Realistic Analysis of Air Conditioning Technologies

Prakash Dhamshala[1]

Abstract - This paper deals with the use of a computer code, Transient Analysis of Building Loads and Energy Technologies (TABLET) developed by the author for the analysis of various air-conditioning technologies. Conventional techniques such as thermostat setback or energy recovery as well as emerging technologies such as dedicated outside air Systems (DOAS) and distributed energy systems with turbine inlet air cooling can be considered by use of hourly weather data for various cities across the nation. The utility of such codes is illustrated with cost savings resulting from use of a DOAS system. For the purpose of such analysis, a commercial building with realistic electrical, ventilation, lighting and air conditioning loads are assumed. The code estimates the hourly building loads, energy requirements and annual energy cost for any of the above mentioned technologies as selected by the user employing the existing energy costs for the cities evaluated. The code is windows based developed in the Visual Basic language to facilitate easier interaction for the user. Students who used this code were extremely pleased to have the greater insights and significance of the technologies considered.

Keywords: DOAS, heat pump, air-conditioner, energy recovery device, building loads

Introduction

Software is employed by the practicing engineers. Although the results of using these The energy consumption of a commercial building is primarily due to operation of heating, ventilation and air-conditioning equipment (HVAC) and that consumed by the secondary equipment such as lights, other electrical equipment, computers, copiers, Xerox machines, etc,. The energy costs is directly related to the energy consumption of the HVAC equipment, which depends upon several factors such as the type of equipment, provisions for energy recovery or improving the efficiency of energy utilization, the weather conditions existing on the outside of the building, and finally on the unit costs of energy which may vary with time during the day or from season to season.

The energy efficiency of most of the HVAC equipment vary with loads that depends upon the type of building envelope structure and weather conditions outside for a given hour. The thermal conditions of conditioned air stream coming out of the HVAC equipment are adjusted to meet the energy demand of the building for the given hour. The basic mechanism of heat transfer through the building envelope is truly transient in nature that is affected by the thermal storage capacity of the building materials utilized in the construction of roof, walls and windows. In order to simplify the calculations involved in sizing the HVAC components, the industry [1] employed the cooling load temperature difference (CLTD) method. This method is not helpful in evaluating the energy consumption of the equipment over a period of season or year. Alternate method such as BIN method is used but it will not yield accurate results. Inaccuracies in these methods arise due to lack of using the more detailed hourly weather data as well as poor accounting of thermal storage effects, as is done in some of the available commercial software are very accurate, but it requires considerable amount of experience on the part of the users of such programs. In a three hour first course in HVAC field, it is very difficult to cover such materials in one semester. The author has developed a few user friendly computer programs [2-7] to illustrate the effect of thermal storage as well as the more accurate hourly weather data for a given city. Such programs are very useful in providing the greater insights into the design problems in addition to understanding the significance of the technologies considered as well as in obtaining the more accurate results using realistic data.

---

[1] Professor of Mechanical Engineering, College of Engineering and Computer Science,
The University of Tennessee, Chattanooga, 615 McCallie Ave, Chattanooga, TN 37403

Theory

The estimation of the annual or monthly energy requirements of a building to maintain it at comfortable conditions require the knowledge of building loads for each hour of the year. Hourly loads are estimated typically, since the weather data is available for most of the cities on an hourly basis. The weather data collected for each city reflects the historical average for a period of 20 to 30 years.

Transfer function method takes into account the thermal storage effect of the solar energy, occupants, lights and equipment. Heating load is the rate of heat transfer typically per hour from the building to the outside, while cooling load refers to that from outside to the inside of the building. The types of loads are further classified into sensible and latent loads. The sensible loads occur due to difference in temperatures that consists of convective and radiation components, while the latent loads occur due to difference in humidity levels which are instantaneous. The estimation of radiation component of sensible load is complicated due to thermal storage effect. For instance, the heating or cooling load Q can be considered as the response of a building or room to the effects that the temperature of the space (T_i), the temperature of the environment outside (T_o), or adjoining spaces, and the solar heat transfer rate ($\dot{Q}_{sol}$), etc. have on that building or room. The temperature of the space, the temperature of the environment outside, or adjoining spaces, the solar heat transfer rate, heat energy from occupants, equipment, and lighting $\left(T_i, T_o, \dot{Q}_{sol}, etc\right)$ are known as the driving terms. The Transfer Function Method (TFM) calculates the response of a system by making three assumptions (i) all functions of time are represented as sum of series of values at regular time steps (hourly in this case). (ii) the response of a system is a linear function of the driving terms and of the state of the system. (iii) the response at time t can depend only on the past, not on the future.

A linear series relationship between the response term y_t and the driving term u_t can be represented as

$$y_t = -\left(a_1\, y_{t-1\Delta t} + a_2\, y_{t-2\Delta t} + \ldots + a_n\, y_{t-n\Delta t}\right) + \left(b_0\, u_t + b_1\, u_{t-1\Delta t} + b_2\, u_{t-2\Delta t} + \ldots + b_m\, u_{t-m\Delta t}\right) \quad (1)$$

where the *time step* Δt = 1 hour and a_1 *to* a_n *and* b_0 *to* b_m are coefficients that characterize the system that take into account the thermal inertia or storage effects of the materials and are called transfer function coefficients.

Equation (1) can be generalized to take into account several driving terms. The solar radiation and outside air temperatures are two driving terms that affect the heat gain into the building. A term called sol-air temperature (T_e) is defined to take into effect both the outside air temperature T_o and solar radiation I_t and is expressed as:

$$T_e = T_o + \frac{\alpha I_t}{h_o} - \frac{\varepsilon \Delta R}{h_o} \qquad (2)$$

α = absorptance of surface for solar radiation

I_t = total incident solar load obtained from actual measured values

$\frac{\varepsilon \Delta R}{h_o}$ = long wave radiation factor = -7°F for horizontal surfaces; 0°F for vertical surfaces

$\frac{\alpha}{h_o}$ = surface color factor = 0.026 for light colors and 0.052 for dark colors

The solar incident solar load as given by I_t can be obtained from the measured solar data at a given location in terms of various terms involving solar incident angle (θ_i), declination angle (δ), solar azimuth angle ($\acute{\phi}_s$), the solar zenith angle (θ_s), the surface azimuth angle ($\acute{\phi}_p$), and surface tilt angle (θ_p), surface latitude angle (λ) and finally the hour angle.

The reader may refer to the reference [8] to see the orientation of these angles.

The declination angle (δ) can be given as,

$$Sin\,\delta \quad = \quad -Sin\,23.45^0 \; Cos\,\frac{360^0\,(n+10)}{365.25} \qquad (3)$$

where, n is the day of the year with January 1 being n = 1.

The hour angle (ω) can be estimated in terms of solar time (t_{sol}) from,

$$\omega \quad = \quad \frac{360^0\;(t_{sol}-12\,h)}{24\,h} \qquad (4)$$

The solar zenith angle (θ_s) can be estimated from,

$$Cos\,\theta_s \;=\; Cos\,\lambda\;\; Cos\,\delta\;\; Cos\,\omega \;+\; Sin\,\lambda\;\; Sin\,\delta \qquad (5)$$

Now, the solar azimuth angle (ϕ_s) in terms of solar zenith angle (θ_s) as follows

$$\phi_s \quad = \quad \frac{Cos\,\delta\;\,Sin\,\omega}{Sin\,\theta_s} \qquad (6)$$

Finally, the solar incident angle (θ_i) is given by

$$Cos\,\theta_i \;=\; Sin\,\theta_s\,Sin\,\theta_p\,Cos\,(\phi_s - \phi_p) \;+\; Cos\,\theta_s\;\,Cos\,\theta_p \qquad (7)$$

where, ϕ_s is the solar azimuth angle given by Equation (6) while the surface azimuth angle ϕ_p is the angle made by the surface normal with south direction. The tilt angle of the surface θ_p is the angle of inclination of the surface with local horizontal surface.

Now the total incident solar load I_t is the sum of

(i) the solar direct radiation (I_{dir}) incident normal to the surface

(ii) the solar diffuse radiation (I_{dif}), the diffuse radiation is the radiation scattered from the surroundings and the dust particles present in the atmosphere.

(iii) the solar radiation reflected from the ground

$$I_t \quad = \quad I_{dir}\,Cos\,\theta_i + I_{dif}\,\frac{1+Cos\,\theta_p}{2} + I_{glo,hor}\,\rho_g\,\frac{1-Cos\,\theta_p}{2} \qquad (8)$$

where, $I_{glo,hor}$ is the global horizontal radiation incident on the horizontal surface.

The weather stations in many major cities record hourly data consisting of I_{dir}, I_{dif}, $I_{glo,hor}$, the ambient air temperature, the dew point temperature, the relative humidity, wind speed and direction, cloud cover factor and many other data. The research scientists obtained the average of 25 to 30 years of such data and call this data as the typical meteorological year (TMY) for that city. Use of such data provides a more accurate solar loads.

Conductive Heat Gain

The conductive heat gain (or loss), $\dot{Q}_{cond,t}$ at time t through the roof and walls can now be expressed as:

$$\dot{Q}_{cond,t} \quad = \quad -\sum_{n\geq 1} d_n\, \dot{Q}_{cond,t-n\Delta t} + A\left(\sum_{n\geq 0} b_n\, T_{os,t-n\Delta t} - T_i \sum_{n\geq 0} c_n\right) \qquad (9)$$

where A = area of the roof or wall, can be in units of m^2 or ft^2.

Δt = time step, which is 1 hour.

$T_{os,t-n\Delta t}$ = sol-air temperature of outside surface at time t

b_n, c_n, d_n are the coefficients of conduction transfer function and the values of these coefficients for different types of roofs or walls commonly employed in building industry are listed in reference [1]. The computer code developed by the author has these values in its memory and provides the appropriate values for the materials chosen by the user of the program.

t = hour for which calculation was made
Δt = time interval (1 hr)
n = number of hours for which and values are significant, typically four
A = area of element under analysis

The heat gain into the building does not translate into the building load instantaneously.

Cooling Load

The total cooling load, Q_t is given as

$$Q_t \quad = \quad Q_{rf} + Q_{sc} \qquad (10)$$

$$Q_{rf} \quad = \quad \sum_{i=1}\left(v_o\, q_{t,i} + v_i\, q_{t,i-\Delta t} + v_2\, q_{t,i-2\Delta t} + \ldots\right) - \left(w_1\, Q_{t-\Delta t} + w_2\, Q_{t-2\Delta t} + \ldots\right) \qquad (11)$$

$$Q_{sc} \quad = \quad \sum_{j=1}\left(q_{c,j}\right) \qquad (12)$$

where: Q_{rf} = sensible cooling load from heat gain elements having radiant components.

v and w = room transfer function coefficients, listed in [1] selected per element type, circulation rate, mass, and/or fixture type.

q_t = each of i heat gain elements having a radiant component; selected appropriate fractions for processing.

Δt = time interval (1 hr)

Q_{sc} = sensible cooling load from heat gain elements having only convective components.

q_c = each of j heat gain factors having only convective component

$$Latent\ Q \quad = \quad \sum_{n=1}\left(q_{c,n}\right) \qquad (13)$$

q_c = each of n latent heat gain elements.

The above equations are presented here to provide the glimpse of what is involved in the calculations of heat gains and cooling loads. The code iterates for about six times for each hour of the total 8760 hours in a year and it further iterates the Equation (11) for another six times for each hour of the total 8760 hours in a

Loads due to Ventilation and Infiltration Air

The building loads due to ventilation and infiltration are instantaneous, they don't have the radiation component in them and they are given as

$$q_{sensible} = 1.10\,Q\left(T_o - T_i\right) \qquad (14)$$

$$q_{latent} = 4840\,Q\left(W_o - W_i\right) \qquad (15)$$

$$q_{total} = 4.5\,Q\left(h_o - h_i\right) \qquad (16)$$

where: Q = sum of ventilation airflow – as per ASHRAE Standard 62 and infiltration cfm
$T_{o,}$ $T_{i,}$ = outside, inside air temperatures, °F
$W_{o,}$ $W_{i,}$ = outside, inside air humidity ratio, lb (water) / lbm d.a
$h_{o,}$ $h_{i,}$ = outside, inside air enthalpy, Btu/lbm d.a

$$Latent\ Q = \sum_{n=1}\left(q_{c,n}\right) \qquad (17)$$

q_c = each of n latent heat gain elements.

The space heat load will be the summation of the loads from each of the above. The maximum values of hourly heat gain and loss represent the building peak cooling and heating loads, respectively.

Dedicated Outside Air System (DOAS)

Standard HVAC system takes a mixture of outdoor air and return air, conditions it, and then returns it to the building space at the desired temperature and humidity levels. In order to maintain a space at specified temperature and humidity levels, the cooling system, or chiller, has to cool down the mixture of outside and return air to a much lower temperature well below the dew point of the mixed air to remove or dehumidify the excess moisture to the desired levels. To accomplish this, the compressor or air-conditioner consumes greater power to maintain a low evaporating or coil temperature. Typically, near the peak cooling loads, the air temperature is reduced to a very low value after the required dehumidification takes place within the cooling coils. The reheating of the dehumidified air is frequently done, so that the temperature of the supply air to the space is at a comfortable level. In certain applications and geographical regions, the cost of reheating can be significant. Energy costs associated with air-conditioning the outside air can be increased further if the building occupancy is not uniform in each zone of the building.

A Dedicated Outdoor Air System (DOAS) is a constant air volume system that is designed to deliver the required amount of air to each space or zone of the building evenly. This system complies 100% with ASHRAE Standard 62. DOAS unit requires about 20 to 30% less outdoor air and also can provide 100% of the space's latent load, when needed. With the DOAS unit, the sensible cooling and heating is performed separately. The sensible conditioning is decoupled from the latent conditioning of the air. The sensible conditioning of the air can be performed effectively and economically by circulating the chilled water through panels or coils imbedded in the ceiling and walls of the building. The radiant heating or cooling from the panels or wall coils can be supplemented with the conditioned fresh air from the DOAS unit to the space or zone and is delivered through a smaller duct system from the DOAS.

The energy cost savings resulting from use of the DOAS unit is attributed primarily to the following factors:

- It uses the optimal (minimum) amount of ventilation air and also complies 100% of ASHRAE Standard 62.

- The pumping power of conditioned air through the ducts is considerably reduced due to lower air flow rates.
- The latent loads are completely handled by the DOAS unit so that the sensible loads can be handled by the sensible chiller units operating at higher evaporating temperatures, and thus at peak efficiency levels.
- Radiant panel cooling and heating of the buildings to meet the sensible loads can be very cost effective due to reductions in capital costs associated with larger ducts, larger fans, and low pumping cost of water as compared to that of air.
- Additional costs savings are possible due to the use of enthalpy and heat wheels.

The purpose of this investigation is to assess the potential energy cost savings resulting from use of integrated DOAS unit with CRCP and energy recovery wheels at various climatically different regions within the U.S

Description of DOAS System

The DOAS unit consists of a separate heat pump unit in parallel with a sensible chiller as shown in Figure 1. In addition to DOAS unit, the Figure 1 also shows an enthalpy wheel and a heat wheel. The enthalpy wheel allows the transfer of both latent (moisture) and sensible (heat energy) between the two air streams flowing through it, while the heat wheel allows only the transfer of heat energy due to temperature difference between the streams. The heat wheel placed downstream of the DOAS unit provides the free reheat energy to the dehumidified air leaving the DOAS system. The enthalpy wheel will be very effective during the cooling season by cooling and dehumidifying the incoming outside fresh air with the outgoing exhausted air. In certain applications, especially with greater latent loads caused due to large number of occupants, the enthalpy wheel will be counter productive in winter by transferring the moisture to the incoming fresh air and thus increasing the latent loads. The dashed lines indicates the typical winter operation, while the solid lines show the air streams during the cooling season. The dotted lines indicate the hot or chilled water flow to the CRCP of the building space.

A Typical Operation in Cooling Season

The exhaust air from the building space at state 6 enters the heat wheel and heats up the dehumidified air leaving the DOAS system at state 4. The exhaust air leaving the heat wheel at state 7 splits into two streams. The stream at state 9 has the same mass flow rate as that of the incoming fresh air at state 1 and enters the enthalpy wheel to partially dehumidify and cool the outside air, while the flow rate at state 8 is mixed with the fresh air leaving the enthalpy wheel to provide a larger required flow rate to the DOAS system. This is required in the event of having a larger latent load. The minimum flow rate required by the heat pump of the DOAS system is frequently larger than the minimum required to meet the ASHRAE Standard-62. The added mass flow rate of the DOAS unit above that required for proper ventilation only increases the sensible capacity of the DOAS system thus reducing further the loads on the sensible chiller.

A Typical Operation in Heating Season

If latent loads are zero or negligible:
The outside fresh air picks up the heat energy from the exhaust air from the building space in the heat wheel, after mixing with the added re-circulated return air it enters the DOAS system for additional sensible heating.

If latent loads are negative (Humidification Required):
The outside fresh air can pass through the enthalpy wheel and after mixing with the added re-circulated return air it enters the DOAS unit for additional sensible heating, if required.

If latent loads are positive in:

The fresh outside air will bypass enthalpy wheel and passes through the DOAS system for dehumidification, this situation adds to the sensible heating unit, but can recover a certain amount by passing through the heat wheel.

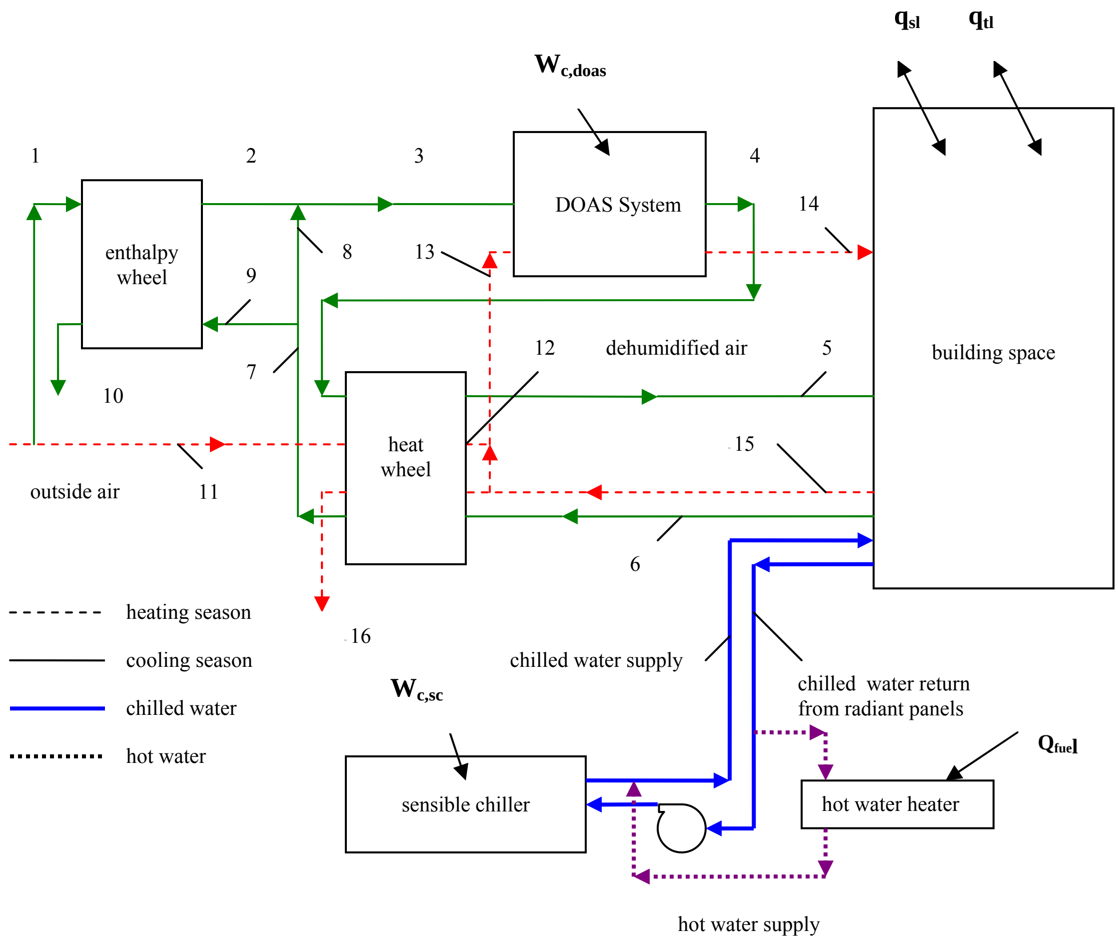

Figure 1 Schematic of a Typical DOAS System

Estimating the energy cost savings resulting from use of DOAS unit boils down to determining the operating cost through the power supplied to the DOAS unit ($W_{c,doas}$) and the sensible chiller ($W_{c,sc}$) unit along with the pumping costs of the chilled water and the air-duct systems as shown in the Figure 1. To determine the $W_{c,doas}$ and the $W_{c,sc}$ the following equations are gathered first based on fundamental laws of the energy balance and heat exchanger analysis. The energy balance on the enthalpy wheel gives,

$$m_o\, c_p\, (t_1 - t_2) = \varepsilon_s\, m_o\, c_p\, (t_1 - t_9)$$

or,
$$t_2 = t_1 - \varepsilon_s\,(t_1 - t_9) \qquad (18)$$

$$m_o\, h_{fg}\, (\omega_1 - \omega_2) = \varepsilon_l\, m_o\, h_{fg}\, (\omega_1 - \omega_9)$$

or
$$\omega_2 = \omega_1 - \varepsilon_l\,(\omega_1 - \omega_9) \qquad (19)$$

where ε_s = the sensible effectiveness of the enthalpy wheel.
ε_l = the latent effectiveness of the enthalpy wheel.
m_o = mass flow rate of the outside air assumed to be minimum of the two streams.
c_p = the specific heat at constant pressure of the air streams assumed to be

constant for both streams.

t = the temperature at the state indicated by the subscript.

ω = the humidity ratio at state indicated by the subscript.

h_{fg} = enthalpy of vaporization assumed to be constant for both streams.

The temperature and humidity ratio of the mixture of the two moist air streams at state 8 and state 2 is given by,

$$t_3 = \frac{m_{ad}\, c_p\, t_7 + m_o\, c_p\, t_2}{(m_o + m_{ad})c_p} \qquad (20)$$

$$\omega_3 = \frac{m_{ad}\, h_{fg}\, \omega_7 + m_o\, h_{fg}\, \omega_2}{(m_o + m_{ad})h_{fg}} \qquad (21)$$

where m_{ad} = the mass flow rate of the added re-circulated air through the DOAS system.

It may be noted that the temperature at states 7, 8 and 9 are the same as that at state 7.
The performance of the DOAS unit or any air-to-air heat pump or air-conditioner depends primarily on the air flow rate through the evaporator, the outside air temperature and the wet-bulb temperature of the air entering the evaporator coil. For a given air flow rate through the heat pump, the sensible (q_s) and total cooling capacities (q_t) are primarily dependent upon the out side air temperatures (t_o) and the wet-bulb temperature (t_{w3}) of the air entering the evaporator coil as shown below.

$$q_s = f_1(t_{w3},\ t_o)$$

$$q_t = f_2(t_{w3},\ t_o)$$

and the latent capacity, $q_l = q_t - q_s$

where, f_1 and f_2 are empirical relations that can be developed by curve fitting the data from the manufacturer's catalog for the heat pump employed in the DOAS unit as shown in the Figure 1. The following equations are developed for a more efficient heat pump recently introduced into the market:

$$q_t = 1000\left\{\frac{121.9558 - 0.191343 t_o + 0.00240729 t_o^2 - 8.16E-06 t_o^3 - 3.44073573 t_{wi3} + 0.023492256 t_{wi3}^2}{1 + 0.000114722 t_o - 0.03059504 t_{wi3} + 0.000218688 t_{wi3}^2}\right\} \qquad (22)$$

$$q_s = 1000\left\{\frac{1093.056962 - 0.00097147 t_o + 7.37374E-05 t_o^2 - 510.815689\ln(t_{wi3}) + 59.44701593(\ln(t_{wi3}))^2}{1 + 6.32442E-05\, t_o - 0.2490781\ln(t_{wi3})}\right\} \qquad (23)$$

$$W_{c,c} = \left\{\frac{6.924437819 + 0.009056\, t_o - 0.19511196\, t_{wi3} + 0.001479819\, t_{wi3}^2}{1 - 0.00064041 t_o - 0.0234025 t_{wi3} + 0.000172891 t_{wi3}^2}\right\} \qquad (24)$$

where t_{wi3} = the wet-bulb temperature of the air entering the DOAS unit, and
$W_{c,c}$ = power supplied to the compressor, while in cooling mode.

The above three equations give the computational relations for the total and sensible cooling capacities and the Equation (24) is for the required compressor power while the heat pump is working in a cooling mode. The heating capacity and the compressor power to the heat pump in the heating mode is dependent upon the dry-bulb temperature of the outside air and that of the return air to the condenser for a given air flow rate, as shown by the following two equations.

$$q_{heat} = 1000\left\{59.5051 - 0.1663157\, t_r + 0.7692\, t_o - 0.0068521\, t_o^2 + 0.00016263\, t_0^3 + 1.45926E-05\, t_o^4 - 2.1E-07\, t_o^5\right\} \qquad (25)$$

$$W_{c,h} = \left\{ \frac{6.452007887 - 0.04451453\ t_r - 0.00899686 t_o - 0.00025825 t_o^2 + 1.44139E-05 t_o^3}{1 - 0.00830784\ t_r - 0.00232997 t_o - 1.4804E-05\ t_o^2 + 1.2214E-06\ t_o^3} \right\} \quad (26)$$

where t_r = the dry-bulb temperature of the return air to the condenser, and
$W_{c,h}$ = power supplied to the compressor, while in heating mode.
The energy balance across the DOAS system gives,

$$t_4 = t_3 - \frac{q_s}{m_d\ c_p} \quad (27)$$

$$\omega_4 = \omega_3 - \frac{q_l}{m_d\ h_{fg}} \quad (28)$$

where m_d = mass flow rate of the moist air flowing through the DOAS system.

Since the Equations (24), (25) and (26) involve the wet-bulb temperature of the moist air at the state 3, the following Equation [29] can be employed to determine iteratively the wet-bulb temperature at state 3, provided the dry-bulb temperature and humidity ratio at state 3 are known.

$$\varpi_3 = \frac{(1093 - 0.556 t_{w3})\ \omega_{s3} - 0.24\ (t_3 - t_{w3})}{1093 + 0.444\ t_3 - t_{w3}} \quad (29)$$

The energy balance on the heat wheel gives,

$$m_d\ c_p\ (t_5 - t_4) = m_d\ c_p\ (t_6 - t_7)$$

or

$$t_4 = t_5 - (t_6 - t_7) \quad (30)$$

$$t_7 = t_6 + \varepsilon_{2s}\ (t_6 - t_4) \quad (31)$$

where ε_{2s} = the effectiveness of the sensible heat wheel.

The above equations from Equation 18 to 31 involve 14 unknowns namely $t_2, t_3, t_{w3}, t_4, t_5, t_7, t_9, \omega_2, \omega_3, \omega_{s3}, \omega_4, q_s, W_{c,doas}, W_{c,sh}$. All of the above 14 unknowns are time dependent and can be solved by the following methodology.

Computational Methodology

It is assumed that the system is operating with minimum duct losses or gains and the mass flow rates are such that balance flow conditions exist in the enthalpy and heat wheels during their operation. The sensible and latent effectiveness are assumed to be constant at 0.7 and 0.65, respectively, under all conditions. The energy costs, electric consumption and demand costs are assumed to be constant over the year. *The DOAS unit is employed in a medium sized commercial building of floor area 4000 ft^2 (371.6 m^2) with 66 people working during the day with 2 W/ft^2 (21.53 W/m^2) of lighting load and a total of 25 kW of equipment and the normal loads associated with hot water consumption and an infiltration rate of 0.1 ACH.* The loads during the non-office hours, weekends and holidays are set at minimum values with a realistic thermostat setbacks. Using the hourly weather data of the city, the hourly (q_s, q_t, q_l, $W_{c,doas}$, $W_{c,sc}$ *and* Q_{fuel}) and peak building and equipment loads are estimated by use of TABLET computer code developed based on transfer function method. The energy costs associated with the HVAC system consists of the cost of power supplied to the DOAS unit ($W_{c,doas}$), the cost of power supplied to the sensible chiller unit ($W_{c,sc}$), the fuel cost associated with heating unit (Q_{fuel}) and the cost associated with pumping the fluids as shown in Figure 1. The energy costs are evaluated following the procedure described for a typical cooling period as follows:

1. The latent load of the building is obtained to check, if latent cooling is required, that the latent rate of heat extraction is the same as latent load of the building and is equated to the latent capacity of the DOAS unit. The latent heat capacity of the unit is obtained from Equations (23) and (24), which are functions of the outside air temperature, t_o, and the wet-bulb temperature, t_{w3}, of the entering air. Using the Newton-Raphson technique, the wet-bulb temperature, t_{w3}, is obtained

iteratively. Once the wet-bulb temperature is computed, one can determine the sensible capacity, q_s, and compressor power $W_{c,c}$ of the DOAS unit from Equations (23) and (24).

2. Since the heat wheel does not transfer any moisture between the streams, and humidity ratios at states 6, 7, 8 and 9 are the same as that of the building space also knowing the latent effectiveness of the enthalpy wheel, one can determine the humidity ratio (ω_2) from the Equation (19) and, ω_3, from the Equation (21).
3. Knowing the humidity ratio ω_3 and the wet-bulb temperature, t_{w3}, at state, 3, one can find the actual dry-bulb temperature, t_3, of the moist air entering the DOAS unit by using the Equation (29).
4. The exit temperature, t_4, of the moist air from the DOAS unit can be obtained from Equation (27) as the sensible capacity of the DOAS unit is estimated from the Equation (23) from the knowledge of t_{w3} and t_o.
5. Knowing the sensible effectiveness, ε_{2s}, of the heat wheel, and the temperature t_4 and t_6 one can determine the exit temperature, t_7, of the exhaust air from Equation (31).
6. Recognize that temperatures at state 7, 8 and 9 are the same, i.e $t_7 = t_8 = t_9$.
7. Finally, the supply air temperature, t_5, can be estimated from Equation (13).
8. One can determine the corrected sensible load of the building by adjusting the sensible cooling provided by the DOAS unit.
9. The power required by the sensible chiller is obtained from the corrected sensible load and characteristics of the sensible chiller. For simplicity, it is assumed that a centrifugal chiller operating at a higher evaporating temperature can have an average, COP, of 7, and hence the power supplied to the sensible chiller, $W_{c,sc}$, is obtained by dividing the corrected sensible load with the average COP.
10. In heating season, the heating load is met by circulating the hot water produced in the hot water boiler as shown by the dotted lines and the heating capacity, q_{heat}, and the compressor power supplied to the heat pump of the DOAS unit is obtained from the outside air temperature, t_o, and the return air temperature, t_r, of the space from Equations (25) and (26), respectively. The fuel supplied to the hot water heater is obtained by dividing the heating load of the hour with the boiler efficiency. The fuel cost would be the product of the fuel supply and fuel cost for the hour.

Results

The results obtained from detailed computer simulations for several geographical locations are listed in the Table 1. These cost savings are obtained from detailed computer simulations using the same cost structure for the fuel and electric demand the energy cost and consumption charges in order to assess the influence of different climatic conditions of the various regions. It may be noted that savings are related only to operating cost and takes no account of potential capital cost savings or additional charges. For instance, for the city of Houston, the peak cooling load is reduced by more than 4 tons of capacity, which may amount to a reduction of nearly $ 4000 in the capital cost of the chillers. There exist a reduction of the capital cost of duct construction and fan sizes due to reduced air flow rates. It has been reported that the ceiling space of an integrated DOAS unit can reduce by nearly 50 percent due to smaller size ducts. The energy cost savings resulting from use of various technologies and for various cities are shown in Figure 2-7.

Conclusions and Recommendations

The results of shown in Table 1 show that the integrated DOAS unit with CRCP and the energy and heat recovery wheels has a significant potential of reducing the capital as well as operating costs as compared to conventional systems. The size of the building chosen for this investigation is relatively small as compared to the average size of a commercial building in the U.S. Based on the magnitude of the energy cost savings as a result of the application of the DOAS units, application of DOAS unit has a potential to be a significant cost-effective technology in the near future of HVAC industry. The Figures 2-7 show the significance of each technology in providing energy cost savings. Students indicated in a survey that the program is very simple and user friendly and provided a more realistic analysis tool to evaluate each technology, and allows to get greater insights into each of the technology considered and the factors that influence the energy cost savings. It is recommended that the materials presented in this paper should be provided to the students along with such computer codes in order to understand the quantitative impact of each energy savings technologies and also to appreciate the importance of using more detailed methods such as those illustrated in this paper. The materials presented in this paper helps students to get familiar with the detailed approach of estimating the building loads and also to the various current energy efficient or recovery technologies.

Table 1 Performance Characteristics of DOAS Unit at Various Locations

City	Eqpm Load kW	Number of Occupants	Peak H. Loads Tons (kW) Tot heating/ latent	Peak C. Loads tons[a](kW) Tot cooling / latent	Max Elec. Demand (kW)	E[b]. Demand Cost $/yr	Energy Cost Conv $/yr	DOAS Cost $/yr	% diff [c]
Atlanta	25	66	12.6 / 3.5 (44.3) / (12.3)	18.66 / 4.3 (65.6) / (15.1)	61.8	710	12,905	-	-
Atlanta (DOAS)	25	66	12.6 / 3.5 (44.3) / (12.3)	15.4 / 1.8 (54.2) / (6.3)	36.7	0	-	9,098	41.8
Chattanooga	25	66	13.28 / 3.6 (46.7) / (12.7)	19 / 4.8 (66.8) / (16.9)	61.8	660	12,747	-	-
Chattanooga (DOAS)	25	66	13.28 / 3.6 (46.7) / (12.7)	15.3 / 1.8 (53.8) / (6.3)	36.6	0	-	9,065	40.6
Chicago	25	66	16.07 / 3.8 (56.5) / (13.4)	18.3 / 4.3 (64.3) / (15.1)	57.7	306	11,047	-	-
Chicago (DOAS)	25	66	16.07 / 3.8 (56.5) / (13.4)	15.1 / 1.9 (53.1) / (6.9)	36.6	0	-	8,926	23.8
Houston	25	66	12.26 / 3.5 (43.1) / (12.3)	20.1 / 6 (70.7) / ((21.1)	62	615	12,322	-	-
Houston (DOAS)	25	66	12.26 / 3.5 (43.1) / (12.3)	15.8 / 2.3 (55.6) / (8.1)	36.9	0	-	9,262	33
L . A	25	66	7.8 / 2.9 (27.4) / (10.2)	15.47 / 2.6 (54.4) / (9.1)	61.4	646	12,763	-	-
L . A (DOAS)	25	66	7.8 / 2.9 (27.4) / (10.2)	14.1/ 1.5 (49.6) / (5.3)	36.3	0	-	9,123	39.9
New York	25	66	14 / 3.8 (49.2) / (13.4)	17.7 / 3.0 (62.2) / (10.6)	44.4	0	11,312	-	-
New York (DOAS)	25	66	14 / 3.8 (49.2) / (13.4)	15.06 / 1.6 (53) / (5.6)	36.8	0	-	8,949	26.4

a- roof: 3"in,w/4"l.conc deck sus ceiling, walls: 4"l.w.conc block, w/1" in face brick, b- first 50 kW demand free, but penalizes by $ 9.89 /kW and energy cost of $ 0.06/kWh c- based on operating cost only

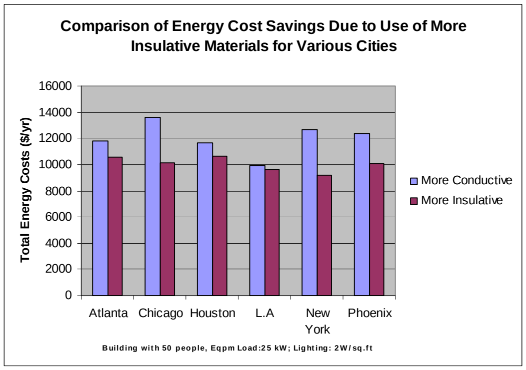

Figure 2. Energy Cost Savings due to More Insulative Materials

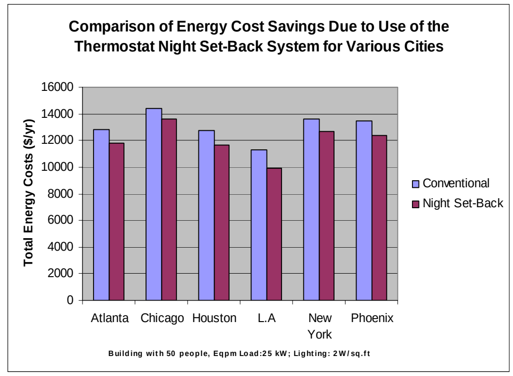

Figure 3. Energy Cost Savings due to Night Set-Back

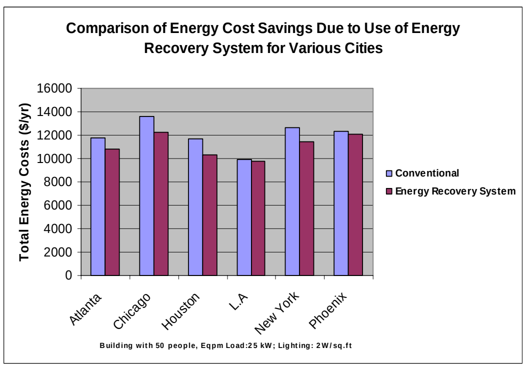

Figure 4. Energy Cost Savings due to Energy Recovery Systems

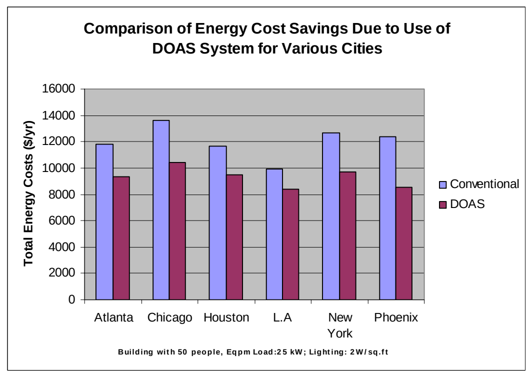

Figure 5. Energy Cost Savings due to Use of DOAS Systems

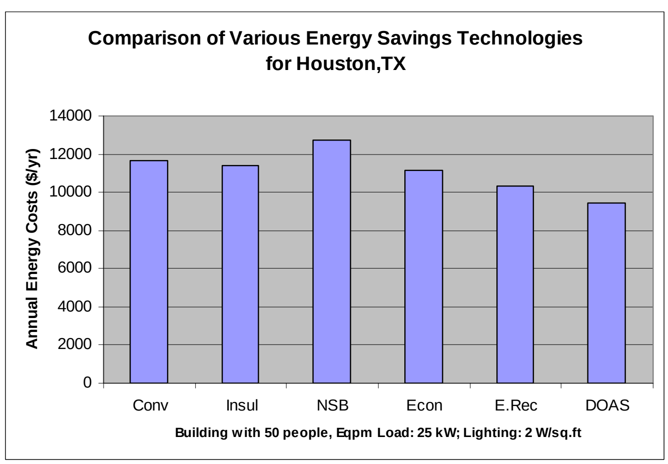

Figure 6. Energy Cost Savings due to Various Technologies

Nomenclature

a_n, b_n, and c_n are coefficients of conduction transfer function

c_p = the specific heat of moist air (J/kg.^{0}C).

f_1 = a mathematical function that relates the sensible capacity to entering wet-bulb temperature and outside air temperature.

f_2 = a mathematical function that relates the total capacity to entering wet-bulb temperature and outside air temperature.

h_o = convective heat transfer coefficient for outside air, W/m^2.K

h_{fg} = enthalpy of vaporization (J/kg).

I_{dif} = diffuse solar radiation, W/m^2

I_{dir} = direct solar radiation, W/m^2

I_t = total solar radiation, W/m^2

$I_{glo,hor}$ = global solar radiation on a horizontal plane, W/m^2

m_{ad} = mass flow rate of additional re-circulated moist air through DOAS system (kg/s).

m_d = mass flow rate of moist air through the DOAS system (kg/s).

m_o = mass flow rate of the moist air from the outside (kg/s).

q_{heat} = heat capacity output of the heat pump (kW).

q_l = latent cooling capacity of the heat pump (kW).

q_s = sensible cooling capacity of the heat pump (kW).

q_t = total cooling capacity of the heat pump (kW).

Q = volumetric flow rate of air, L/s or cfm

Q_{rf} = sensible cooling load from radiant portion (W)

Q_{sc} = sensible cooling load from convective portion (W)

T_i, t_i = moist air temperature at state, i (^{0}C).

T_o, t_o = outside air temperature (^{0}C).

T_e = sol-air temperature (^{0}C).

t_n = dry bulb temperature at state, n (^{0}C)

t_{w3} = wet-bulb temperature of moist air at state, 3 (^{0}C).

$W_{c,c}$ = power supplied to the compressor of heat pump in cooling mode (kW).

$W_{c,h}$ = power supplied to the compressor of heat pump in heating mode (kW).

$W_{c,sc}$ = power supplied to the compressor of the sensible chiller (kW).

Greek Letters

α = absorptance of solar energy
δ = declination angle
λ = latitude angle
ε = emittance of outside surface
ω = hour angle
θ_i = solar incident angle
θ_p = surface tilt angle
θ_s = solar zenith angle
ρ_g = reflectance of ground surface
Φ_s = solar azimuth angle
ε_s = sensible effectiveness of heat or energy wheel.
ε_l = latent effectiveness of energy wheel.
ε_{s2} = sensible effectiveness of secondary heat wheel.
ε_t = total effectiveness of energy wheel.
ω_i = humidity ratio of a moist air at state , i
ω_{si} = humidity ratio of a saturated moist air at state , i

References

1. " ASHRAE Handbook of Fundamentals" American Society of Heating, Refrigeration and Air-conditioning Engineers, Atlanta, GA 2001
2. " Energy Cost Savings due to Use of Energy Recovery System with Dedicated Outside Air Systems (DOAS)", Seminar presented ASHRAE Annual Meeting in Kansas City, MO June 2003.
3. " Energy Cost Savings with Use of DOAS Systems in Various Cities in U.S" paper to be presented at ASME World Congress, Washington D.C, Nov 2003.
4. " A Multi-Purpose Thermal Design Project that Works " by Prakash R. Damshala and Robert Bailey, published in The International Journal of Mechanical Engineering Education, vol 30, No 2, April, 2002.
5. " Thermoeconomic Analysis of Cogeneration System for HVAC Applications in Commercial and Industrial Buildings "by Prakash R. Damshala and James Nathan Pugh M.S thesis completed in August 2004.
6. "A Computer Design Project for an Energy Recovery System" by Prakash R. Damshala, a paper presented at ASEE southeastern conference held at Charleston, SC in April 2001
7. " Numerical Analysis of Solar Storage (Trombe) Wall for Identifying Optimal Energy Recovery Conditions "by Prakash R. Damshala and Robert Bailey, a paper presented at ASEE southeastern conference, April 2000 at Blacksburg, Virginia.
8. " ASHRAE Handbook of Fundamentals, chapter 30." American Society of Heating, Refrigeration and Air-conditioning Engineers, Atlanta, GA 2001
9. " Heating and Cooling of Buildings" by J.F. Kreider., and A. Rabl, McGraw-Hill publication, 1994, New Yor, N.Y

Prakash R. Dhamshala

The author is a professor of mechanical engineering at the University of Tennessee, Chattanooga.He obtained his M.S from University of Miami, Fl and Ph.D from Georgia Tech, GA. He has been teaching thermal science courses in thermodynamics, heat transfer, thermal systems and air-conditioning and refrigeration course at this school for over 20 years. He is a member of ASHRAE, ASME and former member of ASEE. He is active in reviewing and publishing papers for ASEE, ASHRAE and ASME.

Appendix

These instruction illustrate the procedure to evaluate the energy cost savings potential of various advanced energy technologies. You interact with a single screen containing several windows and buttons and is divided into six zones A, B, C, D, E and F. The data such as outside air dry bulb temperature, relative humidity, ambient pressure, direct, diffuse and total solar radiation on horizontal and normal planes are inputted from the files containing in the code once user select a city. The user will provide data on building envelop and load profiles during the working, and off-days and holidays, and time of use energy costs.

The following procedure may be employed to obtain the simulation results.

1. Select the city in window A

2. Select the roof # 8 in area B

3. Select the wall #8 in area B for East, South, West and North walls

4. Select # 1 for floor, NE, SE,SW and NW walls

5. Selection the option "Off" for Economizer, Energy Recovery and DOAS systems in area C

6. Select the option "Off" for CHP system and press the button # 1to calculate the building peak loads

7. When you see the peak load values in windows in area D, then press the button # 2 to estimate the HVAC equipment loads

8. When you see the message "Done" in the window X, press the button # 3 to calculate the energy costs

9. Once you see the results in window E, you have a choice of printing the results by pressing the button # 4 or see the bar chart of monthly energy costs by pressing the button # 5 (You may print the chart if needed, provided the computer is connected to a printer) or see the temperature and building load profile on peak cooling day by pressing the button # 6 or see the pie chart of various building energy costs by pressing the button # 7

10. If you want to rerun the program with CHP unit, then select the option "on" for the CHP system, enter the total annual energy cost from window E of the conventional system without CHP into the window "Y" then press the clear button # 8 and finally press button # 1

11. Repeat steps 7 through 9

12. You can repeat the above procedure for various cities by selecting the city in window A or for various energy costs scenarios as shown in window area F