Energy Efficient HVAC Systems in Office Buildings – with Emphasis on Air Distribution Systems

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This paper gives examples of performance ratios for analysing the energy efficiency of HVAC systems and subsystems in office buildings, particularly air distribution systems. Energy performance ratios, primarily for design purposes but possible to use at energy audits, are given for the HVAC system divided into 1) the air handling system {(a) air distribution system and (b) air conditioning equipment} and 2) the room system (local equipment) connected via 3) the water distribution system and 4) the heating and cooling supply system (plant). Numerical values of energy performance ratios, energy performance targets are given for the air distribution system.

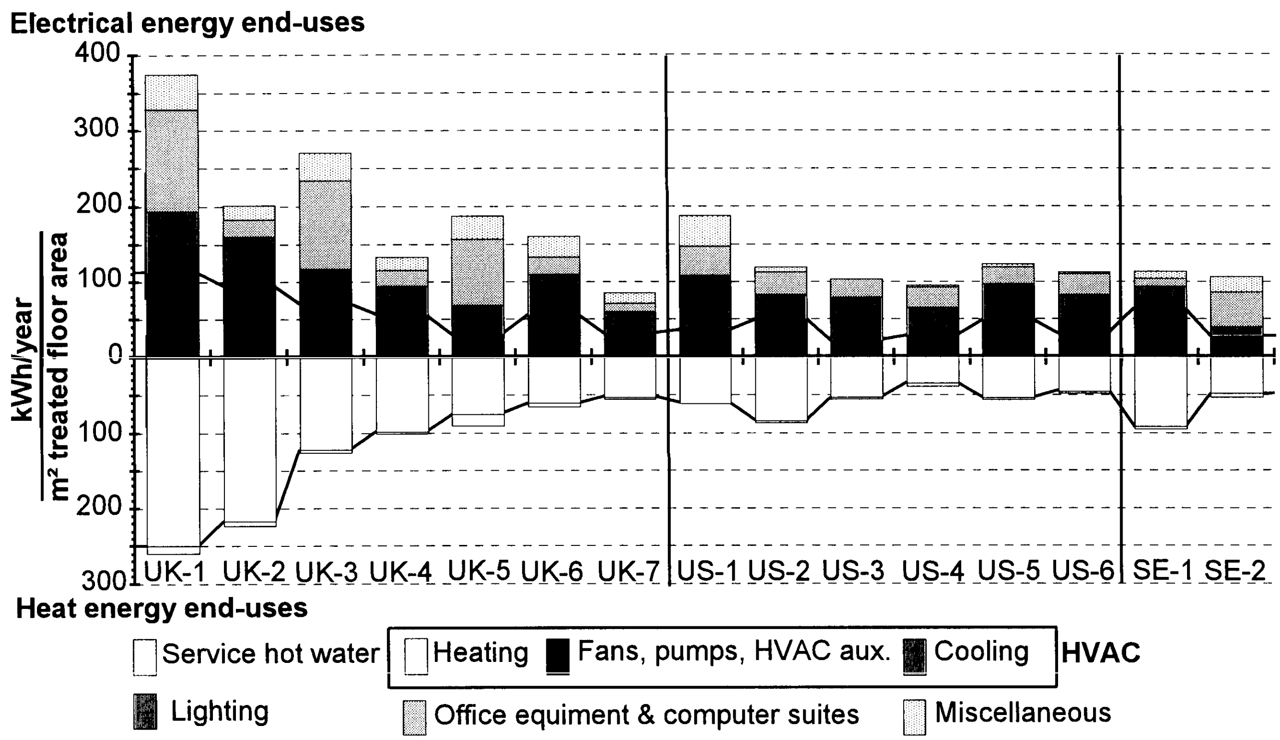
For the air distribution system, the Specific Fan Power *(SFP)* at design conditions for audited Swedish and Danish office buildings (mainly CAV systems) vary typically between 2 kW/(m^3/s) to 4 kW/(m^3/s), which is higher than the prescriptive criterion in ASHRAE Standard 90.1-1989. In audited Danish VAV systems with simple two-speed motor control, the fan annual electrical energy usage is typically between 40-50% of a CAV system; with frequency inverter control the fan power is proportional to the average air flow rate raised to a factor just less than 2 at the design air flow rate. Although for supply air fans the factor is 1.5-2.0. A combination of energy performance ratios and marginal cost analysis is suitable for analysing the impact of stepwise measures thereby improving the energy efficiency of HVAC systems. With input data for a typical Swedish office CAV system the marginal internal rate of return is acceptable (real marginal interest rate >6%) for decreasing the *SFP* to just above 2 kW/(m^3/s).

Introduction

In commercial buildings a substantial part of the energy end-uses is connected to the HVAC system. Figure 1 shows monitored energy end-uses for modern office buildings in the United Kingdom (BRECSU 1991, Jagemar et al. 1994), the USA (Piette 1994[1]), and Sweden (Nilson & Hjalmarsson 1993). The buildings are situated in temperate coastal climates. If necessary, floor areas are recalculated to treated area in accordance with rules of thumb in BRECSU 1991. Electricity inputs to heat recovery chillers in heat pump mode are multiplied with the annual COP (~2.5-3.0) to obtain heat energy end-uses. Most of the buildings are designed to be energy efficient (UK-5-UK-7; USA-1-USA-6). Most striking are the large differences in annual heat energy usage between the typical British office buildings, and the Swedish office buildings that are situated in a colder climate. The large electricity need for office equipment and computer suites in some buildings (UK-1; UK-3; UK-5; SE-2) are mainly due to the computer suites and associated cooling equipment. The HVAC electrical energy end-uses vary considerably between the buildings. A typical prestige British office building (UK- 1) uses a larger amount of electricity for HVAC purposes than the total electrical energy end-uses in building UK-7.

Naturally, nearly all heat energy end-uses for the buildings in Figure 1 are connected to the HVAC system. The HVAC part of total electrical energy end-uses varies between 15-45%. The fans dominate the HVAC electrical energy end-use, which is partly due to the temperate climate that allows for free (economizer) cooling by ambient air. Consequently, the HVAC energy usage, and particularly the energy usage of air distribution systems, is an important energy end-use and design guidance for energy efficient HVAC systems is needed.

This paper defines energy performance ratios that characterise HVAC systems from an energy end-use point of view. It also gives numerical values, energy performance targets for energy efficient air distribution systems. The majority of the performance targets that designate an energy efficient level vary between different



UK-1 British air conditioned prestige office building. Typical energy end-uses.
UK-2 British air conditioned standard office building. Typical energy end-uses.
UK-3 British air conditioned prestige office building. Good Practice energy end-uses.
UK-4 British air conditioned standard office building. Good Practice energy end-uses.
UK-5 British natural ventilated/air conditioned office. Erected 1987. Treated floor area 13,600 m². 4-pipe fan coil system.
UK-6 British air conditioned office. Erected 1980. Treated floor area 23,475 m². Water loop heat pumps, 2 main AHUs.
UK-7 British air conditioned office. Erected 1987. Treated floor area 5,020 m². VAV system. Adjustable speed fans.
US-1 Energy Edge office building. Erected 1988. Treated floor area 32,524 m². VAV system. Heat recovery chillers.
US-2 Energy Edge office building. Erected 1987. Treated floor area 1,984 m². VAV system.
US-3 Energy Edge office building. Erected 1987. Treated floor area 6,664 m². Water loop heat pumps.
US-4 Energy Edge office building. Erected 1988. Treated floor area 2,077 m². CAV system, night flushing.
US-5 Energy Edge office building. Erected 1987. Treated floor area 1,140 m². VVT system.
US-6 Energy Edge office building. Erected 1986. Treated floor area 714 m². CAV system.
SE-1 Swedish office building. Erected 1988. Treated floor area 9.100 m². CAV system & induction units, heat recovery chillers. 25% vacancy (results in lower lighting & office equipment electrical energy end-uses. higher heat energy end-uses).
SE-2 Swedish office building. Erected 1985/85. Treated floor area 18,350 m². VAV system, only computer suite cooling.

Figure 1. Monitored Electrical and Heat Energy End-Uses for Modern Office Buildings in the United Kingdom, the USA (Pacific Northwest), and Sweden

climates, as well as between different HVAC system types. Here the description is mainly concentrated on the air handling and air distribution system. Jagemar et al. 1994 contains a more complete description including other parts of the HVAC system.

Performance Ratios to Express Energy Efficiency

General

Performance ratios for the specific energy end-uses for the building services in a building, or for the building as an entirety, are valuable tools when analysing various solutions regarding energy efficiency. This applies both to the design phase and when evaluating an energy audit of an existing building or a building services system, such as the HVAC system. Energy performance ratios used for analysing the energy efficiency of commercial buildings and their building services systems must fulfil three prerequisites: 1) Distinguish between different types of energy; 2) Distinguish between different energy end-uses; 3) Make comparisons possible between similar buildings. When these energy performance ratios are given numerical values they become energy performance targets characterizing energy efficiency.

The first point must imply a clear distinction between the usage in the building of work (electricity) and thermal energy (heat), respectively. The combined energy usage can be expressed as "total", "primary" or "source" energy usage if different types of energy are given different values, but this is not done here. The focus of this paper is on the annual energy usage of an office building and its HVAC system, not on peak powers (demands). The proposed energy performance ratios cannot be applied directly to HVAC systems incorporating load shifting control strategies, such as cool storage. However, typically a low peak demand results in low annual energy usage, particularly if no load shifting control strategies are used.

The second point proposes first a division between the energy usage of *the HVAC system* and of *the building,* wherein *the building* includes the energy usage of the more "building related" services systems, such as the lighting system or the office equipment. Different energy performance ratios, as independent of each other as possible, can be used for each building services system, as well as for subsystems inside a certain system. The building and the building services systems should be divided into several levels, representing different energy end-uses. Here three levels are presented: 1) The building seen as an entirety including all building services systems, e.g., HVAC-system, lighting system and office equipment; 2) A major building services system in the building, e.g., the HVAC-system or the lighting system; 3) A subsystem to one air more major building services system, e.g., the air distribution system, which is a part of the air handling system and which, in turn, is a part of the HVAC system. To make it easier to distinguish the two latest levels, Figure 2 shows a division of the HVAC system into different subsystem and types of HVAC equipment.

Heating, Ventilating and Air Conditioning System			
Air Handling System			**Room System**
Air Distribution System *Air Handling Units* *Fans* *Ducts* *etc* Controls	Air Conditioning System *Air Heating & Cooling Coils* *etc.* Controls	**(Plant) Water Distribution System** **Heating & Cooling Supply System** Controls	Local Equipment *Baseboard radiators* *Fan coil units* *etc* Controls

Figure 2. Division of the HVAC System into Subsystems and Types of HVAC Equipment

In Figure 2 there is a clear distinction between the central-air-handling system and the local-room system. These two systems are connected through the water distribution system and the heating/cooling supply system (plant).

The third point means that the energy performance ratios must refering to some ground that is common at least for all similar buildings. The most well-known and used common ground is the treated (heated/cooled/ventilated) floor area, which always should be well defined and explicitly given. Energy performance ratios referring to the treated area are useful when comparing entire buildings including all building services systems, but can be deceptive when analysing for example the air distribution system as the air flow rate per treated floor area varies both between buildings and for different design solutions of the same building. On the other hand, different buildings or design solutions for the same building with different types of air conditioning systems, i.e. all-air system, air-water system, or all-water system, can be compared with performance ratios based on the treated floor area.

Performance ratios can refer to factors other than the floor area. One is the design cooling load for the building [kW, Btu/h] (the heat surplus in the building that the HVAC system has to transport out from the building to keep the indoor temperature at an acceptable low level on the design day), which can be used when analysing the HVAC system, as this parameter rates the system. Here again different types of air conditioning systems can be compared for the same building or between similar buildings. A disadvantage with using the cooling load is that it is normally only known in a design situation. When analysing the air handling system and the air distribution system, the design specific electrical fan power, or the specific annual electrical fan energy usage, can refer to the design air flow rate through the system, i.e., the building. If the cooling load is met by an all-air system, this performance ratio is also suitable when analysing the HVAC system in an energy audit.

Figure 3 gives an overview of the main performance ratios that can be used when analysing the HVAC system and its subsystems. As the focus of this paper is on annual energy usage only, these are shown in the figure, except for the air distribution system where the specific electrical fan power is used. A deeper analysis of the performance ratios presented in Figure 3 is made in Jagemar et al. 1994.

Air Handling System–Air Distribution System

Definition of Specific Fan Power and Utilization Factor. As Figure 1 shows, the air distribution system is typically a major electrical energy end-use. In all-air systems, as well as in air-water systems based on

	PERFORMANCE RATIO Design	Audit	Examples of energy end-uses
Building & HVAC System Annual Electrical and Thermal Energy:	kWh/year / kW cool. load	kWh/year / m²	Electrical energy to HVAC system & other building services systems in the building, e.g. lighting, supply of office equip. Thermal energy to HVAC system
HVAC System Annual Electrical and Thermal Energy:	kWh/year / kW cool. load	kWh/year / m²	Electrical energy to fans, pumps, water chillers/heat pumps Thermal energy to boilers, cogeneration equipment etc.
Air Conditioning Equipment and Air Handling System Annual Electrical, Heating & Cooling Energy:	kWh/year / kW cool. load	kWh/year / m³/s	Electricial energy to fans depending on air flow rate control (CAV/VAV) Heating energy to central air heating coils Cooling energy to central air cooling coils
Air Distribution System Design Specific Electrical Fan Power:	kW / m³/s	kW / m³/s	Electricity to fans due to pressure drops in air handling units and duct system, and fan's total efficiency (fan/drive/motor)

Figure 3. Summary of Specific Energy End-Use Performance Ratios that Can Be Used when Analyzing the HVAC System and Its Subsystem

induction units, fans in the central air handling units dominate. Even in fan-coil systems, the fans in the room equipment may use a large amount of electricity as they sometimes run non-stop throughout the year, providing both cooling and heating.

A natural ground to refer the fan electricity usage is the design air flow rate. This yields the performance ratio Specific Fan Power, *SFP,* which typically refers to design conditions. It is important to realise that the *SFP* is *not* a measure of the fan annual energy usage, but rather a measure of the potential of the air distribution system to yield low fan electrical energy requirements. For air distribution systems with short run times, the optimal *SFP*, from an energy-economic point of view, is higher than for systems with longer run times.

In a design situation the efficiency of the total air distribution system is of main interest. In this case the Specific Fan Power, *SFP*, can be defined according to Equation (1) as the sum of the design fan power of all the fans in the air distribution system $\left(\sum \dot{W}_t^{Design}\right)$ over the design air flow rate through the building $\left(\dot{V}_{through\ bld}^{Design}\right)$.

$$SFP = \frac{\sum \dot{W}_t^{Design}}{\dot{V}_{Through\ bld}^{Design}} \left[\frac{kW}{m^3/s}\right] \tag{1}$$

If it is desirable to pressurise the building, the design air flow rate is typically the supply air flow rate, otherwise it is the exhaust (return) air flow rate, which is normally the case in Scandinavian countries.

Figure 4 shows the Specific Fan Power, *SFP* as a function of the sum of the total pressure rise for the supply and exhaust (return) fans, and of the efficiency of the fans (referred to the fan shafts). Typical Scandinavian efficiencies for the fan-motor belt drive ($_{drive}$ = 90%) and the electric motor ($_{motor}$ = 85%-4 poles) have been assumed at fan powers 3-10 kW.

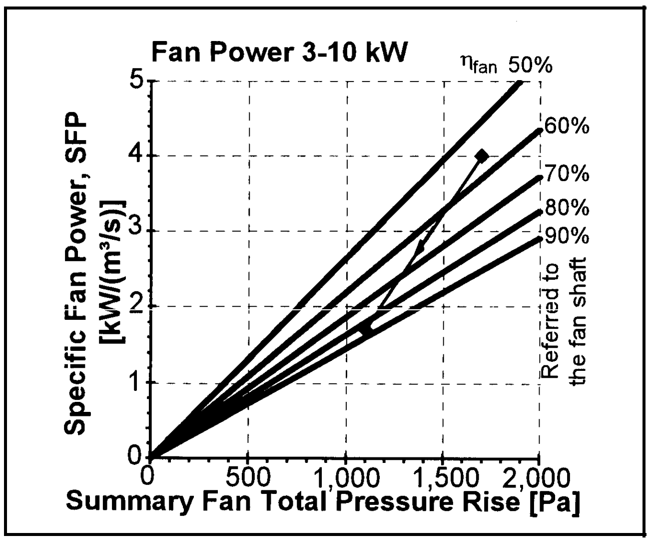


Figure 4. *SFP* Referred to the Design Air Flow Rate Through the Building. The example shows how to reduce *SFP* from 4 kW/(m^3/s) to about 1.7 kW/(m^3/s).

Specific Fan Power According to Standards and Guidelines. In building codes, standards, etc. there exists two ways of stating the Specific Fan Power for different types of air distribution systems, CAV or VAV systems, respectively: (1) *SFP* is expressed for design conditions according to, for example ASHRAE/IES Standard 90.1-1989 (ASHRAE 1989). A VAV system can then be allowed to have a higher *SFP* than a CAV system and still use the same amount of annual electrical fan energy; (2) *SFP* is expressed for different air flow rates that ideally reflects the yearly average conditions according to, for example the Scandinavian voluntary used Guidelines R2 (SCANVAC 1991). Here, the same *SFP* is given for a CAV system at design conditions and for a VAV system at an air flow rate of 80% of the design. The British CIBSE Building Energy Code uses a similar approach.

The Specific Fan Power is given numerical values as a prescription criterion in ASHRAE 1989. The design supply air flow rate should be used. SCANVAC 1991 gives the *SFP* for three different Ventilation and Air-Condition System *(VAS)* quality classes, with an electrically efficient class inside the highest quality class. The largest air flow through the building should be used. The *VAS* denotations can be extended to also include performance ratios for VAV systems such as the utilization factor *u* (compares the annual fan energy usage for a VAV system with that of a CAV system) or the Specific Fan Energy, *SFE*. These performance ratios are defined in Equations (2) and (3).

$$\upsilon = \frac{\sum W_t}{\sum \dot{W}_t^{Design} \cdot \tau} \left[\frac{kWh/year}{kW \cdot h/year} = 1 \right] \qquad (2)$$

$$SFE = \frac{\sum W_t}{\dot{V}^{Design}_{Trough\ bld}} \left[\frac{kWh/year}{m^3/s} \right] \qquad (3)$$

where:

ΣW_t = the annual electrical energy usage for all fans in the air distribution system

τ = the annual run time for the fans

Also included in the *VAS* classes are requirements for measurability and adjustability, cleanability, and airtightness.

Figure 5 compares the requirements of the Specific Fan Power in ASHRAE 1989 and in SCANVAC R2 by means of a diagram with *SFP* as a function of the summarised pressure rise for the supply and exhaust fans and the fans' total efficiency, shown here as areas for radial fans with an impeller with backward and forward curved blades, respectively. Ideal air flow rate control has been assumed when recalculating the requirements in SCANVAC R2 to design air flow rates.

The *SFP* requirements in ASHRAE/IES Standard 90.1-1989 and the VAS 1500 class are quite similar. However, it must be remembered that in most Scandinavian new air distribution systems air-to-air heat exchangers are used instead of economizer dampers. These induce extra pressure drops on both the supply and exhaust air sides which corresponds to an increased *SFP* of 0.2-0.5 kW/ (m^3/s). Consequently, for the Scandinavian systems a *SFP* of about 2 kW/(m^3/s) for CAV systems and 3 kW/ (m^3/s) for VAV systems equals the requirements in Standard 90.1-1989.

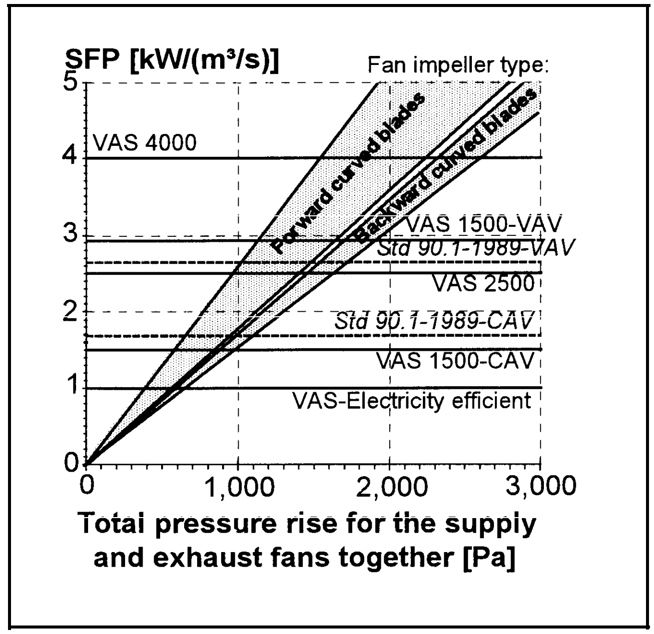


Figure 5. Maximum Allowed Specific Fan Power, *SFP* According to ASHRAE/IES Standard 90.1-1989 and SCANVAC Guidelines R2 *(VAS)*, Respectively

Specific Fan Power According to Audits. Most available audits give the common performance ratio fan annual electrical energy usage per treated floor area, but a few audits show the Specific Fan Power. Figure 6 shows the Specific Fan Power, SW, for 12 monitored Danish office buildings (Olufsen 1993), four monitored Swedish office buildings (Nil son & Hjalmarsson 1993) and four air handling units in an American university building (Lorenzetti & Norford 1992). The two first reports give the *SFP* for individual air handling units (incl. both supply and return/exhaust air fans) as well as for the whole building. All Danish and Swedish whole building data are based on measurements of fan powers and air flow rates. The Danish and American data for individual air handling units are also based on measurements whereas Swedish data are based on design air flow rates and electric motor name-plate powers. For these data the "real" *SFP* typically is about 70-90% of the *SFP* based on design air flow rates and name-plate powers because of the oversizing of the motor and somewhat lower air flow rates than the design.

Figure 6 shows no systematic differences between Sweden and Denmark, although the Danish electrical energy price for commercial buildings is nearly twice that for Sweden. Most of the air handling units have a SW between 2-4 kW/(m^3/s). Comparing Figures 5 and 6 shows that none of the Scandinavian air distribution systems fulfil the

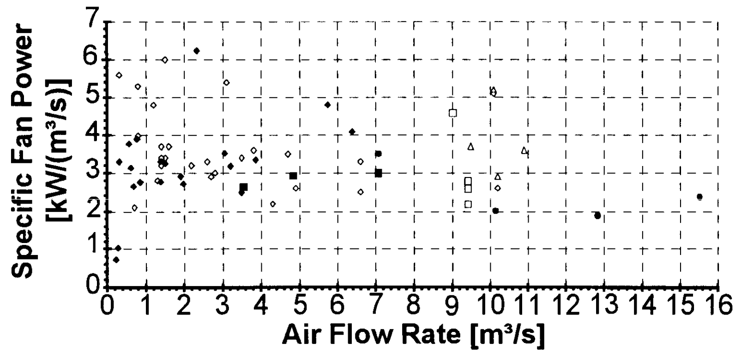


Figure 6. *SFP* for 12 Danish Office Buildings, Four Swedish Office Buildings, and Four Alr Handling Units m an American University Building (VAV). The majority of the air distribution systems are CAV systems.

maximum allowed *SFP* for CAV systems in ASHRAE/IES Standard 90.1-1989. This is natural since they were constructed before this standard was in force. However, the data also partly reflects the concentration on heat energy conservation (oil reduction) in the HVAC design process in Scandinavia.

VA V Systems: Utilization Factor and Adjustable Speed Drives. By changing the air distribution system from having a constant air flow rate (CAV) during the whole run time to a variable air flow rate (VAV), the annual electrical energy usage for the fans can be reduced considerably. Figure 7 shows the utilization factor, i.e. the fan energy usage compared with a CAV system, for the VAV systems audited in Olufsen 1993. Only one system has a continuous variable air flow rate control through a frequency inverter, while the remaining systems have a simple two-speed motor control. In these systems the maximum air flow rate is used between 7-30% of the run time which gives an average air flow rate around 60% of the design. The curves in Figure 7 express different forms of air flow rate control where the fan power is proportional to the air flow rate raised to a power α. At ideal control $\alpha = 3$ and linear control gives $\alpha = 1$. Figure 8 gives some curves for fan power versus air flow rate with frequency inverter control.

Figure 7 shows that for VAV systems with a two-speed motor, the fan energy usages are between 40-50% of a CAV system and the fan power is proportional to the air flow rate raised to about 1.5. For the VAV system with frequency inverter control, the fan energy usage is about 60% compared to a CAV system and the fan power is proportional to the air flow rate raised to about 2.

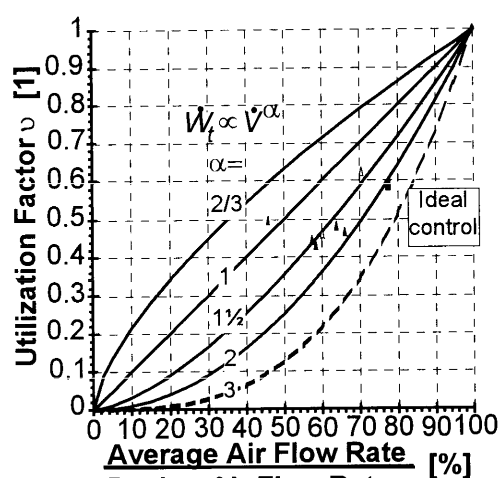


Figure 7. Utilization Factor As a Function of the Annual Average Air Flow Rate for the VAV-Systems in 12 Monitored Danish Office Buildings

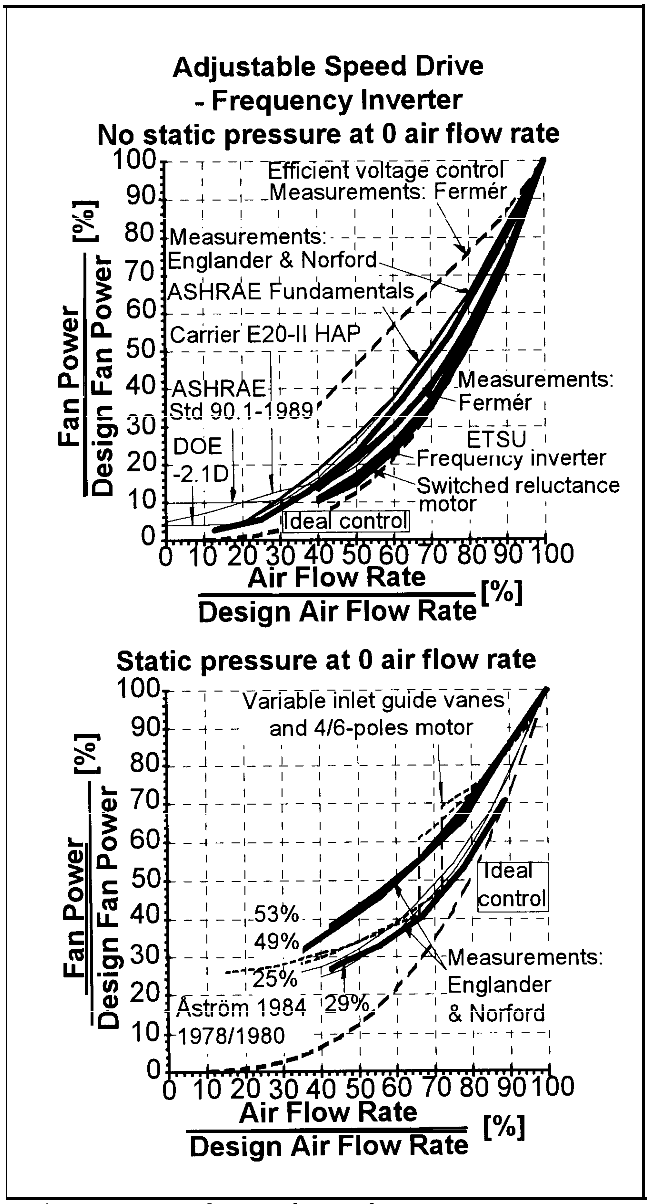


Figure 8. Compilation of Data for Fan Power Versus Air Flow Rate for Variable Speed Drives with Frequency Inverters. Measurements are shown in bold lines.

The fan power versus air flow rate curves will in actual cases not follow the simple lines shown in Figure 7. Figure 8 shows a compilation from the literature of curves for adjustable speed drives with frequency inverters (ASHRAE 1992; ASHRAE 1993; Carrier 1992; ETSU 1992; Englander & Norford 1992; Fermer 1994,[2] Åström 1978/1980 & 1984). The most important factor to consider is whether the VAV fan is a return (exhaust) air fan or a supply air fan. To ensure a proper function of the VAV boxes, the supply air fan has to provide a static pressure at the outlet for zero air flow rate. This considerably distorts the curves from the simple lines presented in Figure 7. A return air fan does not have to ensure a static pressure at the outlet and actual curves follow more closely the ones in Figure 7.

The upper part of Figure 8 shows data for fan power versus air flow rate for fans with no static pressure rise at zero air flow rate (return air fans). If this figure is compared with Figure 7 it can be seen that the power α is between 2 and the ideal 3. Measurements agree quite well with default curves from handbooks etc. As a comparison, measurements are also shown for efficient voltage control of an AC motor and for a switched reluctance motor. The curve for the latter comes very close to the ideal control.

The lower part of Figure 8 shows data for fans providing a static pressure at zero air flow rate. Here field measurements (Englander & Norford 1992) agree rather well with data from a manufacturer (Åström 1978/1980 & 1984). Measurements show that there is a considerable difference between a static pressure ratio of 50% and 25-30%, respectively. In VAV systems in office buildings, the air flow rate seldom falls short of 30-40% of the design and typically the average air flow rate is around 60% (Englander & Norford 1992, Vattenfall 1987). The lower part of Figure 8 shows that at this air flow rate, the power α is between 1.5 and 2. At this average air flow rate there is a small difference between the use of frequency inverter or variable inlet guide vanes with a two-speed motor (4/6 poles). This assumes a static pressure ratio of about 25% and the use of a fan with rather narrow impeller, well suited for inlet guide vane control (Åström 1978/1980 & 1984).

Analysis Order: Increased Fan Efficiency or Decreased Total Pressure Drop. When analysing an air distribution system the question arises where to start the analysis, with the fan or with the air handling unit and the duct system including system effects. These are additional pressure drops that occur in the connections between the fan and duct system due to the non-ideal air velocity fields in the fan inlet or outlet. As a screening criterion, the Specific Fan Power for the Individual fan, *SFPI* can be used. This parameter is defined as the design fan power for the individual fan over the design air flow rate through the fan.

If *SFTI* is reduced more by decreasing the fan's total pressure rise (= system pressure drop) than by increasing the fan's total efficiency, the analysis should start with the air handling unit and the duct system (fittings); otherwise the analysis should start with the fan. Through partial differentiation of the *SFPI* it can be shown that only if the fan's total pressure rise (in [kPa]) is less than the fan's total efficiency (in [1]) the analysis should start by decreasing the pressure drops in the air handling unit and the duct system. Consequently, if *SFPI* > 1 kW/(m^3/s) the analysis should always start by increasing the fan's efficiency. If *SFPI* < 1 kW/(m^3/s) the analysis may start by decreasing the pressure drops, but the profitability of increasing the fan's total efficiency is typically higher than when decreasing the pressure drops. Figure 9 shows *SFPI* as a function of the fan's total pressure rise and fan's total efficiency.

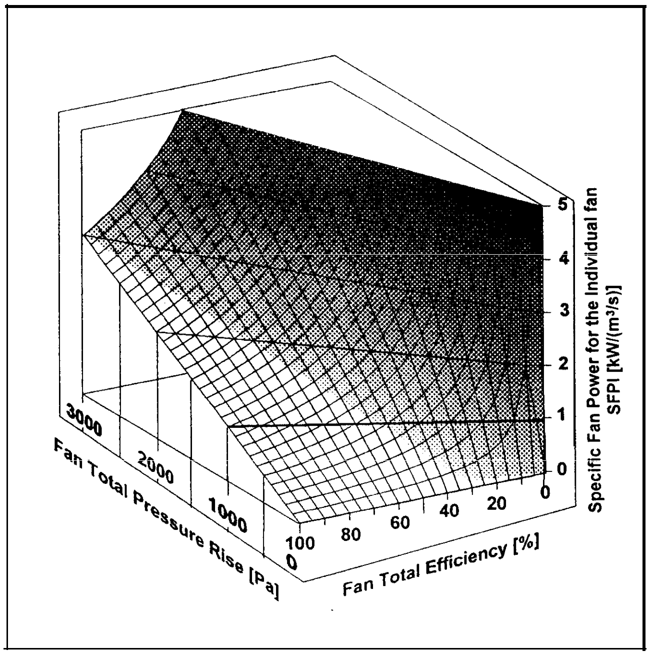


Figure 9. The Specific Fan Power for the Individual Fan, *SFPI* as a Function of the Fan's Total Efficiency and the Fan's Total Pressure Rise

Depending on fan type and size the fan's total efficiency varies from 35% to 80%. Consequently, if the fan's total pressure rise is higher than somewhere between 350 Pa to 800 Pa the analysis should always start by improving the fan's total efficiency as far as this is economically justified. Therefore, it is only for small exhaust ventilation systems that it can be considered suitable to start the analysis with the air handling unit and the duct system instead of with the fan.

After the analysis of the fan, the marginal profitability of the measures on the air handling unit and the duct system that result in the largest decrease in the specific fan power should be analysed. By comparing the impact on the total pressure drop of possible measures, the following analysis order is obtained: 1) Change to a centrifugal fan with higher efficiency: either *fan, transmission* or *motor* efficiency. 2) Improve the air flow conditions in the connections between the fan and the duct inlet and outlets, decrease the *system effects. 3)* Decrease the pressure drops of components in the central air handling unit, e.g. by selecting lower face velocities. 4) Decrease the pressure drops of duct fittings. 5) Decrease the air velocity in straight air ducts.

Performance Ratios Combined with Marginal Cost Analysis

General

When considering the profitability of design changes in HVAC systems two approaches are possible, optimisation or marginal cost analysis, respectively. Here marginal cost analysis is promoted because it is suitable when analysing stepwise improvements and very clearly shows the profitability limit, particularly if the marginal internal rate of return method is used. If all parts of the system are analysed according to the marginal profitability of various improvements, this results in a system solution where all parts are equally strong from an economic point of view. This solution will of course be equal to the one obtained by an optimisation process.

When carrying out a marginal cost analysis of a measure that decreases the running costs, two different criteria may be used 1) the profitability, expressed as pay-back time or internal rate of return, and 2) the maximum allowed investment, which is the present value of the decrease of the annual marginal running costs. The last method is valuable as a screening criterion in an early design stage. Mostly it is relatively easy to determine if the real investment is below that allowed and, consequently, if it is worth analysing the measure's profitability closer. The method of maximum allowed marginal investment is applied to fans and duct systems in Jagemar 1993, and it is further elaborated in Jagemar et al. 1994.

The marginal cost analysis of the HVAC system or a subsystem always starts with the design lay-out that has the lowest investment but which still fulfils the "basic" requirements, e.g. giving the required outdoor air flow rate needed to fulfil the indoor air quality requirements, or a supply air flow rate and a supply air temperature to be able to meet the cooling load in the building and thereby fulfilling the indoor thermal requirements. This system with the lowest investment typically has the highest annual operation costs. Some experience is needed to identify this baseline system layout, but quite often the simplest layout is found to be an appropriate baseline. When analysing HVAC equipment, the baseline in many countries can be chosen as equipment that only just fulfils national codes and standards.

Air Handling System–Air Distribution System

Example of the Marginal Internal Rate of Return when Decreasing the Specific Fan Power. This section applies, as an example, marginal cost analysis on selection of CAV air handling units (AHUs) leading to decreased Specific Fan Power. Doing this, two different approaches are possible, either changing to more efficient fans at a certain AHU size, or increasing the AHU size using the most efficient fans at each size. The upper part of Figure 10 shows the marginal internal rate of return for increasing the AHU size with the most efficient radial fans (large impeller with backward curved blades) at each size. The lower part of Figure 10 shows the marginal internal rate of return for changing to more efficient fans at the smallest AHU size with an air heating coil face velocity 3.3 m/s, a common Scandinavian rule selecting an AHU size. The input data are typically valid for a Swedish office building. However, the use of list prices is doubtful as normally large discounts are given on these prices, perhaps 25-50% depending on the size of the project. Consequently, the marginal profitability is underestimated in Figure 10.

Using the most efficient fans in each AHU size, there is an acceptable marginal profitability only for changing from the smallest possible AHU size to the next smallest one {size 7, *SFP* = 2. 6 kW/(m^3/s)}. If instead the efficiencies of the fans are increased for the smallest AHU size with an air heating coil face velocity <3.3 m/s (size 8), all these changes may be profitable if real interest rates are used (> 6%). When changing to an impeller with backward curved blades the electric motor is decreased one size, thereby giving a very high marginal internal rate of return. The decrease in motor size is due to the fact that a fan with an impeller with backward curved blades has a certain maximal power requirement at a certain speed, whereas this is not the case with forward curved blades.

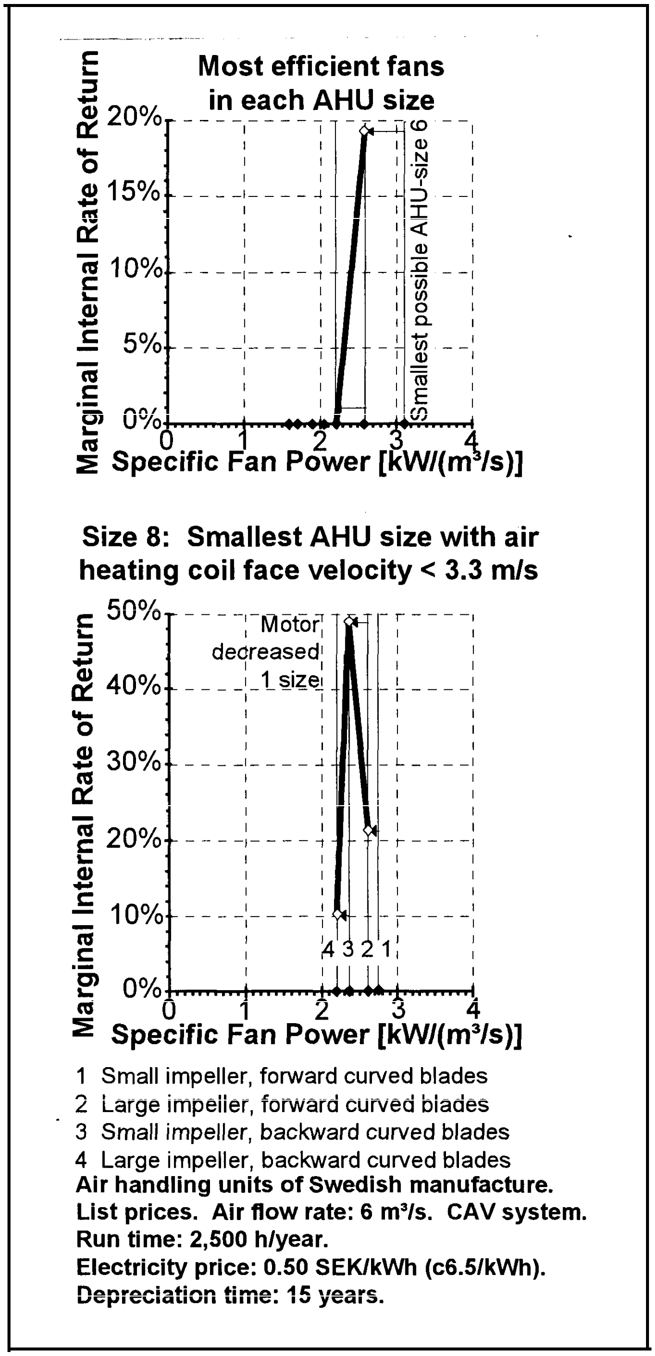


Figure 10. Marginal Internal Rate of Return for Increased AHU Size, or More Efficient Fans at a Fixed AHU Size, as a Function of the Specific Fan Power for a Typical Office CAV System in Sweden

Conclusions

Fans in air distribution systems are a major HVAC electrical energy end-use in office buildings. The performance ratio, Specific Fan Power *SFP* is suitable to characterise the possibilities of the air distribution system to ensure a low fan annual energy usage, but it is not a measure of it. In monitored Scandinavian office buildings, the *SFP* varies between 2 to 4 kW/(m^3/s). A marginal cost analysis shows that an economically justified *SFP* for CAV systems in Swedish office buildings is around 2-2.5 kW/(m^3/s). This means that the air distribution system should fulfil the VAS 2500 quality class in the Scandinavian SCANVAC Guidelines R2. In audited Scandinavian office buildings with VAV systems the average air flow rate is around 60% of the design that agrees with HVAC design default values. The use of simple two-speed motors results in a utilization factor around 0.45, i.e., an annual fan energy usage of 45% of a CAV system. Both measurements and handbook data show that for adjustable speed drives with a frequency inverter the fan power is proportional to the air flow rate raised to 2.0-2.5, presupposed that the fan does not provide a static pressure at zero air flow rate (return air fan). If a supply air fan provides a static pressure ratio of around 25% the curve is distorted, but at an average air flow rate of 60% of the design the fan power is proportional to the air flow rate raised to 1.5-2.0.

Endnotes

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