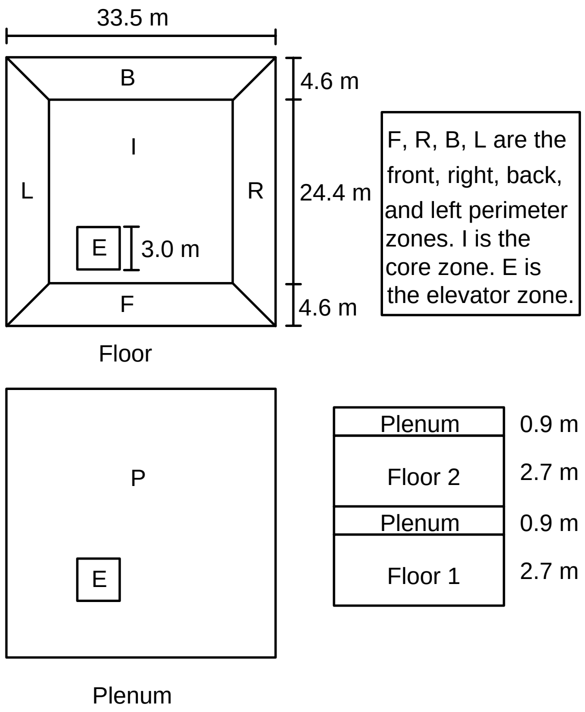

Figure 1 Building Floorplan

The building model was developed so that the thermal envelope construction would satisfy the requirements of ASHRAE Standard 90.1-2001. To meet these requirements the wall, roof and slab constructions varied for the different locations as shown in Tables 1 through 4.

The window properties used in the model are a heat loss factor, U = 3.24 W/m^2-K and a solar heat gain coefficient, SHGC = 0.39 for St. Louis, Bismarck, and Minneapolis and U = 6.93 W/m^2-K and SHGC = 0.25 for Miami and Phoenix.

The internal gains for the occupied spaces are divided into three parts: lighting, receptacle loads, and occupancy. These gains are all applied using a peak value and fraction of peak schedule. The lighting peak is 10.8 W/m^2, the peak receptacle load is 6.8 W/m^2, and the peak occupancy density is 5 persons/100 m^2. The fraction of peak schedules are shown in Figures 2 through 4.

The thermostat setpoints are capable of operating with setback and setup basis depending on whether the building is in heating or cooling mode, respectively. The heating setpoint is 21.1 °C with a setback temperature of 12.8 °C and the cooling setpoint is 23.9 °C with a setup temperature of 32.2 °C. The schedule for the thermostat settings differ between weekdays (hours from 6 to 20 at setpoint), Saturdays (hours from 7 to 14 at setpoint) and Sundays (always at setup/setback). For the first hour of operation at setpoint, the system does not bring any outdoor air into the zone. This pre-occupancy hour is is used to bring the zone back to setpoint from the setup/setback temperature.

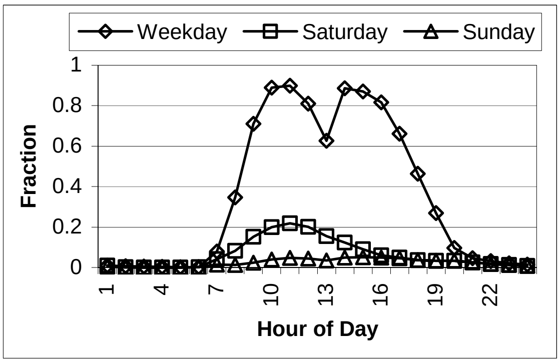

Figure 2 Fractional Occupancy Schedule

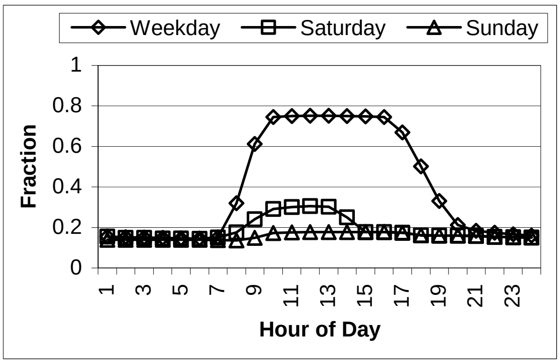

Figure 3 Fractional Lighting Schedule

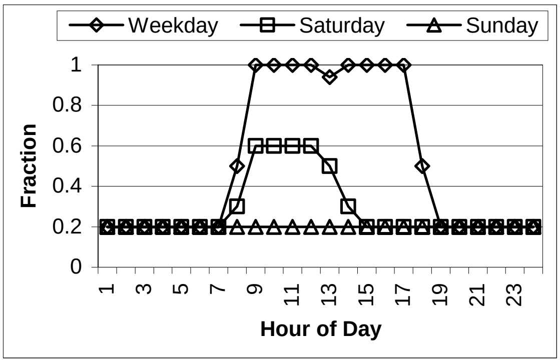

Figure 4 Fractional Receptacle Load Schedule

The amount of outdoor air provided to each zone was 0.43 l/s/m^2 to meet the requirements of Table 6-1 of ASHRAE Standard 62.1 (ASHRAE 2004). The infiltration and interzonal airflows are calculated using the CONTAM airflow modeling program linked with TRNSYS. The leakage area value used for the exterior envelope of the building is 1.3 cm^2/m^2 at 10 Pa which is equivalent to a leakage rate of 2.6 m^3/(h•m^2) at 75 Pa or approximately equal to the tightest buildings included in a review of commercial building airtightness measurements (Persily 1998).

System Models

Four different HVAC systems were modeled in this study: Baseline, Baseline with Economizer, Simple DOAS, and Full DOAS.

Baseline:

The baseline system modeled included Water Source Heat Pumps (WSHPs) with a Cooling Tower and a Boiler serving the common loop. Each zone has its own WSHP rejecting/extracting heat from the common loop. Since this common loop was to be the same in all of the cases, the power comsumption of the ciculating pump was assumed to be the same and thus was neglected in the study. The unconditioned outdoor air for each zone is supplied to each individual heat pump and thus the heat pump blower is on at all times when the zone is occupied.

The WSHP is modeled as a single-stage liquid source heat pump. The heat pump conditions a moist air stream by rejecting energy to (cooling mode) or absorbing energy from (heating mode) a liquid stream. This model is based on user-supplied data files containing catalog data for the capacity (both total and sensible in cooling mode), and power, based on the entering water temperature to the heat pump, the entering water flow rate and the air flow rate. Other curve fits are used to modify the capacities and power based on off-design indoor air temperatures. With the performance calculated from the entering conditions the conditioned air stream, after both sensible and latent effects are applied, is then added to the building zone. The conditions of the zone, both sensible and latent, are then calculated based on the weather, internal gains, and conditioned air stream.

The boiler is modeled as a simple fluid heater. It is assumed to have enough capacity to always maintain the required temperature and calculates the required input energy to maintain the temperature. In this system the boiler setpoint is the minimum temperature of the liquid stream and the setpoint used in the model is 15.6 °C and the boiler efficiency is 0.81 (typical for a natural gas boiler). It is also assumed that there are no losses from the tank to ambient.

The cooling tower is modeled as a single cell counterflow cooling tower and sump which rejects heat from the liquid stream to the environment. The tower fan has three speeds: natural convection (no airflow), low and high. The tower performance is calculated using a mass transfer analog and the leading coefficient was 1.0 with an exponent of – 0.6.

To make the comparison between the different systems equivalent, supply and exhaust fans for the ventilation air were added to the base model. Since they only had dampers and ductwork pressure drops to overcome they were assumed to be 0.37 kW fans.

Baseline with Economizer:

The second system added an economizer to the baseline system. Each WSHP has its own economizer that internally determines an appropriate mixture of outdoor and return air that will result in air delivered to the zone at the same enthalpy as air that would be delivered by a cooling coil to satisfy the space load; except for the Phoenix model where it is controlled based on temperature rather than enthalpy. The economizers are active any time that the zone is calling for cooling and the outdoor air enthalpy (temperature in Pheonix) is less than the zone enthalpy (temperature). If the increased outdoor is not sufficient to meet the cool load then the cooling coil is activated to meet the remaining load.

To make the comparison between the different systems equivalent, the same supply and exhaust fans for the ventilation air were used as in the base model.

Simple DOAS:

In this system the baseline WSHP system was augmented with a DOAS that consisted of only a preheat coil and an enthalpy wheel. The outdoor air for all of the zones is treated with the exhaust air from all of the zones in a single DOAS system. All of the required ventilation air is brought in using the DOAS and the WSHP simply treats recirculated air. There is also no economizer system included.

The preheat coil is modeled as a simple air heating device that maintains the outdoor air above a minimum intake temperature to prevent frost build-up in the enthalpy wheel. The setpoint for this system is – 4 °C. The coil is a gas coil with an efficiency of 0.8 and no external losses to the ambient.

The enthalpy wheel uses a "constant effectiveness – minimum capacitance" approach to model an air to air heat recovery device in which two air streams are passed near each other so that both energy and possibly moisture may be transferred between the streams. In this model both the sensible and latent effectivenesses are set to 0.8. The enthalpy wheel is assumed to be energized and rotating whenever outdoor air is required and use a 0.37 kW motor.

To make the comparison between the different systems equivalent, supply and exhaust fans for the ventilation air were added to the model. They were assumed to be 0.56 kW fans. The added fan power over the baseline system is to account for pressure losses associated with the ERV components, primarily the enthalpy wheel heat exchanger.

Full DOAS:

In this system a DOAS was added to the baseline WSHP with the design intent of meeting all of the latent loads of the zones as well as some of the sensible loads. This system consists of a preheat coil, an enthalpy wheel, a cooling coil, a sensible energy wheel and fans (Shank and Mumma 2001). A system diagram is shown in figure 5. All of the outdoor air for the zones is treated with one DOAS.

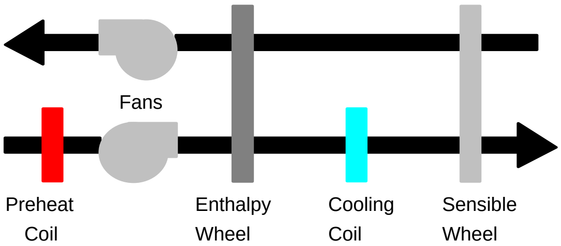

Figure 5 Full DOAS System Diagram

The preheat coil is modeled as a simple air heating device that maintains the outdoor air above a minimum intake temperature to prevent frost build-up in the enthalpy wheel. The setpoint for this system is –4 °C. The coil is a gas coil with an efficiency of 0.8 and no external loses to the ambient.

In order to meet all of the latent loads of the zones, the enthalpy wheel is controlled based on the outdoor air conditions. It is operated at full speed, modulated speed to maintain a setpoint or turned off. When the outdoor air enthalpy is higher than the exhaust air enthalpy, the wheel is at full speed. When the outdoor air enthalpy is less than or equal to the exhaust air enthalpy and the outdoor air dewpoint is higher than 11.1 °C, the enthalpy wheel is off. Otherwise the wheel speed is modulated. The full-on and off conditions are straight-forward to simulate, but the modulating speed condition offers some difficulties. While modulating the speed of the enthalpy wheel is a widely used method of control, the manufacturer's data rarely contains the information on how the performance varies with changing rotational speed. A modeling method for determining the sensible and latent effectiveness for a modulated enthalpy wheel (Jeong et al 2003) was modeled in TRNSYS but some shortcomings were determined. The algorithm uses a supply air humidity ratio setpoint, but the referenced paper does not describe the conditions for determining this value. In these simulations, the humidity ratio at the supply air drybulb and dewpoint temperatures was used, though it is unusual to control to both these conditions at the same time. The algorithm also uses a value called the driving force ratio (DFR) that is used to determine whether the latent or sensible efficiency leads at the current state of outdoor and return air conditions. It was found that problems could arise when the drybulb temperatures or the humidity ratios were very close or when they differed in opposite directions – leading to infinite or negative values for DFR. The calculations had to be bounded to prevent this condition from occurring in the model.

Another missing piece of the algorithm was the power required to rotate the enthalpy wheel. A review of catalog data for enthalpy wheels showed that the typical motor size for these type wheels is 0.37 kW to 0.56 kW. Since modulating the rotational speed of the enthalpy wheel would likely have only a small effect on the power consumption, it was decided to have the enthalpy wheel draw its 0.37 kW every timestep that the wheel is energized. The rating values of sensible and latent effectiveness of the enthalpy wheel for this analysis are both 0.8.

The cooling coil is controlled in two different ways. When the outdoor air dewpoint is greater than 7.2 °C then the leaving air drybulb temperature is set to 7.2 °C. When the outdoor air dewpoint is less than or equal to 7.2 °C, then the leaving air drybulb temperature is controlled to 12.8 °C. While controlling the leaving air temperature is easy enough the difficulty lies in determining how much input energy is required. Since there is not a chiller available to provide cold water the cooling coil would most likely be a direct expansion (DX) coil. Determining the power consumption of a DX coil system (compressor and condenser) is also not easy. The approach taken in this model was to regress some available performance data to approximate how the DX coil system COP varies with condenser drybulb and evaporator wetbulb temperatures. The regression equation was then normalized to the COP at standard ARI rating conditions. The resulting equation is:

$$COP = COPstd*(1.283142-0.01762*CondDB + 0.017961*EvapWB)$$

Once the energy required by the cooling coil is determined, the power input to the DX coil system can be approximated.

The sensible wheel is used to provide the required reheat to maintain the supply air temperature above 12.8 °C with an effectiveness of 0.8. To approximate the energy required to rotate this wheel the same assumptions were used as for the enthalpy wheel, i.e., 0.37 kW of power consumed for every timestep the wheel is energized.

Fan power is dependent on the fan curve of the actual fan and the pressure drop through all of the components in the system. We do not have enough of this information to model the fan performance in detail, so an alternative method to approximate the fan power was used. Based on product selection data for a DOAS unit based on the airflow required and pressure drop through the components it was determined that a 1.12 kW supply fan and a 0.75 kW exhaust fan would be required. At every timestep that the DOAS system was active, the fans drew 1.12 kW and 0.75 kW of power respectively. The added supply fan power (over the baseline system) accounts for pressure losses associated with the preheat coil, cooiling coil, enthalpy wheel and the sensible wheel, while the added exhaust fan power is associated with pressure losses due to the enthalpy and sensible wheels.

RESULTS

The predicted annual electric and gas consumption for the whole office building for each of the systems for the different climates are shown in Table 5 for St Louis, Table 6 for Bismarck, Table 7 for Minneapolis, Table 8 for Miami and Table 9 for Phoenix.

From the annual energy consumption, annual HVAC energy costs were calculated assuming $0.08 per kWh for electricity and $0.60 per therm for natural gas. As shown in Figure 6, the simple DOAS resulted in savings ranging from 14 % to 37 % and the full DOAS in slightly higher savings ranging from 21 % to 38 %. Although modeling different systems, buiding spaces and loads, other simulation studies have reported similar predicted savings of 14 % to 27 % (Khattar and Brandemuehl 2002) and 42 % (Jeong et al. 2003).

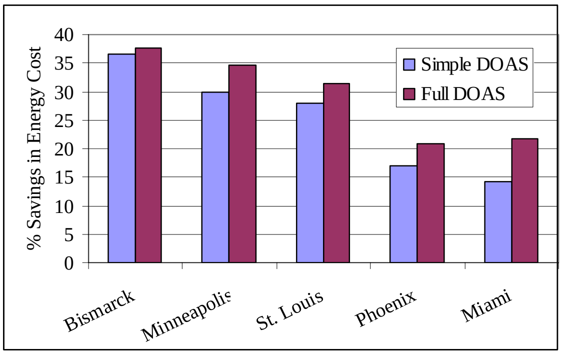

Figure 6 Savings in Annual Energy Costs Relative to Baseline WSHP system

CONCLUSIONS

Both the simple and full DOAS show promise in reducing the energy consumption of the WSHP system in the office buildings in all of the climates. The more complex DOAS did not show significant improvement over a simple DOAS consisting of only a preheat coil and enthalpy wheel except in Miami. The more complex DOAS was intended to meet the entire latent load of the space allowing the sensible load to be met with a radiant system (Jeong et al. 2003). Using a radiant system may increase the savings enough to justify the added cost of the more complex DOAS system. The system modeling still showed latent cooling being provided by the WSHPs in the zones. While this does not mean that the radiant system would not provide adequate comfort to the occupants without surface condensation, further study of this issue is needed.

ACKNOWLEDGMENT

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Table 1
External Wall Layer Properties for St. Louis, Miami and Phoenix

	Thickness	**Conductivity**	**Density**	**Specific Heat**	**Resistance**
Description	m	W/m-K	kg/m^3	kJ/kg-K	m^2-K/W
Face brick	0.092	0.879	1922	0.921	0.10
Vertical wall air layer					0.16
Gypsum board	0.0127	0.160	800	0.837	0.079
Steel studs w/mineral wool	0.089	0.0751	288	1.298	1.2
Gypsum board	0.0159	0.160	800	0.837	0.099

Table 2
External Wall Layer Properties for Bismarck and Minneapolis

	Thickness	**Conductivity**	**Density**	**Specific Heat**	**Resistance**
Description	m	W/m-K	kg/m^3	kJ/kg-K	m^2-K/W
Face brick	0.092	0.879	1922	0.921	0.10
Vertical wall air layer					0.16
Gypsum board	0.0127	0.160	800	0.837	0.079
Steel studs w/mineral wool	0.089	0.0751	288	1.298	1.2
Expanded polystyrene	0.0254	0.0277	29	1.214	0.88
Gypsum board	0.0159	0.160	800	0.837	0.099

Table 3
Roof Layer Properties for All Locations

	Thickness	**Conductivity**	**Density**	**Specific Heat**	**Resistance**
Description	m	W/m-K	kg/m^3	kJ/kg-K	m^2-K/W
Built-up roofing	0.0095	1.63	1120	1.47	0.058
Polyisocyanurate insulation	0.0634	0.0242	24	1.59	2.62
Fiber board sheathing	0.0128	0.0554	288	1.298	0.23

Table 4
Slab Layer Properties for All Locations

	Thickness	**Conductivity**	**Density**	**Specific Heat**	**Resistance**
Description	m	W/m-K	kg/m^3	kJ/kg-K	m^2-K/W
Concrete normal weight	0.127	1.31	2240	0.837	0.097

Table 5
Power Consumption (kJ) for St Louis

	Base	Base with Economizer	Simple DOAS	Full DOAS
Heat Pumps	1.70E+08	1.67E+08	1.07E+08	6.68E+07
Tower	7.32E+06	7.09E+06	7.38E+06	4.27E+06
Supply Fan	4.98E+06	4.98E+06	7.46E+06	1.49E+07
Return Fan	4.98E+06	4.98E+06	7.46E+06	9.95E+06
Enthalpy Wheel			1.18E+07	3.36E+06
Cooling Coil				3.09E+07
Sensible Wheel				2.60E+06
Total Electric	**1.867E+08**	**1.839E+08**	**1.413E+08**	**1.328E+08**
Boiler	1.14E+08	1.14E+08	5.01E+07	5.42E+07
Preheat			6.06E+06	6.06E+06
Total Gas	**1.140E+08**	**1.141E+08**	**5.614E+07**	**6.022E+07**

Table 6
Power Consumption (kJ) for Bismarck

	Base	Base with Economizer	Simple DOAS	Full DOAS
Heat Pumps	1.64E+08	1.63E+08	7.82E+07	5.85E+07
Tower	2.88E+06	2.73E+06	2.77E+06	8.97E+05
Supply Fan	4.98E+06	4.98E+06	7.46E+06	1.49E+07
Return Fan	4.98E+06	4.98E+06	7.46E+06	9.95E+06
Enthalpy Wheel			1.18E+07	2.86E+06
Cooling Coil				1.54E+07
Sensible Wheel				1.41E+06
Total Electric	**1.773E+08**	**1.754E+08**	**1.077E+08**	**1.039E+08**
Boiler	2.71E+08	2.71E+08	1.54E+08	1.59E+08
Preheat			3.57E+07	3.57E+07
Total Gas	**2.709E+08**	**2.710E+08**	**1.900E+08**	**1.951E+08**

Table 7
Power Consumption (kJ) for Minneapolis

	Base	Base with Economizer	Simple DOAS	Full DOAS
Heat Pumps	1.57E+08	1.55E+08	8.87E+07	5.92E+07
Tower	4.00E+06	3.84E+06	3.85E+06	1.46E+06
Supply Fan	4.98E+06	4.98E+06	7.46E+06	1.49E+07
Return Fan	4.98E+06	4.98E+06	7.46E+06	9.95E+06
Enthalpy Wheel			1.18E+07	3.04E+06
Cooling Coil				2.12E+07
Sensible Wheel				1.88E+06
Total Electric	**1.708E+08**	**1.686E+08**	**1.192E+08**	**1.117E+08**
Boiler	2.54E+08	2.55E+08	1.53E+08	1.39E+08
Preheat			2.76E+07	2.76E+07
Total Gas	**2.544E+08**	**2.546E+08**	**1.808E+08**	**1.662E+08**

Table 8
Power Consumption (kJ) for Miami

	Base	Base with Economizer	Simple DOAS	Full DOAS
Heat Pumps	3.02E+08	3.00E+08	2.39E+08	1.47E+08
Tower	1.90E+07	1.88E+07	1.81E+07	1.33E+07
Supply Fan	4.98E+06	4.98E+06	7.46E+06	1.49E+07
Return Fan	4.98E+06	4.98E+06	7.46E+06	9.95E+06
Enthalpy Wheel			1.18E+07	4.75E+06
Cooling Coil				6.42E+07
Sensible Wheel				4.80E+06
Total Electric	**3.308E+08**	**3.290E+08**	**2.841E+08**	**2.592E+08**
Boiler	3.26E+05	3.26E+05	3.26E+05	3.26E+05
Preheat			0.00E+00	0.00E+00
Total Gas	**3.257E+05**	**3.257E+05**	**3.257E+05**	**3.257E+05**

Table 9
Power Consumption (kJ) for Phoenix

	Base	Base with Economizer	Simple DOAS	Full DOAS
Heat Pumps	2.49E+08	2.46E+08	1.87E+08	1.51E+08
Tower	1.16E+07	1.14E+07	1.13E+07	1.01E+07
Supply Fan	4.98E+06	4.98E+06	7.46E+06	1.49E+07
Return Fan	4.98E+06	4.98E+06	7.46E+06	9.95E+06
Enthalpy Wheel			1.18E+07	3.37E+06
Cooling Coil				2.34E+07
Sensible Wheel				1.67E+06
Total Electric	**2.710E+08**	**2.671E+08**	**2.255E+08**	**2.145E+08**
Boiler	3.70E+06	3.84E+06	1.41E+06	1.86E+06
Preheat			0.00E+00	0.00E+00
Total Gas	**3.703E+06**	**3.837E+06**	**1.414E+06**	**1.864E+06**