ENERGY SAVINGS BY CHANGING CONSTANT AIR VOLUME SYSTEMS (CAV) TO VARIABLE AIR VOLUME SYSTEMS (VAV) IN EXISTING OFFICE BUILDINGS. - EXPERIENCE FROM A PLANT RECONSTRUCTION BASED ON A NEW SUPPLY AIR TERMINAL DEVICE CONCEPT.

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Summary

In commercial buildings a CAV (constant air volume) system is normally used for ventilation and air-conditioning. In order to keep the room temperature in required level during the summer period quite high airflow rates and higher amounts of energy for heating and distribution of air are needed. By changing the system to VAV operation, considerable energy savings can be achieved. However, such a change becomes meaningful only if the system as a whole is adapted to the new way of operation. The critical components are the supply air terminal devices, which must have good controlling properties, a low noise level and provide a comfortable air flow and temperature pattern in the room within the variable volume flow range in question. The paper accounts for experiences from a reconstruction from CAV to VAV in a 5000 m^2 about 40 year old office building. Investigations were carried out with new type of supply air devices, which were installed in this office building.

Keywords: CAV, VAV, supply air terminal devices, intelligent diffuser, climate control, energy efficiency, ventilation

1. INTRODUCTION

In commercial buildings the need of energy for ventilation and air conditioning may account for a substantial part of the need of energy. There is, at least in Sweden, a large existing stock of 20 to 30 year old office buildings, which are supplied with centralized constant air volume flow (CAV) air conditioning systems with the primary function to keep the room temperature at an acceptable level summertime. At times high air flow rates are needed to master the loads from solar radiation, electrical equipment and people. However, the heat loads are only occasionally at the peak level, which means that in a CAV system the air flow rate is unnecessarily high most of the time. The need of energy for distribution and heating of air becomes high, although the systems are usually provided with heat recovery and are in operation mainly during working hours.

If the system I changed to a air volume flow VAV system, the average air volume flow will be lower and need of electrical energy for of air distribution and heat for supply air heating is reduced. With a decreased average air volume flow the average efficiency of heat recovery will be improved, which also contributes to a decreased need of supply air heating.

One cause of high need of energy may be that the air outlet devices need a quite high supply air temperature if disturbing draft should be avoided. This is the case e.g. when displacement ventilation was installed in quite small rooms, which was a quite frequent in Sweden when existing offices were rebuilt in the early 90ies. For acceptable thermal comfort in these rooms a supply air temperature of at least +19°C is needed in practice. This implies poor cooling capacity of the air supplied and a quite high need of energy for supply air heating. If this type of air supply devices could be replaced with devices, which enable a low supply air temperature without causing draught problems, the need of heat for supply air conditioning will decrease. At the same time the cooling capacity of the supply air will increase and the prerequisites for room temperature control will improve.

However, a change of an existing CAV system to VAV is somewhat complicated from both technical and economic point of view. It is a simple task to change from constant speed fans to variable speed fans. The necessary changes in the rest of the system may be both complicated and costly. Here the availability of supply air devices with appropriate properties is the critical issue. The outlet air device must have good variable air volume flow properties:

- Demand based control by room temperature, by room air quality or by a combination of these should be possible.
- A high pressure difference over the device, at least 100 Pa, must be possible to master without noise problems
- The air movement pattern in the room must be stable independent of the supply air flow rate
- It should be possible to supply air with a low temperature, +15°C or lower, without risk of disturbing draught
- The cost must be acceptable from a life cycle point of view

The paper accounts for experiences from a change from CAV to VAV, based on a new supply air device that fulfils the above criteria. The change is carried through in a 5000 m^2 about 40 year's old office building. The plant was retaken in operation in September 2003 and is now monitored regarding the energy needs, the function of the new VAV and the indoor climate. The paper accounts for the ideas behind the system change and for preliminary results from the monitoring project.

2. METHODS

2.1. Energy savings

A variable air volume system changes the supply air flow rate according to the need of the supported room or zone. By decrease of the average air flow the need of electrical energy for air distribution will decrease. The power demand will vary with the ventilation flow according to the cube fan law [1, 2]:

$$P(kW) = \frac{V_{VAV}\,\Delta p_{tot}}{\eta_{tot}} = \frac{V_{VAV}\,k_1 V_{VAV}^2}{\eta_{tot}} = \frac{k_1 V_{VAV}^3}{\eta_{tot}} \qquad (1)$$

Where P is the power use measured as power input to the fan engine (kW), Δp_{tot} the total pressure drop in the system (kPa), η_{tot} the total efficiency of the fan system (%), k_1 a constant and V_{VAV} is the mechanical ventilation flow in a VAV-system (m^3/s).

Power demand for central heating and cooling can be determined as follows when humidity is regarded [2]:

$$P_{h/c}(kW) = V_{VAV}\,\rho c_p \Delta T \qquad (2)$$

Where $P_{h/c}$ is the power needed for central cooling or heating, V_{VAV} is the mechanical ventilation flow in VAV-system (m^3/s), ρ is the density of air (kg/m^3), c_p is the specific heat capacity for air (kJ/(kg K)), ΔT is the temperature change of airflow (between inlet and outlet) (K). According to presented formula by a decrease of the average air flow rate the amount of heat needed for supply air heating and cooling is decreased. If the supply air temperature can be decreased, the need of heat supply air heating will decrease substantially as shown in fig. 1

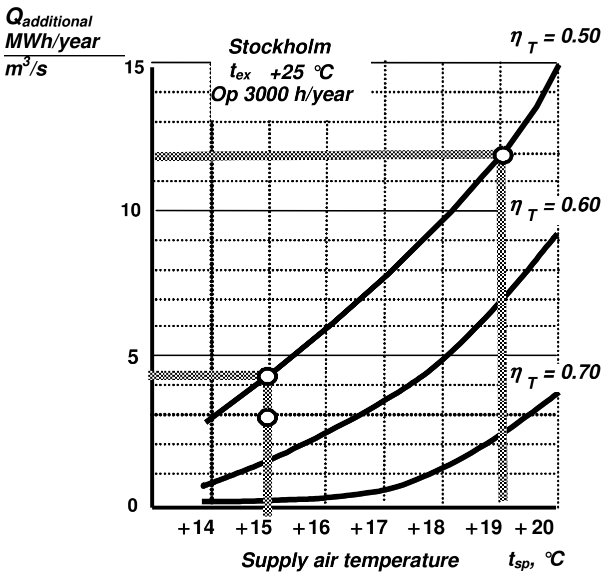

Figure 1. The annual additional energy needed for supply air heating in an air supply unit with different heat recovery temperature efficiencies. Diagram responds to the location of Stockholm. The plant is in operation 3000 hours/year.

Temperature efficiency is calculated according to the formula:

$$\eta_t = \frac{t_{rec} - t_o}{t_e - t_o} \qquad (3)$$

The temperature efficiency of the heat recovery will increase when the air flow through the heat recovery decreases. By a decreased average air flow the average heat recover will increase.

2.2 Problems

A VAV system consists of different components such are central ventilation aggregate with supply and exhaust fans with variable frequency inverters for fan speed control, duct system, variable air volume devices (most commonly VAV boxes on the duct line are used) and room air distribution system. When designing a VAV system for a building it is necessary to understand the importance of those different components. Most common problems associated with VAV systems are most often due to improper design of different components and/or control of the system [3].

The limitations that influence the selection of the VAV system components and also energy savings are due to the indoor climate quality requirements. VAV devices tend to be too noisy in the situations with a high static pressure drop across the device. Problems occur also with the use of improper diffusers, which may induce uncomfortable conditions for occupants [3]. The most common reason for thermal complaints in office buildings is draft [4] and higher velocities with lower temperatures increase the percentage of occupants objecting to drafts [5].

It is desired that the chilled supply air should be rapidly mixed with the air within the space. The mixed airstreams temperature should have reached the design room temperature and air velocity should have met the required level by the time that it enters the occupied zone. For office-type applications, where occupants are seated for much of the time, this is nominally defined as an envelope extending from floor level to a height of 1,2m, and to within 0,15 m of any other room surface [6].

The reason why most of the usual supply air diffusers are unsuitable for VAV systems is that they are meant for working under constant air flow conditions. All the types of air terminal devices that are used to supply air produce a discharge pattern in the form of a *jet*. After entering room the mixing between room air and supply air takes place. Supply air diffusers typically turn the air stream received from the supply air duct through 90° and discharge it horizontally across the ceiling. The natural tendency of a fluid in motion to cling to an adjacent solid surface is called the *Coanda effect.* It is defined as a apparent of a negative pressure or suction which pulls each layer of a air in the jet towards the ceiling. Since the chilled air is naturally denser it tends to cause the supply air stream to fall away from the ceiling towards occupied zone (negative buoyancy force). If the strength of the Coanda effect is weaker than the negative buoyancy force the supply air jet will ´dump´ or fall away from the ceiling and cause discomfort in the occupied zone. This tends to happen when the air flow rate supplied by the diffuser reduces till the certain point. According to the research and evaluations made this may occur at the supply velocities below 1,5 m s^{-1} [6]. In order to have the required thermal comfort in the room with the VAV system the chosen supply air diffuser must maintain the velocity of the supply air stream from the diffuser with the decreasing supply air flow rates. It means that the device must handle the pressure differences in different working situations.

In order to save energy with the VAV system the supply air temperature must be quite low to handle the heat surplus with less amount of air flow rate. This is however limited parameter in most of the supply air devices due to the occurrence of ´dumping´. The aim of this paper is to introduce the investigations that have been made with the new type of VAV supply air diffuser, which allows supplying the air into room with low temperature (around +15°C) without causing any thermal comfort problems. This diffuser also meets the noise requirements in different air flow rate conditions. The working principle of this diffuser is following. Diffuser varies the airflow rate according to the demand of the room, which is expressed by the room temperature. They contain advanced control- and regulating equipment. Sensors for measuring room temperature, supply air temperature and air flow rate and also presence sensor are built in to the device. If temperature raises above the set point the motor inside starts to open the distance between the lamella plates and gives more air flow in order to keep the room temperature in required level. In the minimum air flow rate the lamellas are nearly closed. Every supply air device can be programmed for two low air flow rates and maximum air flow rate. Programming can be handled by using palm-computer.

A schematic picture of this diffuser is given in figure 2.

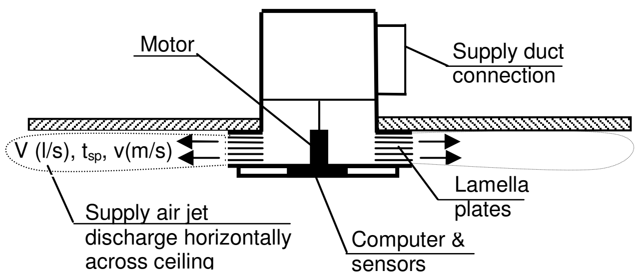

Fig. 2. VAV supply air diffuser

2.3 Investigated building

This new type of VAV supply air diffuser has been tested in several different houses. This report describes the investigations that were carried out in the old building, where reconstructions were made two years ago. During these reconstructions the ventilation system was also changed from CAV (Constant air volume) to VAV system.

Investigated building is a university building in Chalmers in Gothenburg, which was built in 1950-ies. In 2003 reconstructions were made in one part of the building which consists of 5000 m ^{2} office area (5 floors). An air handling system was also rebuilt, except that more than 10 years old air handling unit and duct system was kept the same. Maximum air flow rate of this unit is 5, 6 m^3/s.

The main idea of renewing this old system was to make it from constant air volume system (CAV) to variable air volume system (VAV) in order to save expenses for the energy. Ordinary supply air devices were changed to the new type of variable air volume diffusers (Lindinvent) described above. Variable frequency inverters for fan speed control were installed to both supply and exhaust air fans in order to maintain a specific static pressure in the main duct near the air handling unit. Besides that a cooling coil was installed to the unit for cooling down the supply air during summertime. Cooling coil was connected to the local university cold water system.

Supply and exhaust air flow rates in different floors are balanced by measuring respective air flow rate in the two main supply air branches and exhaust air branch on each floor and regulating the flow rates through the damper on the exhaust air brunch.

Air handling system is working during normal working hours. The supply air temperature from the central unit is ca +14 - +15°C. Because the air distribution ducts are not insulated the supply air temperature rises some degrees in the duct system. During the night time the ventilation system is switched off or running under minimum air flow rates.

Air handling system has also a night cooling. When room temperature gets higher than +21°C and outdoor temperature decreases below +17°C, all the supply air devices will get a signal for full opening and central unit starts. After the room temperature gets lower than +23°C or if the outdoor temperature gets higher than +17°C the ventilation unit stops.

3. RESULTS

3.1 The Results of the energy consumption

Investigated ventilation system has a rotating wheel type heat exchanger with a temperature efficiency ca´ 80 % at a maximum air flow rate. In VAV system a lower air flow rate will increase the temperature efficiency.

In order to save energy for operating fans, variable frequency inverters for fan speed control were installed to both supply and exhaust air fans. Specific static pressure is maintained in the main duct after the ventilation unit.

After reconstructions a cooling coil was installed to the ventilation unit. The main reason for this was to improve the temperature control in rooms. The supply air temperature in this system was +18°C and the cooling effect was certainly not enough to deal with the heat surplus in the rooms. Installing a cooling coil to the system gives an additional need of energy.
Table 1 gives the comparison of the air handling system parameters before and after the reconstructions.

Table 1. Comparison of the air handling system before and after the reconstructions. Temperature efficiency and specific fan power of the VAV system are given as an average over the year. *- to compare two different systems and different indoor climates, the annual cooling energy for CAV system was calculated, although there was no cooling before.

System parameter	**Before rebuilding CAV system**	**After rebuilding VAV system**
Heat recovery -temperature efficiency	75%	av 78%
Maximum air flow rate	6,5 m^3/s	5,6 m^3/s
Operation time	3500 h/year	3500 h/year
Supply air temperature	18°C	15°C
Exhaust air temperature	21°C	21°C
Outdoor climate	Göteborg	Göteborg
Specific fan power- SFP	2,5 kW/m^3/s	av1,2 kW/m^3/s
Necessary annual heat energy	12,5 MWh_{heat}	0 MWh_{heat}
Necessary annual electric energy	57 MWh_{el}	21 MWh_{el}
Necessary annual cooling energy	44,0 $MWh_{cooling}$*	56,7 $MWh_{cooling}$

4.3 Measurements of the thermal climate

Measurements of the thermal comfort and noise were made in different office rooms each floor. Rooms were selected on the basis of having at least one room each floor and also some rooms with two supply air devices in the same room. Selected rooms were typical office rooms. Figure 4 shows a typical room and the location of the measuring instrument.

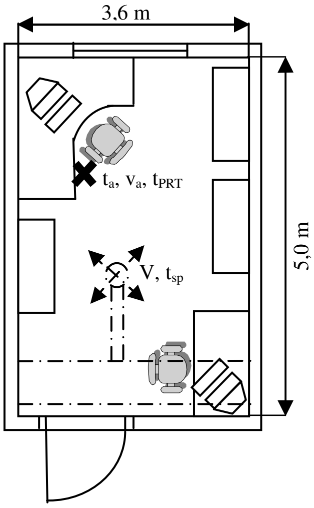

Fig. 4. Typical room and the location of the measuring instrument (location marked with X)

Thermal comfort measurements were made in the conditions where the diffusers were working with maximum air flow rate (40 l/s) and with the minimum supply air temperature, since these are the conditions that may cause the sensation of draught. To measure the diffusers work in conditions, in some measured rooms the diffusers were forced to open in maximum mode by changing the set point of the room with the palm-computer. All the rooms had a set point +23°C.

Measurement instrument “Brüel & Kjaer Model 1213 – Thermal Climate Analyzer” was used for this project. This instrument allows to measure following parameters: room temperature, plane radiant temperature in two opposite directions, air velocity, standard deviation of air velocity, air humidity. These indoor climate parameters were measured during ca 10 minutes time after simulating the diffuser for the maximum mode. Transducers were placed at the level of sitting person’s head (1,1m above the floor) near the working place (See fig. 4).

To avoid difficult and time consuming measurements for calculating mean radiant temperature, measurement was taken with plane radiant temperature sensor in two opposite surfaces, where one was the window direction in every measured room. Assumption was made that other surfaces temperatures in the room do not differ from the measured ones.

Table 2 below describes measurement results in different rooms. The values are given as an average over the measurement period (ca 10 minutes).

Table 2. Thermal comfort parameters in measured rooms. Air temperatures are given as an average over the measurement period. Average air velocity and SD are given as an average over the 3 minutes before last measured minute. Diffusers were working with the maximum mode with supply air rate 40 l/s.

Room nr.	Supply air temp. t_{sp}, °C	Room temp. t_a, °C	Air velocity, v_a, m/s	Standard deviation of air velocity m/s, **SD**
1	14,7	22,7	0,12	0,04
2	15,0	23,4	0,08	0,03
3	14,7	21,4	0,02	0,07
4	15,6	22,6	0,10	0,03
5	15,2	23,5	0,10	0,03

For evaluating the draught rating it can be calculated by using following Eq. (2) [7]:

$$DR=(34-t_a)(v_a-0,05)^{0,62}(0,37\cdot v_a\cdot Tu+3,14) \quad [\%] \qquad (4)$$

where Tu is turbulence intensity and expressed in Eq. (3)

$$Tu=\frac{SD}{v_a} \quad [\%] \qquad (5)$$

In these formulas v_a, local mean air velocity, is calculated as an average air velocity during 3 minutes before last measured minute, because during the last minute a lot of air movement was due to the people entering the room.

Figure 5 below expresses the draught rating in measured rooms.

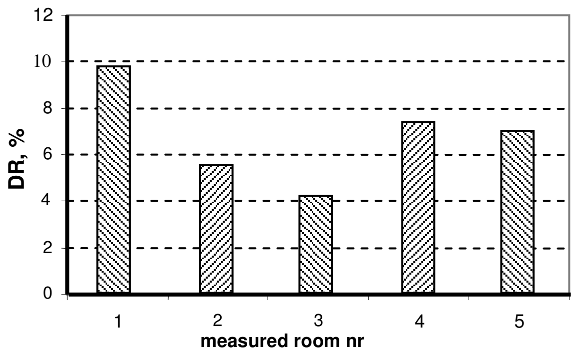

Fig. 5. Draught rating in measured rooms. Calculated according to eq. 2.

According to thermal comfort standard [7], the draught rating DR, should not exceed 15% based on the model of draught for 15 % dissatisfied due to draught. According to the measurement results the DR values did not exceed 10% value in the conditions were supply air flow rate was maximum and supply temperature minimum. This gives the conclusion that investigated new type of variable air volume diffusers can be used with lower supply air temperatures.

4.4 Measurements of the Sound

In order to investigate the sound pressure level generated by VAV device sound measurements in the rooms were carried out. Measurements were made during the full operation of the diffusers (with the maximum airflow rate 40 l/s). Measurement time was ca 10 minutes starting from the device simulation. Measurement instrument "Brüel & Kjaer type 2260, application BZ7206 version 2.1" was used. Sound pressure levels were measured in different frequencies on one third of the octave band.

Problems with the background noise occurred during the measurement time. To evaluate just the sound generated by the VAV device, an assumption was made that it can not exceed the minimum measured sound pressure level in the room. Also to avoid possible influence from the central ventilation unit rooms in the lower floor have only taken in to account. Measurements were made in two rooms in the lower floor and in both rooms the minimum sound pressure level was between 26 dB(A) and 27 dB(A).

The diffusers producing company has made their own laboratory measurements about the sound levels produced by their VAV device. Noise level with different airflow rates and pressure drops dependent on the slot opening is given in figure 8.

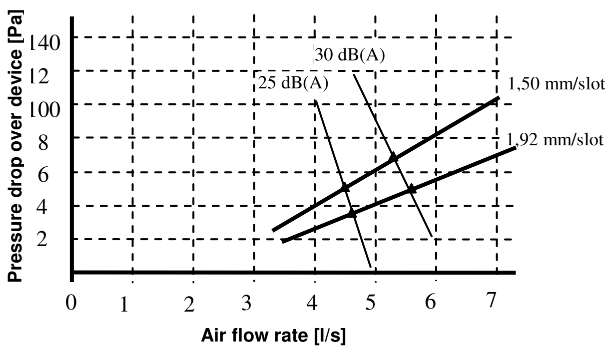

Fig. 8. Laboratory measurements. Noise level with different airflow rates and pressure drops dependent on the slot opening.

According to these laboratory measurements noise from the diffuser should not exceed 25 dB(A) if the airflow rate is around 40 l/s.

4.5 Flow control properties

Every supply air device is programmed for two low air flow rates and maximum air flow rate. If the room is empty and the room temperature is under +23°C, the supply air device is working with the minimum air flow rate 5 l/s. When somebody is in the room the air flow rate increases to 10 l/s. If the room temperature increases over +23°C increases the airflow rate from the diffuser to the maximum 40 l/s in order to keep the room temperature under the desired level.

4.6 General function

Air handling system of this test house is working during normal working hours. The office rooms in this building are quite small (ca´18 m^2) and usually during working hours there is always some extra heat surplus due to office equipment and people in the room that exceeds the heat transmission of the room. The supply air temperature is therefore kept quite low (+15 -+16°) all year around. If the heat surplus of the room is low or there is no heat surplus in the moment at all, the supply air devices work under the minimum air flow rate and the radiator system compensates the cooling effect of the supply air. The supply air diffusers in the rooms are functioning in the way that according to the measurements made, the low supply air temperature is not affecting the thermal comfort of the room occupants. This fact gives big advantage towards energy saving.

Conclusions

A lot of investigations have been made to prove that VAV (variable air volume) system is one of the most energy efficient ways for building air-handling system. Since there is still a problem with the high cost of this system, the possible effort of it should be maximized. While designing a VAV system for a building it is necessary to understand that different parts of this system have a significant role for the total functioning of it. Also costs of the different parts vary a lot. Installing variable frequency inverters for the fan speed control for both supply and exhaust air fans is one of the simplest and probably cheapest part of the total VAV system. Building air distribution parts however demand much bigger investments and they also play a significant role in the total functioning of the whole ventilation system. Therefore they must fulfil certain requirements. Providing good indoor climate with less use of energy is part of it.

In this article a concept of energy savings and improving indoor climate was presented. VAV system with low supply air temperatures gives a big amount of energy savings. This was proven by the investigations and tests made in the reconstructed old building. Lower supply air temperatures also require diffusers that will maintain required indoor climate. The main problems that occur with VAV devices are noise and unsatisfied thermal comfort. Measurements that were made in our test house showed that these new type of supply air diffusers provide good climate control. They work quietly and do not cause any problems with the thermal comfort in the rooms even with low supply air temperatures.

Nomenclature

η_t	temperature efficiency
t_{rec}	supply air temperature after the recovery heat exchanger (°C)
t_o	outdoor temperature (°C)
t_e	exhaust air temperature (°C)
V_d	air flow rate from the diffuser (l/s)
h	diffusers opening (%)
$t_{a,dif}$	average room temperature according to the diffusers data (°C)
t_{sp}	supply air temperature from the diffuser (°C)
t_a	average room temperature
t_{PRT}	average plane radiant temperature
v_a	average air velocity in the measured zone (m/s)
φ	relative humidity, RH %
DR	draught rate in the measured zone (%)
Tu	Turbulence intensity in the measured zone
L_{peq}	equivalent sound pressure level (dB)
L_{Smax}	maximum sound pressure level with slow time weighting (dB)
L_{Smin}	minimum sound pressure level with slow time weighting (dB)
L_{pAeq}	A weighted equivalent sound pressure level (dB(A))
L_{pCeq}	C weighted equivalent sound pressure level (dB(C))

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