Performance Analysis of Dual-Fan, Dual-Duct Constant Volume Air-Handling Units

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ABSTRACT

Dual-fan, dual-duct air-handling units introduce outside air directly into the cooling duct and use two variable speed devices to independently maintain the static pressure of the hot and the cold air ducts. Analytical models have been developed to compare fan power and thermal energy consumption of dual-fan, dual-duct constant volume air-handling units with single-fan, dual-duct constant volume air-handling units. This study shows that the dual-fan, dual-duct system uses less fan power and less thermal energy during winter, and uses more thermal energy during summer. Thermal energy performance can be significantly improved if the thermal energy penalty can be decreased or eliminated.

INTRODUCTION

The single-fan, dual-duct (SFDD) constant air-handling unit has been used in most medical facilities as well as office buildings and library facilities since the 1940s. It is especially popular in hot and humid climates. The SFDD unit provides good room relative humidity control and good air circulation, and requires less maintenance. Over the decades, it is conceived that the SFDD constant air volume system consumes more fan energy and thermal energy than single duct systems. This dilemma, along with other operational problems, is created by the use of a constant speed fan. When a constant speed fan is used, both the minimum and the maximum static pressures are significantly higher than the design values under partial loads. Terminal box dampers are used as throttling devices to consume fan head. The excessive static pressure also causes a higher total airflow through the AHU and creates noise and vibration problems. Converting SFDD systems to DFDD systems allows both the hot and cold duct static pressures to be maintained at their set points [ASHRAE 2000, and Warden, 1996]. It significantly decreases fan power consumption and minimizes air leakage through terminal dampers.

From the late 70s and early 80s, the DFDD system was proposed and installed in many buildings [Gaggioli, 1978, Haines, 1981, Kettler, 1981 and Linford, 1981]. The performance of the DFDD system is generally acknowledged in practice [Schuler, 1996], and its potential energy savings is often determined using an hourly simulation program. However, there is no analytical model available for engineers to investigate the potential energy savings and optimize the control strategies. In this paper, analytical models are developed to investigate the advantage and disadvantage of the DFDD system. Graphic tools are designed for potential energy savings calculation.

MODELS

Thermal energy and fan power models are developed for both the SFDD system and the DFDD system. Potential energy savings are calculated by comparing the performance of the DFDD system and the SFDD system under the same ambient and load conditions.

Single-fan, dual-duct (SFDD) System

In the SFDD system, outside air mixes with return air. The heating coil maintains the required hot air temperature. The cooling coil maintains the cold deck temperature. A fan supplies both the hot and the cold air to terminal boxes. Dual-duct terminal boxes mix the hot and cold air to accommodate different space loads.

Mixed air temperature and enthalpy ($T_{m,b}$ and $h_{m,b}$) are functions of outside air temperature and enthalpy (T_{oa}, h_{oa}), room conditions (T_r, h_r), and the outside air intake ratio (β).

$$T_{m,b} = \beta \cdot T_{oa} + (1-\beta)T_r \tag{1}$$

$$h_{m,b} = \beta \cdot h_{oa} + (1-\beta)h_r \tag{2}$$

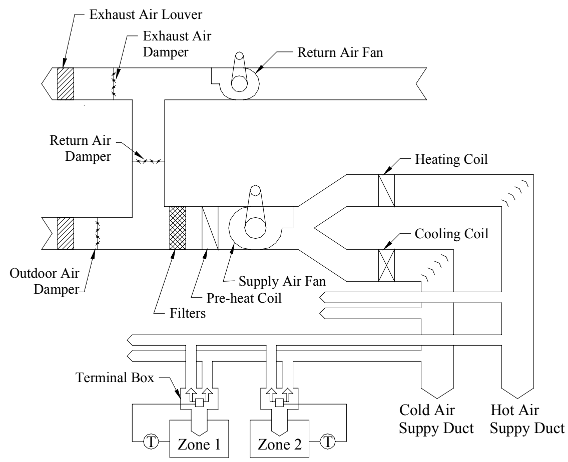


Figure 1. Schematic diagram of the SFDD system

Where: $\beta = \dot{m}_{oa} / \dot{m}_d$

The pre-heat coil prevents extremely cold air from entering the air-handling unit. If the mixed air temperature is too low, the pre-heat coil warms it up to the set point ($T_{p,b}$). In this study the value of the air leaving the pre-heat coil is the larger of either the cold deck set point ($T_{c,d}$) or the mixed air temperature ($T_{m,b}$). The value of the air leaving the cooling coil ($T_{c,b}$) is the smaller of either the cold deck set point or the mixed air temperature. The value of the air leaving the heating coil ($T_{h,b}$) is the larger of either the hot deck set point ($T_{h,d}$) or the mixed air temperature.

$$T_{p,b} = \max[T_{c,d}, T_{m,b}] \tag{3}$$

$$T_{c,b} = \min[T_{c,d}, T_{m,b}] \tag{4}$$

$$T_{h,b} = \max[T_{h,d}, T_{m,b}] \tag{5}$$

The heating and cooling energy consumptions of the SFDD system are:

$$E_{c,b} = \begin{cases} \dot{m}_{c,b} \cdot (h_{m,b} - h_{c,d}) & T_{m,b} > T_{c,b} \;\&\; T_{m,b,dew} > T_{c,d,dew} \\ \dot{m}_{c,b} \cdot c_p \cdot (T_{m,b} - T_{c,b}) & T_{m,b} > T_{c,b} \;\&\; T_{m,b,dew} \le T_{c,d,dew} \\ 0 & T_{m,b} \le T_{c,b} \end{cases} \tag{6}$$

$$E_{p,b} = \max[0, \dot{m}_d \cdot c_p \cdot (T_{c,b} - T_{m,b})] \tag{7}$$

$$E_{h,b} = \max[0, \dot{m}_{h,b} \cdot c_p \cdot (T_{h,b} - T_{p,b})] \tag{8}$$

The fan power can be calculated using the design airflow rate ($\dot{m}_d$), static pressure difference (H, fan head), air leakage rate (ε), characteristic conversion factor (C), and fan efficiency (η). The fan power is constant when the flow is assumed to be constant.

$$E_{f,b} = C \times \dot{m}_d \times H_d \times (1+\varepsilon)^3 / \eta \tag{9}$$

Dual-fan, dual-duct (DFDD) System

Figure 2 presents a schematic diagram of the DFDD system. Static pressure sensors are installed in both the hot and cold air ducts. A control system modulates the fan speed to maintain the set point in each duct. When the cold airflow rate is larger than the outside air intake rate, outside air is supplied through the cooling duct. When the cold airflow rate is smaller than the outside air intake rate, however, a portion of the outside air is supplied through the heating duct.

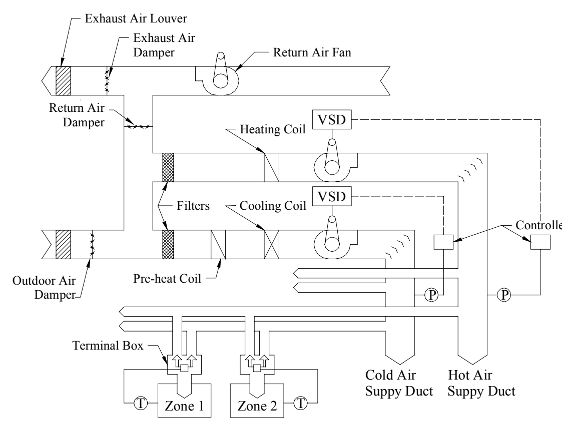


Figure 2. Schematic diagram of DFDD system

The mixed air temperatures ($T_{m,c}$, $T_{m,,h}$) are functions of the outside air temperature (T_{oa}), the room conditions (T_r), the outside air intake ratio (β) and the cold airflow ratio (γ_o).

$$T_{m,c} = \begin{cases} T_r - \dfrac{\beta}{\gamma_o}\left[T_r - T_{oa}\right] & \gamma_o > \beta \\ T_{oa} & \gamma_o \le \beta \end{cases} \tag{10}$$

$$T_{m,h} = \begin{cases} T_r & \gamma_o > \beta \\ \dfrac{1}{1-\gamma_o}\left[(\beta - \gamma_o) \cdot T_{oa} + (1-\beta) T_r\right] & \gamma_o \le \beta \end{cases} \tag{11}$$

Similarly,

$$h_{m,c} = \begin{cases} h_r - \dfrac{\beta}{\gamma_o}\left[h_r - h_{oa}\right] & \gamma_o > \beta \\ h_{oa} & \gamma_o \le \beta \end{cases} \tag{12}$$

Where: $\gamma_o = \dot{m}_{c,o} / \dot{m}_d$

The value of the air leaving pre-heat coil ($T_{p,o}$) is the larger of either the cold deck set point ($T_{c,d}$) or the mixed air temperature ($T_{m,c}$). The value of the air leaving cooling coil ($T_{c,o}$) is the smaller of either the cold deck set point or the mixed air temperature ($T_{m,c}$). The value of the air leaving heating coil ($T_{h,o}$) is the larger of either the hot deck set point ($T_{h,d}$) or the mixed air temperature ($T_{m,h}$).

$$T_{p,o} = \max[T_{c,d}, T_{m,c}] \tag{13}$$

$$T_{c,o} = \min[T_{c,d}, T_{m,c}] \tag{14}$$

$$T_{h,o} = \max[T_{h,d}, T_{m,h}] \tag{15}$$

For the same building load, the cold airflow of the DFDD system may differ from the cold airflow of the SFDD system since the hot air temperatures may vary. Using energy balance equations, the cold airflow ratio for the DFDD system (γ_o) is correlated to the cold airflow ratio of the SFDD system (γ_b).

$$\gamma_o = \frac{T_{h,b} - T_{h,o} + \gamma_b \cdot (T_{c,b} - T_{h,b})}{T_{c,o} - T_{h,o}} \tag{16}$$

The heating and cooling energy consumptions are:

$$E_{c,o} = \begin{cases} \gamma_o \cdot \dot{m}_d \cdot (h_{m,c} - h_{c,d}) & T_{m,c} > T_{c,o} \ \& \ T_{m,c,dew} > T_{c,d,dew} \\ \gamma_o \cdot \dot{m}_d \cdot c_p \cdot (T_{m,c} - T_{c.o}) & T_{m,c} > T_{c,o} \ \& \ T_{m,c,dew} \le T_{c,d,dew} \\ 0 & T_{m,c} \le T_{c,o} \end{cases} \tag{17}$$

$$E_{p,o} = Max[0, \gamma_o \cdot \dot{m}_d \cdot c_p \cdot (T_{c,o} - T_{m,c})] \tag{18}$$

$$E_{h,o} = Max[0, (1-\gamma_o) \cdot \dot{m}_d \cdot c_p \cdot (T_{h,o} - T_{m,h})] \tag{19}$$

Assuming ideal power modulation (for example, variable frequency drives) and constant fan efficiency, the fan power is calculated using the cold and hot duct airflow rate ($\dot{m}_{c,o}$, $\dot{m}_{h,o}$) and the required fan heads (H_c and H_h).

$$E_{f,c} = C \times \dot{m}_{c,o} \times H_c / \eta \tag{20}$$

$$E_{f,h} = C \times \dot{m}_{h,o} \times H_h / \eta \tag{21}$$

Where,

$$H_c = (H_d - H_{c,\min}) \times \left(\dot{m}_{c,o} / \dot{m}_d\right)^2 + H_{c,\min} \tag{22}$$

$$H_h = (H_d - H_{h,\min}) \times \left(\dot{m}_{h,o} / \dot{m}_d\right)^2 + H_{h,\min} \tag{23}$$

The total fan power for the DFDD system is the sum of the hot air and cold air fan powers.

$$E_{f,o} = \frac{C}{\eta}\left(\dot{m}_c H_{c,\min} + \dot{m}_c (H_d - H_{c,\min})\left(\frac{\dot{m}_{c,o}}{\dot{m}_d}\right)^2 + \dot{m}_h H_{h,\min} + \dot{m}_h (H_d - H_{h,\min})\left(\frac{\dot{m}_{h,o}}{\dot{m}_d}\right)^2 \right) \tag{24}$$

Comparison of Energy Consumption

The energy savings are determined by taking the difference in consumption between the SFDD and DFDD systems under same ambient and building load conditions. The cooling, preheating and heating savings are expressed as the percentages of E_s (6 Btu/(lbm·hr) or 14 kJ/(kg·hr)) required to cool one pound of air from 75°F (23.9°C) and 50% relative humidity to 55°F (12.8°C) and 90% relative humidity.

$$\varphi_c = \frac{E_{c,b} - E_{c,o}}{E_s} \tag{25}$$

$$\varphi_p = \frac{E_{p,b} - E_{p,o}}{E_s} \tag{26}$$

$$\varphi_h = \frac{E_{h,b} - E_{h,o}}{E_s} \tag{27}$$

$$\varphi_f = \frac{E_{f,b} - E_{f,o}}{E_{f,b}} \tag{28}$$

If the minimum fan head at the cooling duct is assumed to be the same as that of the heating duct, introducing equations (9), (22) and (23) into equation (28) gives the potential fan power savings.

$$\varphi_f = 1 - \frac{\alpha}{(1+\varepsilon)^3}\left[1 + \left(\frac{1}{\alpha} - 1\right)\left(\gamma_o{}^3 + (1-\gamma_o)^3\right)\right] \quad (29)$$

SIMULATION RESULTS *AND* ANALYSES

The simulation was conducted on a constant air volume system with 20% outside air intake because dual duct systems are generally installed in medical or research facilities. The design room conditions are 75ºF (23.9°C) and 50% relative humidity. The design cooling coil discharging conditions are 55ºF (12.8°C) and 90% relative humidity.

A Bin analysis approach is used. Under each Bin temperature, energy consumption is simulated for all load conditions (from 100% heating to 100% cooling). The load conditions are represented by different cold airflow ratios (γ_b). For example, $\gamma_b = 0$ is 100% heating and $\gamma_b = 1$ is 100% cooling. The coincident wet bulb temperature of each Bin datum is determined using a San Antonio (TX) weather pattern.

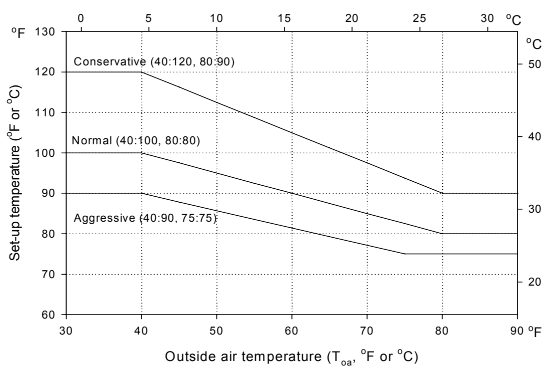
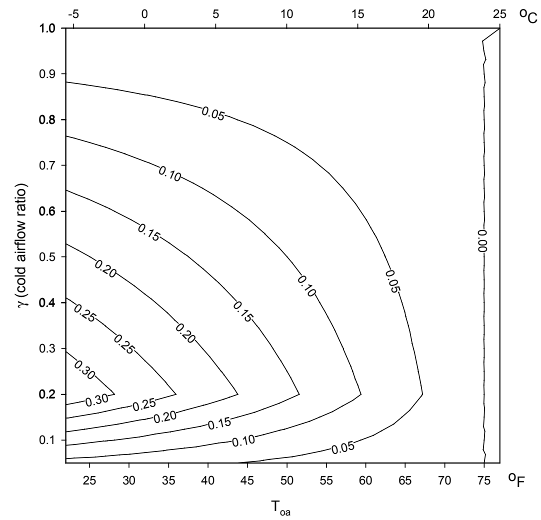


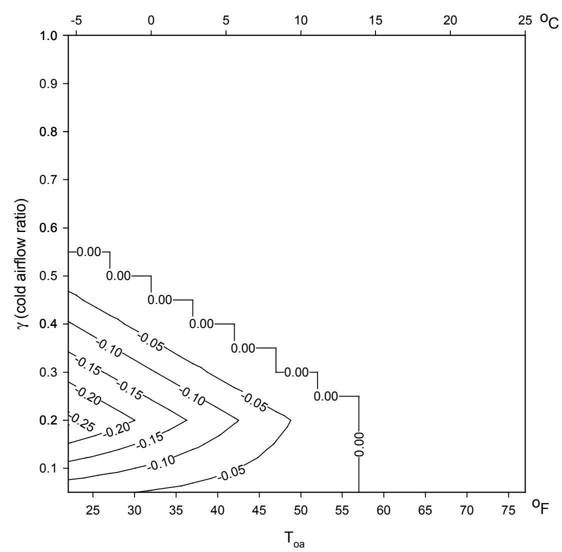
Figure 3. Hot deck temperature reset schedules

The cold deck set point is 55°F (12.8°C). Three hot deck reset schedules are simulated. Figure 3 shows the heating duct schedules used in the simulation. For the normal schedule, the hot deck temperature is 100°F (37.8°C) when the outside air temperature is 40°F (4.4°C) or lower, and the hot deck temperature is 80°F (26.7°C) when the outside air temperature is 80°F (26.7°C) or higher. For the conservative schedule, the hot deck temperature is 120°F (48.9°C) when the outside air temperature is 40°F (4.4°C) or lower, and the hot deck temperature is 90°F (32.2°C) when the outside air temperature is 80°F (26.7°C) or higher. For the aggressive schedule, the hot deck temperature is 90°F (32.2°C) when the outside air temperature is 40°F (4.4°C) or lower, and the hot deck temperature is 75°F (23.9°C) when the outside air temperature is 75°F (23.9°C) or higher.

Heating Saving



(a) Heating savings during winter



(b) Preheating penalties during winter

Figure 4. Heating and preheating energy savings versus ambient temperature and cold airflow ratio during winter (ambient temperature is less than 75°F)

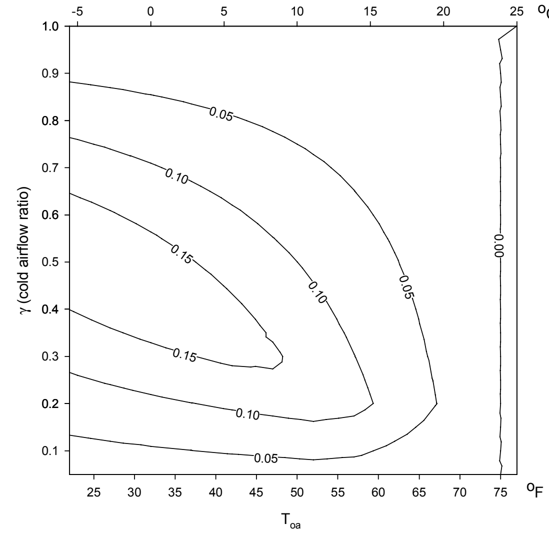
The simulation results show that thermal energy savings is independent of hot deck schedules when the outside air temperature is lower than 75°F (23.9°C). Thermal energy savings varies with hot deck schedules when the outside air temperature is higher than 75°F (23.9°C). Figure 4 (a) presents contour lines of heating savings during winter weather conditions (ambient temperature lower than 75°F (23.9°C)). The abscissa represents the outside air temperature (T_{oa}). The ordinate represents the cold airflow ratio (γ) of the SFDD system.

Heating savings is positive when the outside air temperature is below 75°F (23.9°C). The maximum savings occurs when the cold airflow ratio is the same as the outside air intake ratio (20%). The heating savings increases as the outside air temperature decreases for the same cold airflow rate.

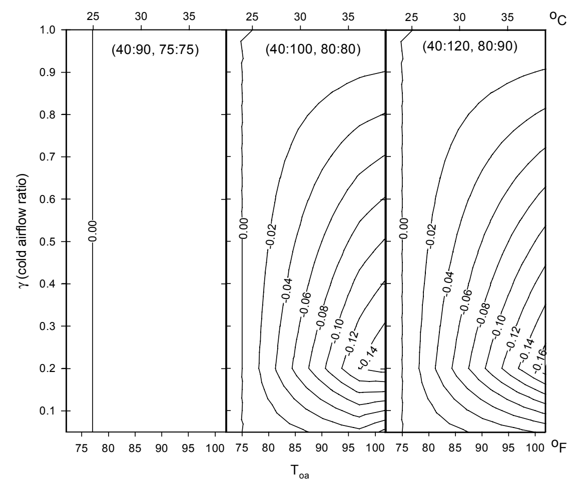
Preheating savings is negative or zero (Figure 4 (b)). The highest penalty occurs when the cold airflow ratio is the same as the outside air intake ratio. The lower the outside temperature, the higher the penalty. A zigzag line separates operations into non-preheating and preheating zones. Above the zigzag line, no pre-heat is required since the mixed air temperature is higher than the cold deck temperature.

Figure 5 (a) shows contour lines of total heating savings calculated as the sum of negative preheating savings and positive heating savings, during winter. The potential savings ranges from 0% to 20%. The maximum heating savings occurs at the preheating zigzag line, where the mixed air temperature of the cooling duct is the same as the cold deck temperature. For a typical building, the total heating savings ranges from 10% to 15% when the outside air temperature is below 55°F (12.8°C). For example, the total heating savings is 15% (0.9 Btu/lbm or 2.1 kJ/kg) when the outside air temperature is 45°F (7.2°C) and the cold airflow ratio is 0.3. If the total building airflow is 100,000 ft^3/min (47.2 m^3/s), the total heating savings is 405,000 Btu/hr (118.7 kW).

Figure 5 (b) shows heating savings with three different schedules during summer weather conditions (T_{oa} >75°F (23.9°C)). The heating savings is negative when the hot deck temperature is higher than 75°F (23.9°C). The higher the hot deck temperature, the higher the energy penalty. The maximum heating penalty occurs when the cold airflow ratio is the same as the outside air intake ratio (20%). During summer, the hot deck temperature should be set as low as possible.



(a) Total heating energy savings during winter



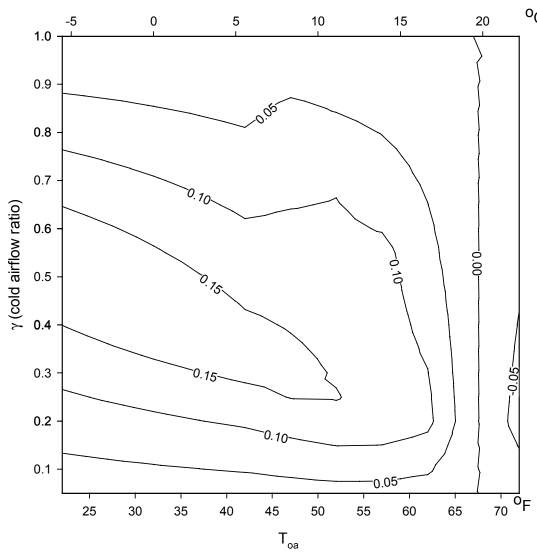
(b) Total heating energy savings during summer

Figure 5. Total heating energy savings versus the ambient temperature and the cold airflow ratio

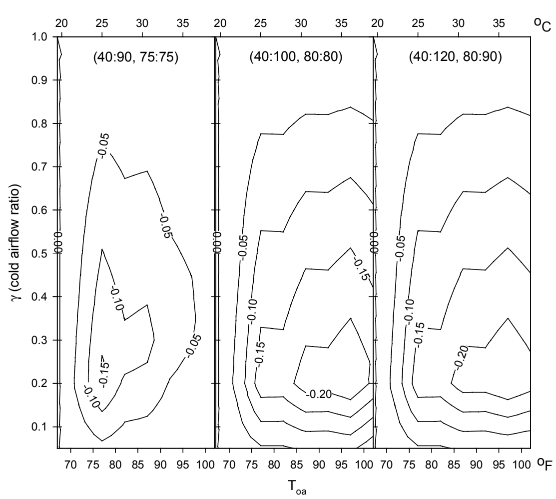
The heating penalty is small for a typical system during summer. When the cold airflow ratio is above 70%, the heating penalty is less than 6%. However, this penalty can be higher at nighttime and during weekends when the cold airflow ratio is lower.

Cooling Savings

Figure 6 (a) shows contour lines of cooling savings during winter. The potential savings ranges from 0% to 25%. The maximum cooling savings occurs when the mixed air temperature of the cooling duct is equal to the cold deck temperature. For a typical building, the cooling savings ranges from 10% to 20% when the outside temperature is below 60°F (15.6°C). For example, the cooling savings is 16% (0.96 Btu/lbm or 2.233 kJ/kg) when the outside air temperature is 45°F (12.8°C) and the cold airflow ratio is 0.3. If the total building airflow is 100,000 ft^3/min (47.2 m^3/s), the cooling savings is 432,000 Btu/hr (126.6 kW).



(a) Cooling energy savings during winter



(b) Cooling energy savings during summer

Figure 6. Cooling energy savings versus the ambient temperature and the cold airflow ratio

Figure 6 (b) shows cooling savings with three different schedules during summer. The cooling energy savings is negative because the DFDD system has more outside airflow through the cooling coil. The higher the hot deck temperature, the higher the cooling energy penalty. The maximum cooling energy penalty occurs when the cold airflow ratio is the same as the outside air intake ratio (20%). The hot deck temperature should be set as low as possible for cooling during summer weather conditions.

The cooling penalty is small for a typical system during summer. When the cold airflow ratio is above 70%, the cooling penalty is less than 10%. For example, the total cooling energy penalty is 15% (0.9 Btu/lbm or 2.1 kJ/kg) when the outside air temperature is 95°F (35.0°C) and the cold airflow ratio is 0.5. If the total building airflow is 100,000 ft^3/min (47.2 m^3/s), the total cooling energy penalty is 405,000 Btu/hr (118.7 kW). However, this penalty can be significant higher at night and on the weekend when the cold airflow ratio is lower.

Fan power savings

Figure 7 shows fan power savings versus the cold airflow ratio and the minimum fan head ratio. The fan power savings is up to 70%. The maximum savings occurs when the hot airflow equals the cold airflow. For a typical system, the fan power savings ranges from 20% to 60% under the following conditions: the minimum fan head ratio ranges of 0.4~0.7, and the cold airflow ratio of 0.05~0.95.

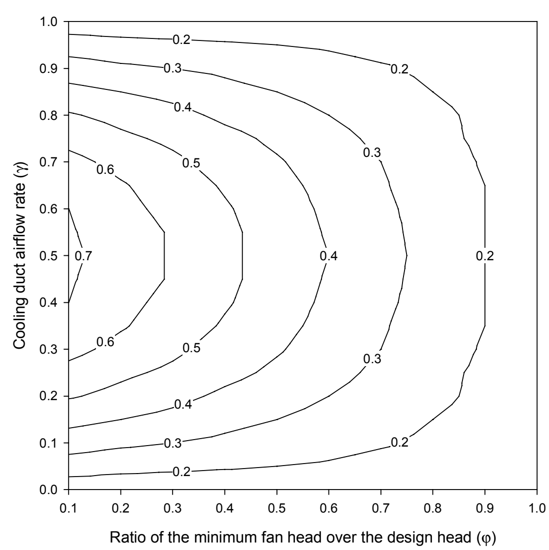


Figure 7. Fan power savings versus minimum fan head ratio and cold airflow ratio

APPLICATION

The annual potential energy savings can be calculated using the models presented in the paper. The method is demonstrated using a hypothetical building in San Antonio. The hypothetical building has a total airflow of 100,000 ft^3/min (47.2 m^3/s) with a fan of 150 hp (111.75 kW). The minimum fan head ratio is assumed to be 0.50.

The number of operation hours is first determined for each bin temperature for both occupied (8:00 AM to 8:00 PM) and unoccupied hours [*Degelman, 1985*]. The average cold airflow is then estimated for each condition. Based on the bin temperature and estimated cold airflow ratio, the percentage cooling and heating savings are determined using Figures 5 and 6. Based on the cold airflow ratio and the minimum fan head ratio, the percentage fan power savings ratio is determined using Figure 7. Table 1 summarizes the results. The thermal energy penalty is identified when the outside air temperature is higher than 65°F (18.3°C).

Based on the hourly energy savings percentage and the number of hours of operation, the annual energy savings can be determined as the sum of the savings for each temperature bin. The results are summarized in Table 2.

Table 1: Summary of the Potential Energy Savings and Penalty Calculation

Temperature (°F / °C)	Occupancy	Hours	Cold Air-flow Ratio	Percentage Energy Savings or Penalty		
				Heating	Cooling	Fan
99.5 / 37.5	Occupied	138	0.9	-1.9%	-2.9%	25.0%
	Unoccupied	0	0.8	-3.8%	-5.7%	34.0%
89.5 / 31.9	Occupied	985	0.9	-1.0%	-2.8%	25.0%
	Unoccupied	50	0.7	-3.5%	-8.3%	41.0%
79.5 / 26.4	Occupied	1095	0.8	-0.7%	-4.5%	34.0%
	Unoccupied	1103	0.6	-1.5%	-8.9%	45.0%
69.5 / 20.8	Occupied	984	0.7	1.3%	-1.2%	41.0%
	Unoccupied	1164	0.5	2.2%	-2.0%	46.0%
59.5 / 15.3	Occupied	639	0.6	5.0%	7.8%	45.0%
	Unoccupied	713	0.4	7.4%	10.3%	45.0%
49.5 / 9.7	Occupied	411	0.5	10.2%	12.4%	46.0%
	Unoccupied	834	0.3	14.1%	15.6%	41.0%
39.5 / 4.2	Occupied	116	0.4	17.0%	17.0%	45.0%
	Unoccupied	423	0.2	10.3%	10.3%	34.0%
29.5 / -1.4	Occupied	12	0.2	8.8%	8.7%	34.0%
	Unoccupied	90	0.2	8.8%	8.7%	34.0%
22 / -5.6	Occupied	0	0.2	7.5%	6.6%	34.0%
	Unoccupied	3	0.2	7.5%	6.6%	34.0%

Table 2: Summary of Annual Energy Savings and Penalty

	Energy (MMBtu/yr (GJ/yr) or Wh/hr)			Cost ($/yr)			
	Heating	Cooling	Fan	Heating	Cooling	Fan	Total
Penalty	-104 (-110)	-589 (-621)	0	-$520	-$2,946	$0	-$3,466
Savings	959 (1,012)	1,017 (1,073)	388,879	$4,795	$5,085	$19,444	$29,324
Net savings	855 (902)	428 (452)	388,879	$4,275	$2,139	$19,444	$25,858

Energy prices: $5/MMBtu or $4.739/GJ for heating and cooling and $0.05/kWh

The DFDD system saves significant amounts of thermal and fan energy for the hypothetical building. The potential annual energy cost savings is estimated to be $25,858/yr, which includes $6,414 for thermal energy ($5/MMBtu or $4.739/GJ) and $19,444 for the fan power ($0.05/kWh).

Heating and cooling energy penalty is significant during summer. The heating penalty is 11% of the potential heating savings. The cooling penalty is 58% of the cooling energy savings. The cost penalty is 12.4% of the net cost savings. Research is suggested on methods to decrease the cost penalty of the DFDD constant volume systems.

CONCLUSION

Models for thermal and fan power energy consumptions are developed for the single-fan, dual-duct constant-volume system and the dual-fan, dual-duct constant-volume system. The annual retrofit energy savings can be estimated using the recommended procedure or the models presented.

The DFDD systems use significantly less fan power than SFDD systems. The DFDD systems consume less heating and cooling energy during mild winter conditions and more heating and cooling during summer conditions. The energy performance of the DFDD systems can be significantly improved if the energy penalty is decreased or eliminated.

This study is limited to constant volume systems without economizers.

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NOMENCLATURE

C = Constant
c_p = Specific heat for dry air (J/(kg·ºC) or Btu/(lbm·ºF))
E = Energy consumption (W or Btu/hr)
H = Fan head (Pa or inH_2O)
h = Air enthalpy (J/kg or Btu/lbm)
$\dot{m}$ = Airflow rate (kg/s or lbm/hr)
T = Air temperature (ºC or ºF)
α = Minimum fan head ratio (H_{min}/H_d, where $H_{min} = H_{c,min} = H_{h,min}$)
β = Outside air intake ratio ($\dot{m}_{oa} / \dot{m}_d$)
γ = Cold airflow ratio ($\dot{m}_c / \dot{m}_d$)
ε = Air leakage rate
η = Fan efficiency
φ = Energy savings

Subscripts

b = Base case systems, single-fan, dual-duct constant volume systems
c = Cooling, cold deck
d = Designed
dew = Dew point
f = Fan
h = Heating, hot deck
m = Mixed
min = Minimum
o = Optimized systems, dual-fan, dual-duct constant volume systems
oa = Outside air
p = Preheating
r = Room air

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