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Impacts of Static Pressure Reset on VAV System Air Leakage, Fan Power and Thermal Energy

Part 2: Case Demonstration for a Typical Climate System

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ABSTRACT

In Part 1 of this paper, the theoretical models, integrating the fan airflow, fan head, air leakage factors, are developed to analyse the impacts of the static pressure reset on both pressure dependent and pressure independent terminal boxes.

In this part, a simulated air handling unit(AHU) system in Omaha NE is used to demonstrate the energy savings performance in one typical climate year. This AHU system has a static pressure reset system and two constant static pressure systems, one having pressure dependent terminal boxes and one having pressure independent terminal boxes. These simulated systems were compared mainly on the basis of fan power energy consumption and thermal energy consumption in totally a year.

The example presents a good agreement with the theoretical model and simultaion results. It was also shown that static pressure reset provides a promising and challenging way for the energy performance in VAV system.

INTRODUCTION

This paper is the second part of impacts of static pressure reset in VAV system performance. In the first part, we developed the theoretical models to analyze the impacts of static pressure reset on fan airflow, fan head, air leakage, fan power and thermal energy for both pressure dependent and pressure independent terminal units. Also, the advantages of the implementation of the static pressure reset are discussed in details on the basis of the fan power and thermal energy savings.

In order to have a practical concept of the energy consumption savings， the static pressure reset is implemented in an simulated air handling system in a typical climate. In this part, the mathematical models for a total year's fan power energy and thermal energy consumption are developed based on the BIN weather data in Omaha, NE, USA, to simulate the AHU systems performance, which includes a static pressure reset system and two constant static pressure systems (one having pressure-dependent terminal unit and one having pressure-independent terminals unit).

SYSTEM DESCRIPTION

Air Handling System

Three air handling systems: one with static pressure reset, one with constant static pressure and pressure dependent terminal units, and one with constant static pressure and pressure independent terminal units. The results of the three simulations are compared on the basis of fan heads, terminal unit air leakage, fan power, fan energy consumption, thermal energy consumption, and duct leakage.

Fig. 1 shows a typical AHU system with air leakage from duct, return air from building, exhaust air and outside air. An economizer is installed enabled when outside air temperature is lower than supply air temperature.

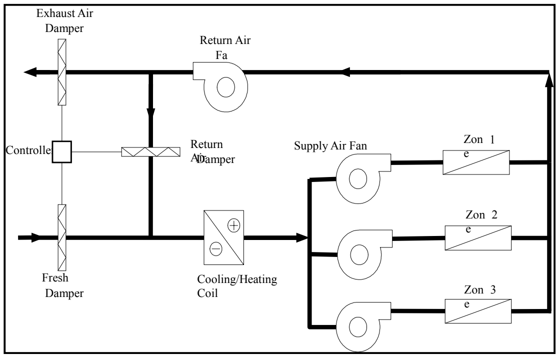

Figure 1. A typical AHU system in the simulation

The modeled air handling system delivers 20,000 CFM supply air to terminal units in a single zone at the design condition. The minimum supply fan airflow ratio is 0.2. The supply fan is modeled as being on a VSD, such that the supply fan airflow can modulate from the design airflow down to a minimum airflow of 30% according to the zone load.

The design supply air temperature T_{sa} is modeled as 55 ℉. The model includes reheat which brings the supply air temperature up to an ideal supply temperature $T_{sa,\ ideal}$ determined by the zone load.

SYSTEM MODEL DEVELOPMENT

Supply Airflow

The supply airflow ratio α is assumed to be equal to the fractional zone load q_{AHU}, although the model enforces a lower limit of 0.3. When static pressure reset is utilized, the airflow ratio of the supply air conditioned during a given hour is taken as α. It is also true when pressure independent terminal units are utilized, as it is assumed that these units will experience negligible leakage.

The instantaneous airflow, in CFM, is thus expressed as

$$Q = \alpha \cdot Q_d \qquad (1)$$

However, when pressure independent terminal units are used in a constant static pressure system, leakage through the terminal unit dampers due to excess static pressure at partial loads will cause the actual airflow ratio to be higher than that determined by the fractional zone load. This actual airflow, denoted as $\alpha_{actural}$, is a function of the airflow ratio α, constant static pressure set point ps, and the terminal unit design pressure $p_{b,d}$.

The model assumes that the terminal units, whether pressure dependent or independent, require a duct static pressure of 2 in.w.g. at the design airflow Q_d Thus, at the design condition for a constant static pressure system, both p_s and $p_{b,d}$ are equal to 2 in.w.g. At partial load, however, p_s remains constant, while $p_{b,d}$ decreases as a function of the airflow ratio α. The model assumes that the terminal units require a minimum value of 0.5in.w.g. for $p_{b,d}$ to overcome the pressure drop of their dampers.

Thus, the terminal unit design pressure in in.w.g, can be expressed as

$$p_{b,d} = \begin{cases} p_s \cdot \alpha^2, 0.5 < \alpha < 1 \\ 0.5, \alpha \le 0.5 \end{cases} \qquad (2)$$

It should be noted that, by the above function, it can be shown that the minimum terminal unit design pressure will be maintained at an airflow ratio of 0.5.

As noted earlier, the actual airflow ratio $\alpha_{actural}$ is a function of α , p_s and $p_{b,d}$, such that

$$\alpha_{actual} = \alpha \sqrt{\frac{p_s / p_{b,d}}{1+\alpha^2 (p_s / p_{b,d-1})}} \qquad (3)$$

This instantaneous actual airflow, in cfm, for the constant static pressure system with pressure dependent terminal units is thus expressed as

$$Q_{actual} = \alpha_{actual} \cdot Q_d \qquad (4)$$

Fan Head

The design fan head $H_{f,d}$ is assumed to be 6 in.w.g. This assumes that the system requires 2 in.w.g of static pressure to maintain all terminal units at fully open positions and a 4 in.w.g. of static pressure drop at the branch supply duct occurs at the design airflow.

The fan head of the air handling system having static pressure reset is modeled by the equation

$$H_{f,reset} = \begin{cases} H_{f,d} (Q / Q_d)^2, 0.5 < \alpha < 1 \\ 1.5, \alpha \le 0.5 \end{cases} \qquad (5)$$

It should be noted that 1.5in.w.g is the minimum fan head that will produce the minimum value of 0.5in.w.g. for $p_{b,d}$.

The fan head of the air handling system having a constant static pressure is modeled by the equation

$$H_{f,const} = p_s + \Delta p_{branch} (Q_{actual} / Q_d)^2 \qquad (6)$$

Fan Power

When modeling the fan power, it is assumed that the fan efficiency at the design airflow is 70% and that the efficiency remains constant for all fractional airflows. In reality, the fan should be selected such that peak efficiency occurs at the design airflow, with fractional airflows resulting in lower fan efficiency. The fan efficiency is intended to be conservative.

Define fan power consumption without static pressure reset as the base case, and with static pressure reset as improved case. The fan power saving ratio is the ratio of fan power consumption difference between base and improved case to the fan power consumption at design condition.

$$\Delta \dot{E}_{fan} = \frac{E_{fan,b} - E_{fan,i}}{E_{fan,d}} \qquad (7)$$

Duct Leakage

Duct leakage through the main supply duct is modeled. Assuming that the main duct is composed of unsealed, rectangular ductwork having 25 joints per 100 linear feet, the leakage class is 48 according to Chapter 32 of the 1997 ASHRAE Fundamentals Handbook.

According to ASHRAE handbook [1], the air leakage can be calculated by

$$CFM_l = \delta \times \Delta p_s^N \qquad (8)$$

Assuming Δp_s is the average of duct static pressure, N=0.5, building pressure is zero, then the leakage at downstream duct can be expressed by

$$\sqrt{\frac{(p_{s,d}+0)/2}{(p_{s,d}+0)/2}} = \frac{CFM_{l,down}}{CFM_{l,down,d}} = 1 \qquad (9)$$

Therefore, the air leakage at the downstream duct is the same with design condition.

Economizer

To predict realistic thermal energy consumption, the model includes a temperature economizer, which is controlled based on low and high temperature limits. These limits are T_L=40 ºF and T_H=60 ºF. Recirculated return air and outside air are mixed to achieve a mixed air dry bulb temperature as close to the design cold deck supply air dry bulb temperature as possible, irregardless of enthalpy, as long as the outside air dry bulb temperature is within the design limits. The model assumes that the air handling system will utilize the amount of outside air required to produce mixed air at the design supply air dry bulb temperature or minimum outside air intake ratio of 20%, whichever is greatest.

Outside Air Intake

The average outside air ratio is calculated as

$$X = \frac{\sum Q_{\text{vent}}}{\sum Q_{\text{supply}}} \qquad (10)$$

The max zone outside airflow ratio is calculated as

$$Z = Max\left[\frac{Q_{\text{vent},i}}{Q_{\text{supply},i}}\right] \qquad (11)$$

Thus, the outside airflow ratio at the air handling unit is

$$Y = \frac{X}{1+X-Z} \qquad (12)$$

The minimum outside air brought in by the system is

$$Q_{oa} = Y \cdot Q_{total} \qquad (13)$$

The total recirculated return air, in CFM is

$$Q_{ra} = Q_{total} - Q_{oa} \qquad (14)$$

Mixed Air

The average dry bulb temperature and mean coincident wet bulb temperature of each of the Omaha BIN data 5 ºF temperature bins was used to fix an outside air condition for each bin data. The nominal temperature of each bin is taken to be the lowest temperature in the bin.

The minimum outside air mass flow rate is calculated as

$$\dot{m}_{oa,\min} = \frac{Q_{oa,\min} \cdot 60}{V_{ra}} \qquad (15)$$

The maximum recirculated air mass flow rate is calculated as

$$\dot{m}_{ra,\max} = \frac{Q_{ra,\max} \cdot 60}{V_{ra}} \qquad (16)$$

Thus, the total supply air mass flow rate is

$$\dot{m}_{sa} = \dot{m}_{oa,\min} + \dot{m}_{ra,\max} \qquad (17)$$

Thermal Energy Saving

Due to the reduction of air flow supply after static pressure reset, the thermal energy, include both cooling energy and heating energy, can be achieved.

The annual thermal energy consumption for the cooling with the economizer used can be expressed as:

$$q_c = \begin{cases} (Annual\,Hours) \cdot \alpha \cdot \dot{m}_{sa} \cdot \left| h_{sa} - h_{ma} \right|, & T_{ma} > T_{sa} \\ 0, & T_{ma} < T_{sa} \end{cases} \qquad (18)$$

The annual thermal energy consumption for the reheating can be expressed as:

$$q_h = \begin{cases} (Annual\,Hours) \cdot \alpha \cdot \dot{m}_{sa} \cdot \left| T_{sa,ideal} - T_{sa} \right|, & T_{ma} > T_{sa} \\ 0, & T_{ma} < T_{sa} \end{cases} \qquad (19)$$

And the total thermal energy consumption is:

$$q_h = q_h + q_c \qquad (20)$$

It should be noted that other parameters used in the excel spreadsheet model can refer to the thermal energy model in part 1 of this paper.

SYSTEM SIMULATION PARAMETER

Bin weather data method is used to simulate the total year load distribution.

Climate Description

AHU system is located in Omaha, NE. As seen from the temperature distribution, the climate in Omaha is a typical mild weather. Figure 2 shows the number of hours per year that Omaha experiences temperatures from 0 to 100℉.

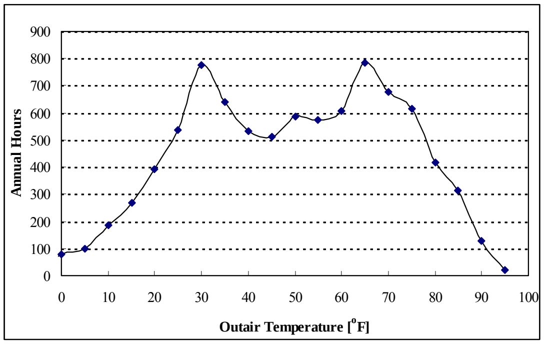

Figure 2. Omaha's annual hours at temperatures from 0 to 100°F

Design Room Air Conditions

Room air dry-bulb, T_{rm}: 75°F
Room air enthalpy h_{rm}: 28 Btu/lbda

Design Return Air Conditions

Return air dry-bulb, T_{ra}: 75°F
Return air enthalpy, h_{ra}: 28 Btu/lbda
Return air sp. Volume, v_{ra}: 13.7 ft^3/lbda

Supply Air Conditions

Supply air dry-bulb temp, T_{sa}: 75°F
Supply air enthalpy, has: 28 Btu/lbda

Economizer Range

Low temp, T_L : 40°F
High temp, T_H : 60°F

AHU Load Function

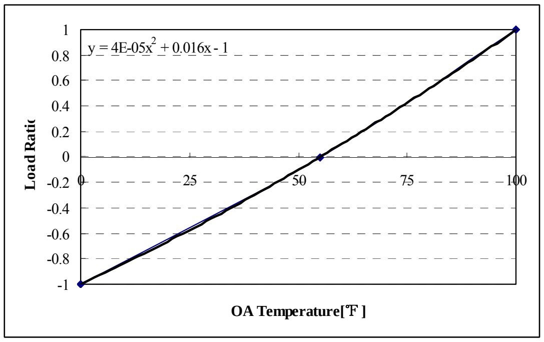

Figure 3. AHU load function.

Static Pressure Design Parameters

Table1
Static Pressure Design Parameters

Description	Value
H_{fd}	6 in.w.g
Ps	2 in.w.g
$P_{b,d,min}$	0.5 in.w.g
P_{branch}	4 in.w.g
η_f	0.7
A_{duct}	100 ft^2
CL	48

Minimum Design Outdoor Air

Table 2
Minimum Design Outdoor Air Parameters

Description	Value
Q	20000 cfm
Avg OA Ratio X	0.2
Max Zone OA Ratio Z	0.2
AHU Intake Ratio Y	0.2
Q_{oa}	4000 cfm
Q_{ra}	16000 cfm
m_{oa}	17531 lb/hr
m_{ra}	70124 lb/hr
m_{sa}	87655 lb/hr

As described previously, the static reset model was developed in the Spreadsheet Model by Microsoft Excel. This model allows for easy manipulation of the model criteria to simulate three air-handling systems: one with static pressure reset, one with constant static pressure and pressure dependent terminal units, and one with constant static pressure and pressure-independent terminal units.

This model compare the results of the three simulations on the basis of fan heads, terminal unit air leakage, fan power, fan energy consumption, thermal energy consumption, and duct leakage.

SYSTEM SIMULATION RESULTS

Table 3 shows a summary of the total annual fan energy and thermal energy consumptions of the three modeled air handling systems. This energy consumption will be examined in details in the following parts.

Table 3
Summary of Fan and Thermal Energy Consumption

System	Fan Energy [MBtu]	Thermal Energy [MBtu]
Static Pressure Reset	95	791
Constant Static Pressure(Pressure Dependant)	254	1176
Constant Static Pressure(Pressure Independent)	139	791

System Fan Heads Comparison

As expected, the three simulated air handling systems exhibit different system fan heads at all partial load conditions. These fan heads can be seen in Figure 4. The static pressure reset system shows the lowest fan heads.

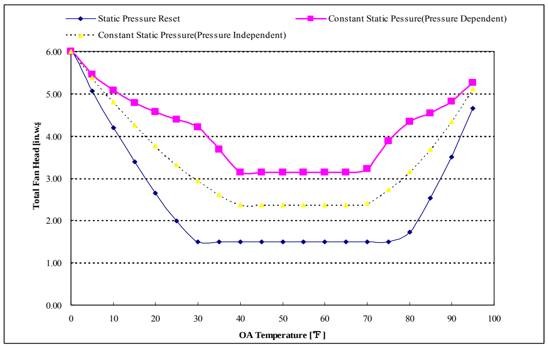

Figure 4. System total fan heads comparison.

AHU leakage

Leakage through terminal units

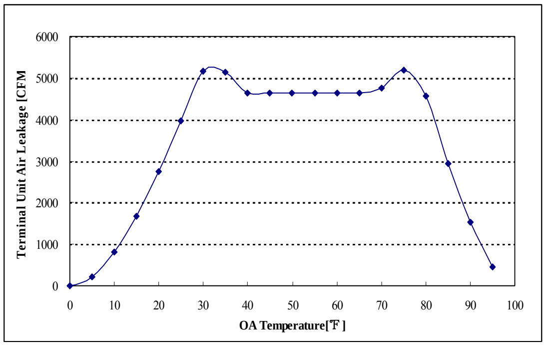

Figure 5. AHU leakage through pressure dependent terminal boxes.

Figure 5 shows the air leakage of the constant static pressure system with pressure dependent terminal units. The static pressure reset model assumed that negligible terminal unit leakage occurred in the systems having either static pressure reset or pressure independent terminal units.

Leakage through duct

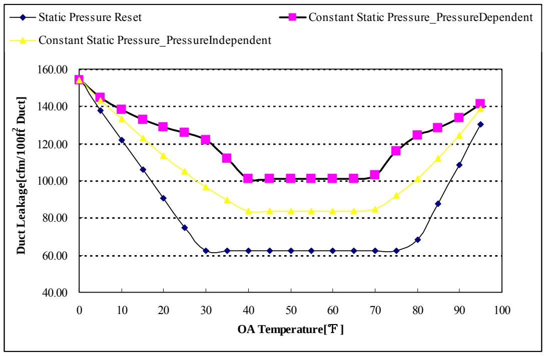

Figure 6. AHU leakage through duct.

In figure 6, the duct leakage in a static pressure reset system is compared with the constant static pressure system. It can be easily seen that the system with static pressure reset has less duct leakage for the less static pressure in the partial load condition.

Fan Power Saving

Pressure dependent

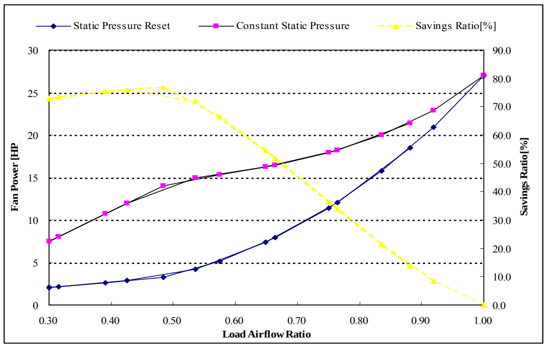

Figure 7. Comparison of fan power with pressure dependent terminal units utilized.

Pressure independent

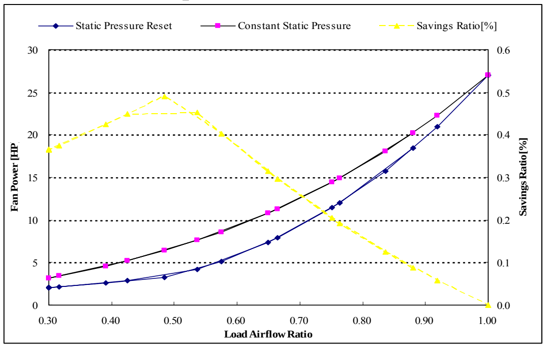

Figure 8. Comparison of fan power with pressure independent terminal units utilized.

In both Figure 7 and 8, the fan power of an air handling system in Omaha with constant static pressure is plotted versus that with static pressure reset across a range of airflow ratios. The difference is the terminal unit type, pressure independent or pressure dependent.

As seen from the figures, the system with static pressure reset consumes less fan power than the one with constant static pressure. For pressure dependent type, the savings ratio can reach high as 80% shown in the secondary y- axis. Much of the savings is attributed to the elimination of air leakage and the reduced fan head by static pressure reset in partial load condition. For pressure independent type, the savings ratio can reach 50%. The savings is attributed to the reduced fan head of the static pressure reset system in partial load condition.

Fan Energy Consumption Comparison

Reductions in fan power due to static pressure reset could result in significant savings in fan energy consumption.

Pressure dependent

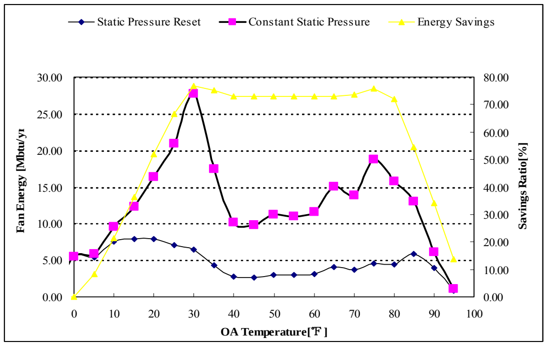

Figure 9. Comparison of fan energy savings with pressure dependent terminal units utilized.

Pressure independent

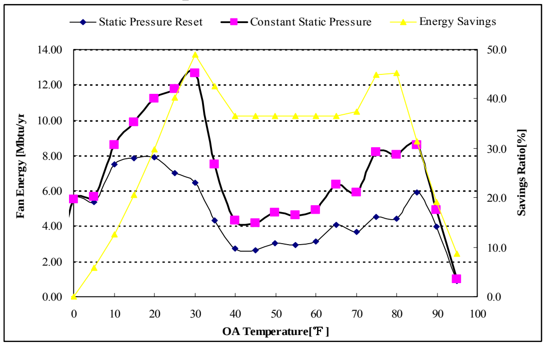

Figure 10. Comparison of fan energy savings with pressure independent terminal units utilized.

Figure 9 compares the fan energy consumption of the air handling system with static pressure reset to the one with constant static pressure and pressure dependent terminal units. As seen from the figure, fan energy savings of over 75% can be achieved through the reductions in airflow and fan head that result from static pressure reset.

Figure 10 compares the fan energy consumption of the air handling system with static pressure reset to the one with constant static pressure and pressure independent terminal units. For pressure independent terminal units, over 35% energy savings can be achieved through the reduced fan head.

Thermal Energy Consumption

Static pressure reset can achieve large amount of thermal energy savings for air handling systems with pressure dependent terminal units through reduced static pressure, and reduced air leakage. In contrast, when pressure independent terminal unit is used, less savings of thermal energy is achieved.

Figure 11 shows the thermal energy savings from static pressure reset in an AHU with pressure dependent terminal units. In the peak heating and cooling seasons, the thermal energy savings are in the range of 20 to 30 percent, which saves a significant amount of energy.

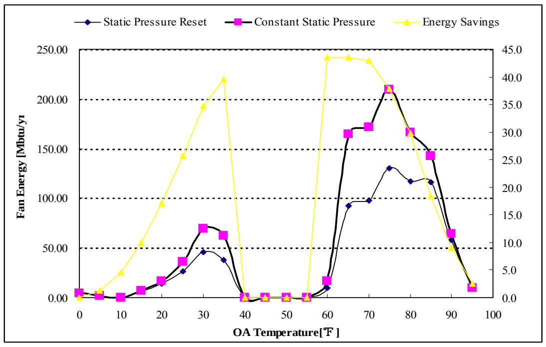

Figure 11. Thermal energy savings with static pressure reset.

CONCLUSIONS

- For a typical AHU system with independent and dependent terminal units, the static pressure reset can significant improve the fan power and fan energy savings.
- When the pressure dependent terminal units used, obvious thermal energy consumption can be reduced.
- Research on the optimal pressure reset method is promising and challenging to make the energy savings minimum.

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