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Technical Note AIVC 65

Recommendations on Specific Fan Power and Fan System Efficiency

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Disclaimer

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International Energy Agency (IEA)

The IEA was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty-four IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D).

Energy Conservation in Buildings and Community Systems (ECBCS)

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use in buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods as well as air quality and studies of occupancy.

The Executive Committee

Overall control of the programme is maintained by an Executive Committee, which not only monitors existing projects but also identifies new areas where collaborative effort may be beneficial.

To date the following have been initiated by the Executive Committee (completed projects are identified by *):

1 Load Energy Determination of Buildings *
2 Ekistics and Advanced Community Energy Systems *
3 Energy Conservation in Residential Buildings *
4 Glasgow Commercial Building Monitoring *
5 Air Infiltration and Ventilation Centre
6 Energy Systems and Design of Communities *
7 Local Government Energy Planning *
8 Inhabitant Behaviour with Regard to Ventilation *
9 Minimum Ventilation Rates *
10 Building HVAC Systems Simulation *
11 Energy Auditing *
12 Windows and Fenestration *
13 Energy Management in Hospitals*
14 Condensation *
15 Energy Efficiency in Schools *
16 BEMS – 1: Energy Management Procedures *
17 BEMS – 2: Evaluation and Emulation Techniques *
18 Demand Controlled Ventilation Systems *
19 Low Slope Roof Systems *
20 Air Flow Patterns within Buildings *
21 Thermal Modelling *
22 Energy Efficient communities *
23 Multizone Air Flow Modelling (COMIS)*
24 Heat Air and Moisture Transfer in Envelopes *
25 Real Time HEVAC Simulation *

26	Energy Efficient Ventilation of Large Enclosures *
27	Evaluation and Demonstration of Residential Ventilation Systems *
28	Low Energy Cooling Systems *
29	Daylight in Buildings *
30	Bringing Simulation to Application *
31	Energy Related Environmental Impact of Buildings *
32	Integral Building Envelope Performance Assessment *
33	Advanced Local Energy Planning *
34	Computer-aided Evaluation of HVAC Systems Performance *
35	Design of Energy Hybrid Ventilation (HYBVENT) *
36	Retrofitting of Educational Buildings *
36 WG	Annex 36 Working Group Extension 'The Energy Concept Adviser' *
37	Low Exergy Systems for Heating and Cooling of Buildings *
38	Solar Sustainable Housing *
39	High Performance Insulation systems (HiPTI) *
40	Commissioning Building HVAC Systems for Improved Energy Performance *
41	Whole Building Heat, Air and Moisture Response (MOIST-EN)
42	The Simulation of Building-Integrated Fuel Cell and Other Cogeneration Systems (COGEN-SIM)
43	Testing and Validation of Building Energy Simulation Tools
44	Integrating Environmentally Responsive Elements in Buildings
45	Energy-Efficient Future Electric Lighting for Buildings
46	Holistic Assessment Tool-kit on Energy Efficient Retrofit Measures for Government Buildings (EnERGo)
47	Cost Effective Commissioning of Existing and Low Energy Buildings
48	Heat Pumping and Reversible Air Conditioning
49	Low Exergy Systems for High Performance Buildings and Communities
50	Prefabricated Systems for Low Energy Renovation of Residential Buildings
51	Energy Efficient Communities
52	Towards Net Zero Energy Solar Buildings
53	Total Energy Use in Buildings: Analysis & Evaluation Methods
54	Analysis of Micro-Generation & Related Energy Technologies in Buildings

Annex 5: Air Infiltration and Ventilation Centre (AIVC)

The Air Infiltration and Ventilation Centre was established by the Executive Committee following unanimous agreement that more needed to be understood about the impact of air change on energy use and indoor air quality. The purpose of the Centre is to promote an understanding of the complex behaviour of air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock.

The Participants in this task are Belgium, Czech Republic, Denmark, France, Greece, Japan, Republic of Korea, Netherlands, Norway and United States of America.

Contents

Scope

This publication explains the principles of designing efficient ventilation systems with low fan power and hence little fan noise. The main topics are:

- Definition, measurement, and rating of specific fan power and fan system efficiency
- How the design of the air handling unit, distribution system, and controls influence pressure losses and fan system efficiency

It is mainly aimed at HVAC professionals (designers, contractors, manufacturers, and maintenance staff), serving as either reference work or educational material. It will also interest building authorities and other decision makers in the construction industry.

The AIVC welcomes comments to this publication, and will take these into account in future editions.

Abbreviations

AHU: Air handling unit (contains fans etc.)
ATD: Air terminal device (e.g. supply diffuser)
CAV: Constant air volume flow rate
FEG: Fan efficiency grade (see page 17)
FMEG: Fan & motor efficiency grade (see page 12)
HVAC: Heating, ventilation & air-conditioning
LCC: Life cycle costs
MEPS: Minimum Energy Performance Standard
SFP: Specific Fan Power (see §2.2)
SPR: Static pressure reset control (see §2.2.3)
VAV: Variable air volume flow rate
VFD: Variable frequency drive (electronic)
VSD: Variable speed drive (generic)

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1 Summary

Energy use for fan operation can be significantly reduced by a 3-flanked approach:

(**1**) The first step is prudent sizing of ventilation rates by minimizing the demand (e.g. low-emission building materials, passive cooling design), and by utilizing efficient air distribution. The latter reduces unnecessary over-ventilation by use of airtight ductwork, careful choice of room airflow principles (i.e. minimizing short-circuiting), and controls for demand-control of flow rate.

(**2**) Perhaps the most important measure is to minimise flow resistance, and hence fan pressure. This is achieved by aerodynamic design of fan inlets/outlets and ductwork layout (including optimal location of plant rooms and duct risers, to reduce duct length), liberal sizing of components in the duct system, and increasing AHU cabinet size, but without oversizing the fan system.

(**3**) Optimize efficiency of the fan system, including the fan, drive, motor, and variable speed drive (i.e. minimize total 'wire-to-air' losses). Oversizing must be avoided, since fan efficiency can decrease significantly if the combination of airflow and pressure rise is not near the combinations giving peak efficiency. Motor and drive efficiencies can also decrease rapidly at low loads. Thus, oversizing and load diversity are key factors affecting system efficiency.

These three measures are far more important than any exploitation of natural driving forces, in climates where heating or cooling is needed. This guide focuses on points (2) & (3).

Besides reducing energy use, energy-efficient systems are generally less noisy than inefficient systems.

To achieve these potential savings, building developers/owners must dictate and verify fan power performance specifications. All countries should have building regulations set limits on fan power, and establish inspection/auditing schemes that include spot checks of fan power. Furthermore, there is a need for simpler calculation tools for calculating duct system pressure loss and specific fan power in buildings, and easy commissioning guidelines.

This report gives summary recommendations at the end of each section, in pink frames like this.

2 Specific Fan Power – Definitions and requirements

2.1 The significance and consequence of fan energy consumption

Fans represent an enormous potential for energy savings to reduce CO_2 emissions. Fans are among the largest single users of motive-power energy, constituting approx 22% of total energy use by motors [21], Fans use 7~8% of energy within in Sweden [24], and approx 9.3% of energy in the UK [20]. Process industry and buildings account for roughly equal parts of the total fan energy use. In comparison, lighting stands for about 10 % of the EU's electricity consumption.

As the building mass becomes renewed with higher indoor climate standards, fan energy consumption could potentially double in the course of 15~20 years unless building regulations are enforced to limit fan power. This electrical energy cannot easily be substituted by low quality renewable energy sources.

An audit of nearly 500 balanced ventilation systems in Sweden in 1995 indicated an average Specific Fan Power (SFP[1]) of 3 kW/(m³/s). Studies in other countries have shown similar or higher values [24]. At the other end of the scale, hybrid ventilation systems[2] can use less than 0.1 kW/(m³/s); see page 15.

An example: In cold climates, the energy consumption of fans in modern commercial buildings can constitute 15~20 % of the building's total energy use. Figure 1 shows the breakdown of total energy use in a typical existing office building in Norway. Assuming a typical ventilation rate of 10 m³/h per m² floor, and a typical SFP of 3.4 kW/(m³/s), operating 3000 hrs/yr, the fan energy accounts for:

$$10\frac{\text{m}^3/\text{h}}{\text{m}^2}\times\frac{1\,\text{h}}{3600\,\text{s}}\times 3.4\frac{\text{kW}}{\text{m}^3/\text{s}}\times 3000\frac{\text{h}}{\text{yr}} = 28\frac{\text{kWh}}{\text{m}^2\,\text{yr}}$$

which is 17% of the total energy use of an efficient (lower quartile) existing office building with 70 % heat recovery efficiency, and a total energy use of approx 165 kWh/(m² yr).

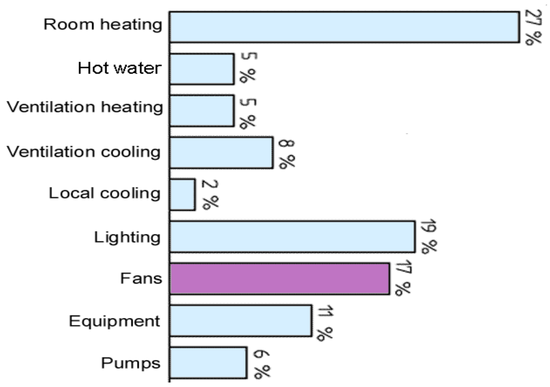


Figure 1: Approximate breakdown of typical energy use in a Nordic office. [Fig. © SINTEF]

Of the 3 components in Figure 1 related to ventilation (i.e. fan energy, ventilation heating & cooling), fan energy is the largest.

The supply fan energy ends up as an internal heat gain in the building, whilst the exhaust fan energy is lost to the outside. Thus 14 kWh/m²yr of fan energy ends up in the building as a heat gain, which is useful in the heating season but causes a cooling load during the cooling season. In cold climates, roughly 60%/20% of the annual supply fan energy is used during the heating/cooling seasons respectively. Of the 60% that is used during the heating season, only 45% ends up as useful space heating, while the remaining 15% is 'lost' if the heat exchanger efficiency is controlled[3] to limit the supply temperature to, say, 18 °C during winter. In warmer climates, a larger portion of the annual fan energy ends up as a cooling load than shown above. Proper evaluation of all these factors should be calculated in each specific case using building energy simulation software.

Recap: Energy use for electrical fan operation is significant and depends on flow rate, operating hours, flow resistance and fan system efficiency. Specific Fan Power (SFP) is a useful measure of these factors. Fan energy use can exceed the energy use for climatising (preheating & cooling) in AHUs with heat recovery. Only part of this energy is recouped for space heating. Moreover, fans use electricity, which is the highest quality form of energy resource[4].

[1] See §2.2 for definition of SFP
[2] Hybrid ventilation = Fan-assisted natural ventilation

[3] By a heat exchanger bypass, or slower rotation speed of rotary heat exchanger
[4] Electricity is the most versatile form of energy and can be converted more efficiently to other energy forms than heat, chemical and mechanical energy.

2.2 **Definition and calculation of SFP**

2.2.1 Overall definition

Specific Fan Power (SFP) is a useful parameter for quantifying the energy-efficiency of fan air movement systems. SFP is a measure of the electric power that is needed to drive fans, relative to the amount of air that is circulated [1]. It is not constant, but changes with both air flow rate and fan pressure rise.

SFP for a given system and operating point[2] is measured as follows:

$$SFP = \frac{\Sigma P}{q_v} \quad [\mathrm{kW/(m^3/s)}] \qquad (1)$$

where

ΣP = sum of all fan powers [kW]

q_v = Gross amount of air circulated [1] [m³/s]

For unbalanced ventilation systems, q_v is the largest of supply and exhaust air flow rates[3]. SFP is further defined in Appendix D of EN 13779.

Dimensional analysis reveals that SFP can be expressed in the following equivalent units:

$$[SFP] \equiv \frac{kW}{m^3/s} \equiv \frac{W}{\ell/s} \equiv \frac{kJ}{m^3} \equiv kPa \qquad (2)$$

2.2.2 Relation to pressure drop and system efficiency

As shown in Eq.(2), SFP can be expressed in units of pressure, since pressure is a measure of energy per m³ air. Eq.(3) below shows the relationship between SFP, fan pressure rise, and fan system efficiency. In the ideal case of a lossless fan system (i.e. η_{tot}=1) the SFP is exactly equal to the fan pressure rise (i.e. total pressure loss in the ventilation system).

$$\eta_{tot} \cdot SFP = \Delta p_{tot} \qquad (3)$$

where

Δp_{tot} = fan total pressure rise, which is equal to the drop in total pressure through the entire ventilation system, from the outside air, including ductwork, AHU, and air transport inside the building, and back to outside [kPa]

η_{tot} = fan system's overall efficiency (combined efficiency of all the components in the fan system) $[0<\eta_{tot}<1]$.

The fan system efficiency (η_{tot}) is the fraction of the electrical power that results in useful driving pressure to transport the air in the ventilation system. See page 12 for more details.

Fan system efficiency is not constant. At low flow rates or low pressures, it falls off sharply from peak efficiency because the motor, belt drive and VFD efficiencies all decrease substantially at low loads (< 40% load). See page 19 for more details.

The pressure drop between two points along the flow path can be expressed as:

$$\Delta p_{12} \approx k \cdot v^n \quad [\mathrm{Pa}] \qquad (4)$$

where

k = constant

v = air speed [m/s]

n = exponent, $1 \le n \le 2$:
(n = 1 for wholly laminar flow,
n = 2 for wholly turbulent airflow)

Modern ventilation systems generally have predominantly turbulent flow in the duct system (n≈2), and predominantly laminar flow through high-pressure-loss AHU components with, such as heating & cooling coils, exchangers, & filters (typically n≈1.4). The total system pressure drop (Eq.4) therefore has an exponent somewhere between n=1.4 and n=2. In systems with low-pressure drop ductwork, the AHU components are the dominating pressure loss, and n will be closer to 1.4 than 2.

For a ventilation system for which half of the total pressure loss occurs in the AHU at maximum flow rate, we have n≈1.65. According to Eq.(4) a 50 % reduction in flow rate ($v = 0.5\, v_{max}$) more than halves the total pressure drop ($\Delta p = 0.375\, \Delta p_{max}$). However, since the fan system efficiency (η_{tot}) decreases at low flow rates, the SFP does not decrease at the same rate

[1] Gross amount of air circulated via the fan(s), by the combined effect of all present driving forces (mechanical and natural)

[2] Operating point: Combination of flowrate & pressure rise

[3] Imbalance of the supply and extract flow rates causes infiltration or exfiltration through the building envelope. The amount of air renewal is the largest of the two, assuming there is no leakage in the AHU (see Figure 8)

as pressure drop. The change in η_{tot} is taken into account in Eq.(6) below.

Also, in systems with flow control devices such as VAV dampers or demand-controlled ATDs, Eq.(4) above is no longer a simple function of flow, but depends on the state of the control points. For example, in the extreme case of controlling fan speed to maintain a constant static pressure rise, Δp_{12} is constant and independent of flow rate. More commonly, VAV control systems maintain a constant static pressure in the duct nearer the ATDs, far from the fan. In this case, the relationship between flow rate and fan pressure rise looks like the system curve in Figure 25 (page 17), which has a non-zero static pressure at zero flow rate. Eq.(6) and Figure 2 below demonstrate the effect of different VAV control methods on part-load SFP.

2.2.3 SFP with variable flow rates

If the ventilation system has different operating points (combination of flow rate and pressure drop) at different times of the year, the annual average SFP can be calculated thus:

$$\overline{SFP_e} = \frac{\sum_{i=1}^{N}\left(\Sigma P_i \, \Delta t_i\right)}{\sum_{i=1}^{N}\left(q_{v,i} \, \Delta t_i\right)} = \frac{\sum_{i=1}^{N}\left(SFP_{e,i} \, q_{v,i} \, \Delta t_i\right)}{\sum_{i=1}^{N}\left(q_{v,i} \, \Delta t_i\right)} \qquad (5)$$

Where there are N different operating modes, each with a duration of Δt_i hours.

There are two ways to calculate part-load SFP:

- Use Eq.(5) if the flow rate and pressure drop is known for the different operating points, and the AHU vendor can calculate/provide SFP values for these operating points.
- Or, for ventilation systems for which the AHU manufacturer has documented only SFP at the design operating point, you can estimate the SFP at each part-load operating point as a function of the air flow rate reduction factor (r). The following generic equation is valid for $0.2 \le r \le 1.0$ [27]:

$$\frac{SFP_{\text{part load}}}{SFP_{\text{max load}}} \approx a + b\,r + c\,r^2 + d\,r^3 \qquad (6)$$

Coefficients for use in Eq.(6):

	a	***b***	***c***	***d***
Poor	1.0	0.0	0.0	0.0
Normal	1.0547	–2.5576	3.6314	–1.1285
Good	0.5765	–1.5030	2.6557	–0.7292
Ideal	0.2869	–0.8836	1.9975	–0.4008

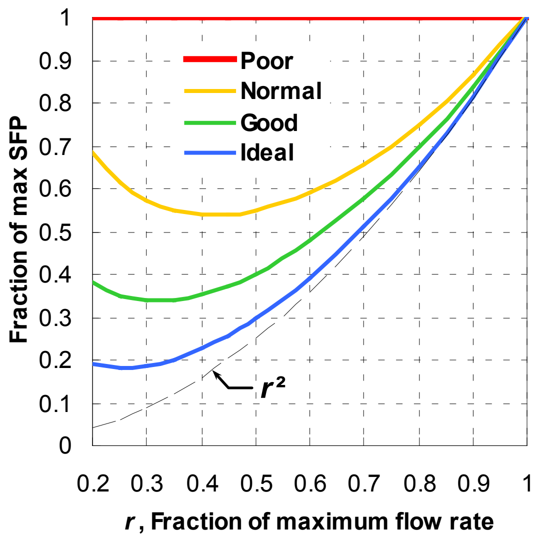


Figure 2: Illustration of Eq. (6) for Poor, Normal, Good, and Ideal systems

- ‘***Poor***’ represents systems with poor efficiency at part load. This includes mostly traditional methods that are now outmoded, such as inlet vane dampers, discharge dampers, variable-pitch fans, and inefficient VSDs such as triacs. The efficiency of some of these systems varies greatly; some may be worse or better than the ‘poor’ curve. It also represents VAV systems for which the fan speed is controlled to maintain a constant fan pressure rise, irrespective of flow rate.
- ‘***Normal***’ represents systems for which the fan pressure drops marginally as flow rate is reduced. This includes VAV systems with the fan speed controlled to maintain a constant static pressure towards the end of the main duct (Figure 3).
- ‘***Good***’ represents systems for which the fan pressure decreases with flow rate. This includes best-practice VAV systems with fan speed regulated by a VFD with a typical Static Pressure Reset controller (SPR, also known as an ‘optimizer’; see Figure 4). SPR constantly tries to minimize duct system resistance by ensuring that the VAV

damper(s) along the present critical path[1] are fully open. VFD controlled AC fans sized <3.7 kW cannot fall in this category, irrespective of pressure control scheme, because these small inverter VFDs have high losses.

- '***Ideal***' represents real systems with efficient VSDs and where the fan pressure falls ideally at low flow rates. This includes VAV systems with perfect SPR control (i.e. 100% open control dampers along on the critical path), or reducing fan speed in CAV systems with fixed duct components (constant *k*-value). For example, night time operation of a CAV system with a flow rate of 20% (r = 0.2) will reduce the SFP to about 19% of $SFP_{max\ load}$. AC fans sized <15 kW cannot fall in this category, irrespective of pressure control scheme, due to higher losses in their VFDs.

The '*ideal*' curve is close to the hypothetical performance curve r^2 (shown as a dashed black line in Figure 2), which assumes zero system losses, and a duct system pressure drop as described in Eq.(4) with an exponent of n=2 for wholly turbulent flow. The hypothetical system's fan power is thus proportional to r^3.

The coefficients in Eq.(6) assume typical ductwork and AHU components, and accounts for the reduction in fan system efficiency (fan, motor & VFD) at low loads.

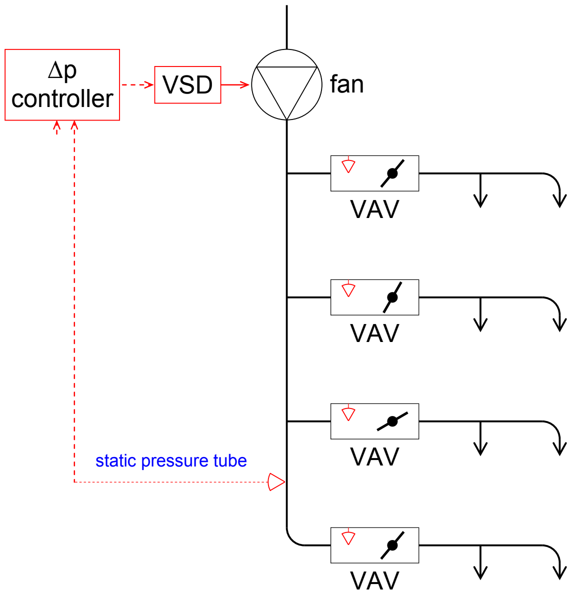


Figure 3: Illustration of constant static pressure control. Red denotes control system; black denotes duct system. The critical path VAV damper is in max position only at times of maximum flow rate demand. [Fig. © SINTEF]

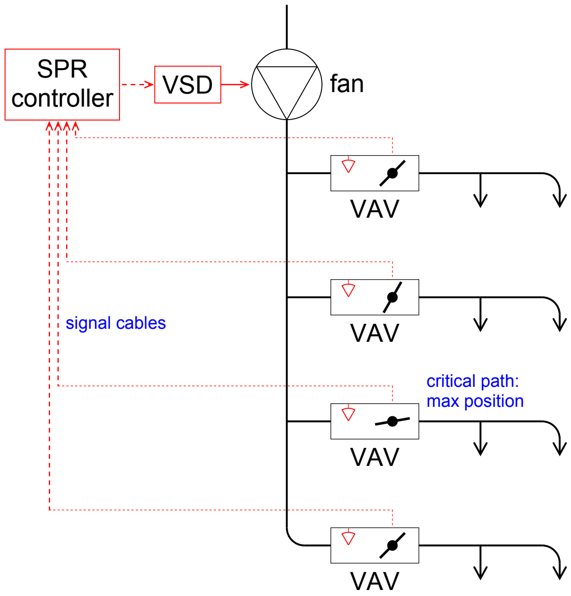


Figure 4: Illustration of SPR control. At any time, at least one VAV balancing damper is in max position (the critical path). Dampers cannot be 100% opened due to need for control authority, i.e. to prevent excessive servo motor wear due to 'hunting'

[1] See page 29 for definition of 'critical path'

Figure 5: A small Variable Frequency Drive (VFD) [Fig. © C.J.Cowie, 2005]

2.2.4 SFP for different system scales

2.2.4.1 SFP_i for individual fans

SFP_i for individual fans is defined by Eq. (1) and (3), which are equivalent.

2.2.4.2 SFP_{AHU} for individual AHUs

For a conventional balanced AHU with 2 fans, SFP_{AHU} should normally be calculated by Eq.(1), but can alternatively be estimated as the sum of the SFPs of the individual fans (SFP_i) thus:

$$SFP_{AHU} \approx SFP_{i,\text{sup}} + SFP_{i,exh} \qquad (7)$$

where
i = individual fan
sup = supply system
exh = exhaust system

This expression is mathematically correct only if the two fans have the same airflow rate. If the air flows differ, then Eq.(1) should be used, with the gross flow rate in the denominator. The gross flow rate is the largest of the true supply and extract flow rates (not the fresh air and exhaust flow rates). Any recirculation/leakage in the AHU should be subtracted from the flow rates. See Figure 8 (page 10)

AHU manufacturers generally measure & document $SFP_{v,AHU}$ (suffix v for verification) with new air filters, dry heat exchangers, and cooling coils. This facilitates easy verification by field-tests after acquisition. If the fans are fed power via externally mounted variable frequency drives (VFD) from another manufacturer, then the power supply is often measured between the VFD and the fan, such that $SFP_{v,AHU}$ ignores VFD losses[1]. HVAC design engineers should therefore check in each case whether the AHU manufacturer has included VFD losses as part of $SFP_{v,AHU}$. $SFP_{v,AHU}$ is normally documented for each duty point that the customer (HVAC designer) specifies, i.e. at design maximum flow rate and possibly also at one or more reduced air flow rates.

Manufacturers should also document $SFP_{e,AHU}$ (suffix e for energy), for use in energy performance calculations. It reflects the average operating conditions over time, i.e. time-averaged pressure drop through the filters (due to build-up of collected dust) and heat exchangers (due to occasional condensation). VSD losses are included, if present (external or internal). Average filter pressure drop can be estimated as the mean of the start and final pressure-drops, where the latter depends on filter change frequency, see Eq.(8). For larger units, the final pressure-drop at design flow rate is typically 250 Pa for F5-F7 filters and 350 Pa for F8-F9 filters [EN 13053, Section 6.8]. Eq.(8) is actually just a simplified application of Eq.(5). Alternatively there are advanced calculation methods that account for the slightly nonlinear rise in pressure drop over time [28].

$$SFP_{e,AHU} \approx 0.5\left(SFP_{\text{new filter}} + SFP_{\text{old filter}}\right) \qquad (8)$$

If the AHU has bypass dampers to allow airflow past heat exchangers outside the heating or cooling seasons, then $SFP_{e,AHU}$ should ideally account for the fraction of the year that the bypass is in operation. This can be calculated with Eq.(5).

Just as for $SFP_{v,AHU}$, $SFP_{e,AHU}$ is normally documented (i.e. measured or calculated) by the AHU manufacturer for each duty-point that the customer specifies. If the customer does not specify any part-load duty points, then the AHU manufacturer can assume a 'default' part-load duty-point defined as follows[10][36]:

- *Flow rate*: 65% of design (maximum) flow rate
- *External pressure*: 65% of design external pressure drop

[1] If the fans are fed power from VSDs that are integrated in the AHU, then the power consumed by the VSDs is measured. Hence SFP_v accounts for VSD efficiency.

AHU manufacturers are increasingly developing proprietary software to calculate both $SFP_{v,AHU}$ and $SFP_{e,AHU}$ to customer specifications of external (ductwork) pressure drop and design flow rate. See also page 17.

2.2.4.3 SFP_{BLDG} for whole buildings

Eq.(1) can be applied where q_v is the building's total ventilation flow rate, and ΣP is the sum power of all the fans in the building's ventilation system that contribute to air change. This should include both supply and exhaust fans in balanced ventilation, exhaust fans in exhaust systems, kitchen extract hoods, and ventilating fans without duct connection (i.e. wall or roof fans). But local recirculating fans, such as ceiling fans and Fan Coil Units without duct connection, are not included in the normal definition of SFP_{BLDG} for whole buildings. This is because they do not contribute to air change. However, the internal heat gain from such fans should of course be accounted for when designing the building.

Table 1 shows an example of calculation of SFP for a whole building. If the ventilation system is not balanced, q_v is the largest of summated supply and summated exhaust flow rates in the building. EN 13779 Annex D gives further explanation, together with a more detailed example.

Table 1: Example of calculation of SFP for a whole building

System	Fan#	Supply flow rate	Exhaust flow rate	Fan power	SFP_{AHU}
		m³/s	m³/s	kW	kW/(m³/s)
AHU 1	S1	0.50	-	0.98	3.66
	E1	-	0.48	0.85	
AHU 2	S2	2.50	-	3.36	2.92
	E2	-	2.38	3.93	
Toilet 1	EF1	-	0.1	0.06	0.60
Toilet 2	EF2	-	0.09	0.06	0.67
Kitchen	EF3	-	0.11	0.06	0.55
SUM building		**3.00**	**3.16**	**9.30**	**-**
Total fan power, ΣP =				**9.30**	kW
Gross airflow, Qv = MAX[ΣQs,ΣQe] =				**3.16**	m³/s
SFP_{BLDG} = ΣP / Qv =				**2.94**	kW/(m³/s)

In general, the value of SFP_{BLDG} calculated as shown above in Table 1 applies to a specific operating mode and season, and is thus just a snapshot in time. Different values of SFP_{BLDG} can be calculated for different modes, such as typical winter daytime, night-time operation, or design peak airflow (The latter is normally the one limited in building regulations). In the case of energy calculations, the time-averaged value $SFP_{BLDG,e}$, can be estimated using Eq.(5).

Many fans are intermittently operated, e.g. kitchen hoods and humidity-controlled bathroom fans. Whether or not such fans should be included in SFP_{BLDG} depends on the assumed operating mode. In the case of systems with a lot of latency (e.g. kitchen hoods in apartment buildings), it is extremely unlikely that all the fans will be operating at the same time. As a general rule, the design peak airflow rate for a building can be defined as the 95% highest value in the probability distribution of the building's total airflow rate. The number of intermittently used fans that are operating at the time of the building's 95% peak flow rate can be estimated using the following Microsoft Excel® function[1]:

$$n = \text{BINOMDIST}(N, p, \alpha) \qquad (9)$$

where

n = Number of intermittently used fans that are operating

N = is the number of intermittently used fans in the building

p = is the probability that any given intermittently used fan operates at any given time, is approx. 0.15 for kitchen hoods & bathroom fans.

α = confidence limit, = 0.95 for design flow rate

For example, in an apartment building with N=10 kitchen hoods, each with p=15% probability of being used during peak use periods, n=3 kitchen fans are being used at α=95% peak design periods. The same equation may be applied in the case of calculating the design flow rate in buildings with bimodal air terminals such as humidity-controlled vents in bathrooms or VAV terminals controlled by presence detection in cellular offices.

It is mentioned above that SFP is not constant for a given ventilation product, but is affected by the unit's operating point when installed in a specific building, i.e. the combination of flow

[1] This equation assumes bimodal and independent operation of equally sized fans. The probability distribution is a binomial Cumulative Distribution Function (CDF).

rate and pressure drop. It is therefore important that manufacturers' performance data (SFP or ΣP & q_v) should be specially tailored to each specific building application, taking into account the correct components in the unit, and with the correct external pressure (i.e. pressure drop though the ductwork, ATDs, air intake/outlets and transfer between rooms). Building energy calculations (e.g. ISO 13790) should be conducted with the same value(s) of SFP_e, preferably using realistic values of part-load SFP_e for each time interval. This can for example be estimated with Eq.(6).

2.2.4.4 SFP_{FCU} for fan coil systems

In the UK, where fan coil units (FCUs) are popular, the building regulations set separate limits on SFP_{BLDG} and SFP for FCU systems. This is because FCU systems are omitted from SFP_{BLDG} when not duct-connected, as explained above. FCUs with 4-pole AC motors can achieve SFP_{FCU} of 0.5~0.8 W/(ℓ/s) depending on capacity, while FCUs with EC motors can achieve 0.15~0.4 W/(ℓ/s). If a building contains various types or sizes of FCUs, then the overall average SFP for all the FCUs in a building may be calculated using Eq.(1), which is repeated below in expanded form to show exactly how SFP_{FCU} can be calculated based on the mains power (P) and flow rate (q) or SFP_i of the individual units:

$$SFP_{FCU} = \frac{\Sigma P}{\Sigma q} = \frac{N_1 P_1 + N_2 P_2 + \cdots}{N_1 q_1 + N_2 q_2 + \cdots} = \frac{N_1 P_1 + N_2 P_2 + \cdots}{\dfrac{N_1 P_1}{SFP_{i,1}} + \dfrac{N_2 P_2}{SFP_{i,2}} + \cdots} \tag{10}$$

where

N = Number of FCUs of a given type, in the building

P = mains power supply to a given type of FCU [W]

q = flow rate for a given type of FCU [ℓ/s]

SFP_i = SFP of an individual FCU, at power P and flow rate q [W/(ℓ/s)], [kW/(m³/s)]

The same approach could be used for other recirculating systems such as ceiling fans.

2.2.4.5 SFP values corrected for duct leakage

Some organizations have suggested a nonstandard definition of SFP that accounts for extra fan energy due to duct leakage. This is basically the same as Eq.(1) except that the flow rate, q_v, is redefined as the *net* air renewal rate at room-level (i.e. the summated terminal flow rates in all rooms). This net airflow rate is less than the gross airflow passing through the fans, due to duct leakage. This approach is not recommended for the following reasons:

- AHU manufacturers avoid this definition of SFP because they obviously cannot predict duct leakage, which varies between buildings.
- It is more labour-intensive, and less accurate, to measure and verify than $SFP_{v,AHU}$
- Duct leakage is an independent property of ventilation systems. It is thus measured separately from SFP, and can be regulated separately in building regulations
- In the case of building energy calculations, it is more correct to account for duct leakage as an independent input parameter, such that the duct leakage flow is calculated and included in q_v.

2.3 Measurement of SFP

A very useful guide to SFP commissioning is given in [36]. This describes which parameters to measure, how to correct for air density and measurement uncertainty. A more general guide to commissioning measurements is given in EN 12599 [7].

2.3.1 Acceptance tests

For acceptance testing of large AHUs, it is advisable to perform the measurements prior to delivery, at either the manufacturer's premises or a third-party laboratory, and under supervision of the customer. This has two main advantages over post-installation field measurements: (a) laboratory methods are generally of higher accuracy, and (b) the influence of the connected duct system can be accurately simulated to obtain the presupposed pressure drops specified in the design. In the case of post-installation tests, the external pressure drops often deviates from that presupposed in the design. Hence it can be difficult to confirm/refute the fan performance claimed by the AHU manufacturer.

2.3.2 Accurate measurement of fan power

Large fan motors are generally fed 3-phase power via variable-speed drives (VSD) that draw non-sinusoidal line currents. Cheap wattmeters are inappropriate for measuring power in such cases. One should ideally use a 'power quality analyzer' or equivalent advanced wattmeter that can measure true-RMS 3-phase power for unbalanced loads with up to 50 harmonics (Figure 17). These cost approximately € 3000 / $ 4000.

Figure 6: Example of an appropriate power meter for 3P3W measurements [Fig. © HIOKI]

Use of these instruments involves connecting at least 6 leads (3 current clamps and 3 voltage leads) to the motor's power supply, ideally between the switchboard and the VFD, so that VFD losses are included. The measured power consumption will be incorrect if these leads are connected in the wrong order to the three phases. To prevent this error, it is sensible to use the power meter's graphical phasor-diagram to check the phase angles (Figure 7). One should always take safety precautions, such as wearing insulated safety gloves.

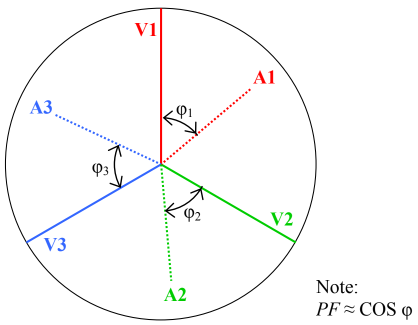


Figure 7: Example of phasor diagram for a 3-phase induction motor, with PF of approx. 0.7

Further, the following equation can be used to double-check the true power consumption:

$$P = \sqrt{3} \cdot \bar{V}_L \cdot \bar{I}_L \cdot PF \qquad (11)$$

where

- P = Total active power use for all 3 phases [kW]
- $\bar{V}_L$ = true RMS line phase voltage, e.g. 230 V in Europe/Asia, 110 V in America/Japan [V]
- $\bar{I}_L$ = true RMS line phase current [A]
- PF = Power factor, can be 0.45~0.8, but is generally in the region 0.65~0.78 for induction motors.

Three-phase induction motors are generally delta-connected, such that the measured delta voltage across two motor terminals is the same as the line voltage V_L. If a VFD is not used, large fans motors are generally connected to a star-delta motor starter in the switchboard. This motor starter contains a relay that starts the fan in a star-connection (wye-connection), and then automatically switches over to a delta-connection for normal motor operation. Older motors may have two separate switches in the switchboard (wye & delta), but power consumption need only be measured at the delta-connected switch.

2.3.3 Measurement of flow rate

Most large modern AHUs have a means to easily measure flow rate. The control panel may display each fan's flow rate, calculated from the measured static pressure rise over the fans' calibrated venturi inlet. However, such measurements can be misleading since they do not correct for factors such as air recirculation or leakage in the AHU. Figure 8 illustrates this problem — neither the supply fan rate, nor exhaust fan flow rate, are equal to the true air renewal rate (shown in green).

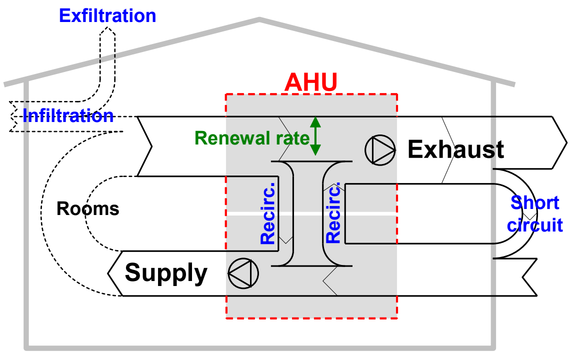


Figure 8: Example of air flow paths in a building with an AHU with recirculation/leakage and short-circuiting outdoors. Unbalanced airflows, cause net infiltration [Fig. © SINTEF]

The only means of correctly measuring all these air flows is by tracer gas measurement. This is very costly, requires a high level of expertise, and is prone to inaccuracy. We have therefore listed some pragmatic recommendations below for simple SFP commissioning measurements based on measured fan flow rates:

- Close recirculation dampers.
- If the AHU has a rotary heat exchanger with purge sector, subtract the purge flow from the exhaust fan flow rate. The purge flow rate can be estimated from the pressure difference over the purge sector, and the sector angle, using tabulated values from the manufacturer of the AHU or heat exchanger.
- Do not correct flow rates for short circuiting outside the building, or internal leakage.
- Do not correct flow rates for duct leakage

Figure 9 illustrates another common issue in AHUs with rotary heat exchangers, which often have a purge sector (Figure 10) to prevent carry-over of contaminants from the return air into the supply air stream. This purge flow rate can constitute >10% of the exhaust fan air flow. If the airflows passing though the supply and exhaust fans are exactly equal, then the system will be unbalanced, causing net exfiltration. If there is no other recirculation or short-circuiting in the system, then the true air renewal rate (shown in green) is equal to the supply air flow rate.

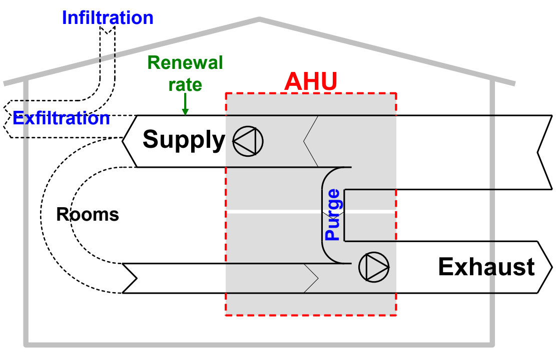


Figure 9: Example of air flow paths in a building with an AHU with a rotary heat exchanger with purge sector. Airflows are incorrectly balanced, causing net exfiltration [Fig. © SINTEF]

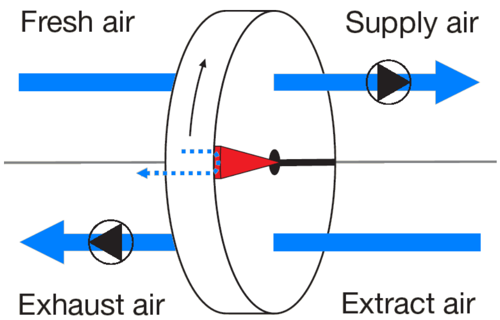


Figure 10: Illustration of a rotary heat exchanger with a purge sector (coloured red)

2.4 Examples of SFP in building codes

In most cases, building regulations set limits on $SFP_{BLDG,v}$ for whole buildings at design (maximum) flow rate. Although building regulations do not normally explicitly state so, this is based on the value of SFP_v for the individual AHUs or fans in the building (i.e. measured with clean air filters and dry coils), to ease on-site verification.

- *UK, 'Part L' of the building regulations (2010):* Exhaust systems: $SFP_{BLDG} \leq 0.6$ kW/(m³/s), Balanced ventilation with heat recovery: $SFP_{BLDG} \leq 1.0$ kW/(m³/s). SFP_{BLDG} at 25% of design flow rate should not exceed SFP at 100%. Motors should have efficiency class IE2 (EFF1). FCU systems $SFP_{FCU} \leq 0.6$ kW/(m³/s).
  $SFP_i \leq 0.2$ kW/(m³/s) for non-ducted local ventilation (wall fans, even if intermittent operation).
- *Finland (2007)*: Max. 2.5 kW/(m³/s) for ordinary systems.
- *Sweden (2006)*: Max. values for systems over 0.2 m³/s: Balanced with heat recovery

2.0 kW/(m³/s); Balanced without heat recovery 1.5 kW/(m³/s), Exhaust with heat recovery 1.0 kW/(m³/s), Exhaust without recovery 0.6 kW/(m³/s). Mandatory inspection scheme ('OVK') includes SFP measurement.

- *Norway (2007)*: Max. 2.5 kW/(m³/s) for dwellings. Other buildings max. 2 kW/(m³/s) during working hours; 1 kW/(m³/s) at other times. Heat recovery is assumed.
- *USA, ASHRAE 90.1 (1999)*: For systems below 9.4 m³/s: CAV: 1.9 kW/(m³/s), VAV: 2.7 kW/(m³/s). For systems above 9.4 m³/s: CAV: 1.7 kW/(m³/s), VAV: 2.4 kW/(m³/s). Fan efficiency requirements will be added in 2010.
- *USA, California Title 24*: For fans over 17 kW: VAV: 2.7 kW/(m³/s), CAV: 1.7 kW/(m³/s).

2.5 Recommended 'good-practice' SFP

The appropriate value of SFP for a specific application depends on the size of the ventilation system, whether it is balanced or has heat recovery, intermittency of operation, and of course costs. These factors are partly accounted for in the following simple equation:

$$SFP_e \le \Sigma A \cdot B \cdot C \qquad [\mathrm{kW/(m^3/s)}] \qquad (12)$$

where

Σ*A*: Summation of the following conditional terms:
=1 for the presence of a supply fan
+1 for the presence of an exhaust fan
+1 for the presence of heat recovery
+1 for small systems, below 0.1 m³/s
+1 for systems below 0.2 m³/s

B: Constant depending on intermittency; either:
=1 for daytime operation of systems, < 4000 hrs/yr
=0.75 for systems with round-the-clock operation
=0.5 for night time operation of VAV systems

C = 2/3. This constant may be reduced in future as building codes are tightened.

Examples:

- A small dwelling with continuous balanced ventilation and heat recovery should have $SFP_e \le 5 \times 0.75 \times 2/3 = 2.5$ kW/(m³/s).
- A large building with balanced ventilation and heat recovery should have $SFP_e \le 3 \times 1 \times 2/3 = 2$ kW/(m³/s) during working hours, and $SFP_e \le 3 \times 0.5 \times 2/3 = 1$ kW/(m³/s) at night.

Eq.(12) above is generally in line with the building regulations in Nordic countries in 2010, and can easily be achieved with modern ventilation products, together with astute building/HVAC design. However, some manufacturers of ventilation products need to further develop their products and competence to achieve these levels. These levels ensure economic good practice; lower values may become more profitable in future as and energy costs increase and technology improves.

Investment in measures that reduce installation SFP should also be profitable. This depends on the criteria of profitability used, on operation of the ventilation system, price for electricity, etc. To prevent sub-optimization (overzealous minimization of SFP), the cost calculation should consider the building's *total* energy consumption. This will account for the influence of SFP on space heating/cooling demand. This avoids the pitfall of selecting less efficient heat recovery units as a means of reducing system pressure drop, such that the reduction in fan power is overshadowed by an increase in space heating/cooling.

3 Fan system efficiency – Definitions and recommendations

3.1 Definition and typical values

Fan system efficiency (also known as *Driven Fan efficiency*) is a measure of the fraction of the supplied electrical power that ends up as a useful air pressure rise across the fans.

The fan system's overall efficiency (η_{tot}) equals:

$$\eta_{tot} \equiv \frac{P_{out}}{P_{in}} = \eta_{vsd} \cdot \eta_{motor} \cdot \eta_{transfer} \cdot \eta_{fan} \qquad (13)$$

where

- P_{in} = Power input, mains electricity [W]
- P_{out} = Useful output from the fan [W] = $\Delta p_{tot} \cdot q_v$ where Δp_{tot} is pressure rise [Pa] and q_v is flow rate [m³/s]
- η_{vsd} = variable speed drive (VSD) efficiency, if present
- η_{motor} = motor efficiency
- $\eta_{transfer}$ = power transfer efficiency (belt, bearings)
- η_{fan} = fan aerodynamic efficiency

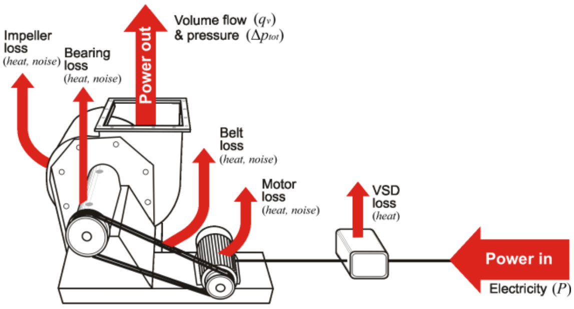


Figure 11: Losses from a traditional belt-driven centrifugal fan [Fig. © FMA], [20]

One must distinguish between total pressure and static pressure rise when declaring fan system efficiency. It is most correct to define the fan's aerodynamic efficiency based on total pressure (i.e. static+dynamic pressures). Nevertheless, the value SFP is the same irrespective of the chosen pressure.

3.1.1.1 Typical values

Measurements of AHUs in random Norwegian commercial buildings in 1998 showed an average system efficiency of η_{tot}=44%. Thus, on average 56% of the electrical power is lost as heat & noise/vibration in the fan system. Smaller systems tend to have a much lower efficiency, typically 15% for residential ventilation[34] . A more recent study concluded that the future potential for improvement of system efficiency is 17.5% in Europe [21].

3.1.1.2 Rating Driven Fan efficiency

New ISO and AMCA standards [4],[8] have been established to define an classification system for fan systems called '*Fan-&-Motor Efficiency Grade*' (FMEG) for fans larger than 125 mm diameter. See Figure 12 (B-wheel fans) and Figure 13 (other fans). The curves account for the physical laws governing efficiency of scale, which makes small fans less efficient than large fans of the same design. Well designed large fan systems can achieve at least FMEG60 (i.e. 60% efficiency for a 10 kW system). FMEG limits can be set in building regulations as minimum energy-performance requirements.

The FMEG curves are defined by the following equation:

$$\eta_{tot,peak} = a \cdot \ln(P) + b + FMEG \qquad (14)$$

where

- $\eta_{tot,peak}$ = fan system efficiency (total pressure) at the operating point giving peak efficiency [%]
- a,b = coefficients, see Table below
- P = fan system input electric power [kW]
- $FMEG$ = efficiency grade [0-100] in steps of 5 %

Table: Coefficients a & b depend on fan type & size

	P [kW]	*a*	*b*
B-wheel	0.125 ≤ P ≤ 10	4.56	-10.50
	10 ≤ P ≤ 500	1.10	-2.60
other fans	0.125 ≤ P ≤ 10	2.74	-6.33
	10 ≤ P ≤ 500	0.78	-1.88

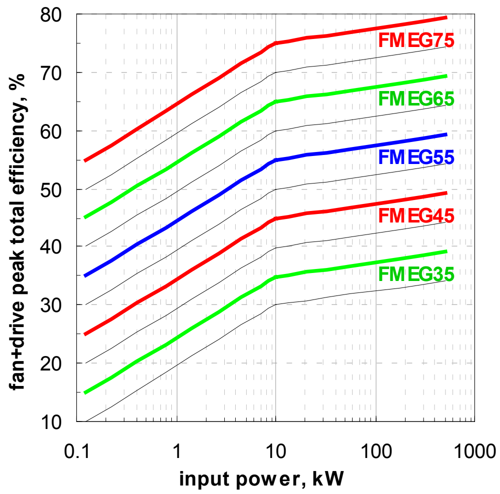


Figure 12: Driven Fan efficiency (η_{tot}) classification system for B-wheel fans, with or without fan housing [ISO 12759]

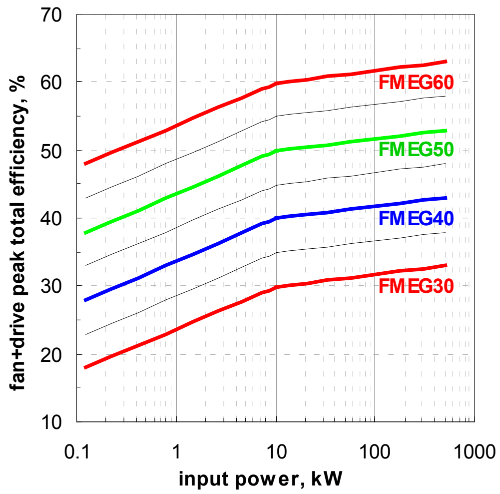


Figure 13: Driven Fan efficiency (η_{tot}) classification system for other fans[1] (axial, F-wheel, T-wheel, or mixed flow fans). [ISO 12759]

3.2 Components of fan system efficiency

3.2.1 Fans – Aerodynamic efficiency

Fan aerodynamic efficiency has two alternative definitions:

$$\eta_{fan,tot} = \frac{\Delta p_{tot} \cdot q_v}{P} \quad , \quad \eta_{fan,stat} = \frac{\Delta p_{stat} \cdot q_v}{P} \qquad (15)$$

where:

$\eta_{fan,tot}$, $\eta_{fan,stat}$ = bare-shaft fan total or static efficiency

Δp_{tot} , Δp_{stat} = total or static pressure rise over fan

q_v = volume flow rate [m³/h]

P = bare-shaft power [W]

As mentioned earlier, it is scientifically most correct to define efficiency based on total pressure ($\Delta p_{\mathrm{fan,tot}}$), not static pressure. It is meaningless to apply the static pressure efficiency for operating points near free-flow conditions (no flow resistance), because the fan generates dynamic pressure but little static pressure (e.g. 100% flow rate in Figure 16 & Figure 18).

For fans with integrated motor drives (such as many EC motor fans), it is not possible to measure η_{fan} directly; only the combined total fan system efficiency can be measured.

The most important factor affecting fan system efficiency is fan shape/size. The potential for future improvements in aerodynamic efficiency of fans has been estimated to be in the range 10~15 %, but depending on fan type and sector, the realistic potential is 3.5~8.3 % [21].

3.2.1.1 Fan types

There are three main types of fan: *centrifugal*, *axial*, and *mixed-flow*. A fourth type, *cross-flow* (or *tangential*) fans will not be mentioned further, due to their very low efficiency.

Centrifugal fans have impeller blades that are arranged like a cylindrical cage. (Figure 11, Figure 14). The blades can be curved forwards (F-wheel), backwards (B-wheel), or straight radial (T-wheel); see Figure 15.

[1] With the exception of cross-flow (tangential) fans, which have their own FMEG curve system

Figure 14: Plenum (plug) mounted centrifugal fans with direct drive [Fig. © Ziehl-Abegg; Swegon]

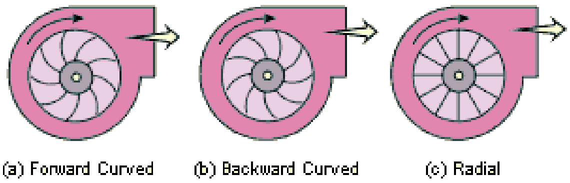


Figure 15: F-wheel, B-wheel and T-wheel centrifugal fans in scroll housing [Fig. © US EPA]

B-wheels are the most efficient centrifugal fans. Although they are usually the best choice in terms of energy efficiency, they may have a limited operating range (pressure changes have little influence on flow rate) and are somewhat noisier than F-wheels. In systems with varying flow rate, these fans must be controlled by a variable speed drive (VSD or VFD). They may be installed in *scroll housing* (i.e. a 'blower'; see Figure 11 & Figure 15) or *plenum housing* (i.e. a plug arrangement with nozzle/venturi/ cone inlet, and free outlet; see Figure 14). Scroll housing gives approx 10% higher efficiency but is less compact than plenum housing. Double-inlet scrolls achieve the best efficiency of all (i.e. inlet from each side of scroll). On the other hand, connecting sharp bends directly to the outlet gives higher system-effect loss for scrolls than for plenums. Plenum/plug fans have become very popular due to their lower cost, compactness, and flexibility due to the ability to offer different take-offs within a short distance from the fan. Plenum fans suffer from low frequency noise (≤ 125 Hz), due to turbulence in the plenum, at the fan inlet and discharge. This noise is difficult to attenuate.

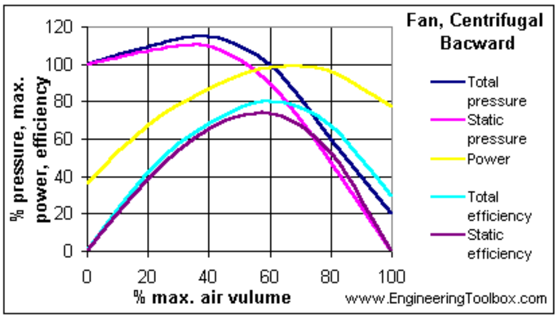


Figure 16: Typical fan curve for B-wheel fan, showing total & static pressure rise, and total & static efficiency. [Fig. © www.engineeringtoolbox.com]

F-wheels were traditionally the most common type in ventilation systems in large buildings, but they are inefficient and are thus now pertinent only in special cases. One of the reasons for the commonness of F-wheels, apart from their quietness, was the practice of selecting very compact AHUs (B-wheels need more space for the same flow rate and moderate pressures). T-wheels are also inefficient, but are used in process industry for air loaded with particulate matter, due to their self-cleaning ability. Both F and T-wheels need scroll housing.

Axial / propeller fans: See Figure 17. These are the best choice for applications with low pressure drop and high flow rate. Axial fans are usually connected directly to the motor shaft, thus avoiding transmission losses [see page 20]. Axial fans are now rarely used in conventional ventilation systems, but are used in systems with very low pressure drop, such as hybrid ventilation. Axial fans are ideal for non-ducted applications, by mounting in a wall ring. They are often mounted with a wire guard grille, which can reduce peak efficiency by 10%.

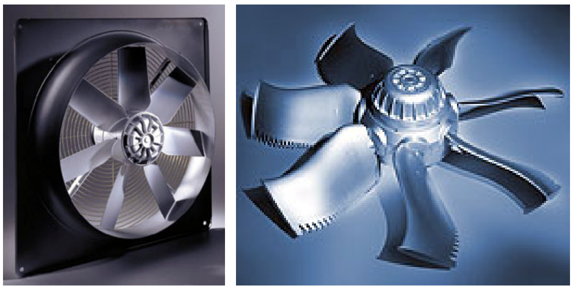


Figure 17: (a) Axial fan in a wall panel, (b) efficient axial fan with variable-pitch swept blades [Figs. © Ziehl-Abegg]

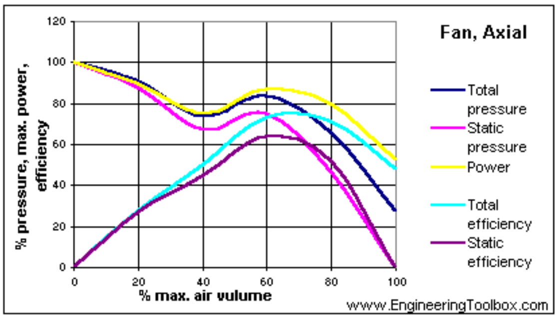


Figure 18: Typical fan curve for axial fan, showing total & static pressure rise, and total & static efficiency.
[Fig. © www.engineeringtoolbox.com]

Mixed flow / axial-radial / diagonal fans are a new kind that delivers more air than centrifugal fans, and higher static pressure than axial fans. They are directly-driven fans, just as axial fans and centrifugal plug fans. They have less outlet losses than axial fans (see §4). This provides more flexibility than with centrifugal or axial fans in relation to fan position, fan outlet (they are ideal for 'plug' installation), and design of other components. On the other hand, mixed flow fans are less efficient than B-wheel centrifugal fans.

Figure 19: Mixed flow fan with variable pitch impellers [Fig. © Continental Fan Mfg. Inc]

Fans for hybrid ventilation: Both axial and centrifugal fans may be suitable for hybrid ventilation. Specialized roof-mounted vertical-axis centrifugal fans (Figure 20) appear to have an advantage over axial fans in that the fan blades do not obstruct flow in the stack when the fan motor is switched off during periods of adequate natural driving forces (wind and/or stack-effect). The alternative, an axial fan located in the stack or throat of the wind cowl, can cause a significant flow reduction when inoperative, and can emit more noise when operating. On the other hand, the blockage effect of a switched-off fan is irrelevant in the case of ventilation systems with a high pressure drop (>10 Pa) that usually exceeds the available natural driving forces. In this case, fan-assistance is needed all the time, so axial fans will be a better option, as they achieve a higher efficiency (Figure 21).

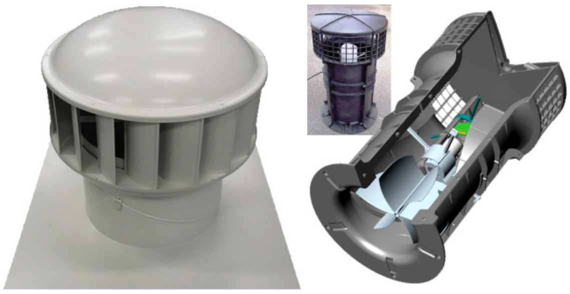


Figure 20: Two examples of centrifugal fans for intermittently fan-assisted stack ventilation. The left hand cowl has SFP≈1, and a discharge coefficient of 0.51<C_D<0.81 (1.5<ξ< 3.7) when free-wheeling. The right hand fan has negligible pressure drop when switched off, and SFP≈0.19 at 10 Pa & 42 ℓ/s with 8 W DC-motor fan with system efficiency 5.2%. Peak system efficiency is approx 6.5% [26].
[Figs. © CSR Edmonds; Aereco]

Figure 21: Example of axial fan for continuous fan-assisted residential stack ventilation. SFP is 0.04 kW/(m³/s) at nominal flow 10 Pa & 56 ℓ/s with 2 W [1] EC-motor fan with system efficiency 28% [26]. [Fig. © TNO]

3.2.1.2 Other efficiency factors

Other factors that affect efficiency include blade material, profile & shape, hub diameter and the number of blades. In general, aerofoil profiles are most efficient, but curved profiles made of uniform thickness sheet metal are only slightly less efficient. Blade shape is also important, for example the most efficient axial impellers have a curved 'machete' or 'teardrop' form (e.g. Figure 17b) while flat 'paddle' or 'clover-leaf' types are inefficient. Clearance between the impellers and fan housing should be small to minimize local recirculation ('blow-by').

[1] Auxiliary energy for DCV controls is 5 W

Having a large number of blades reduces the intensity of stalling (at throttled air flow).

3.2.1.3 Fan size and efficiency

Figure 22 shows how peak fan efficiency depends on size. Larger fans are more efficient than smaller ones because losses do not scale up in proportion with size. The impact of mechanical & volumetric losses, and viscous forces (Reynolds number) are larger for small fans.

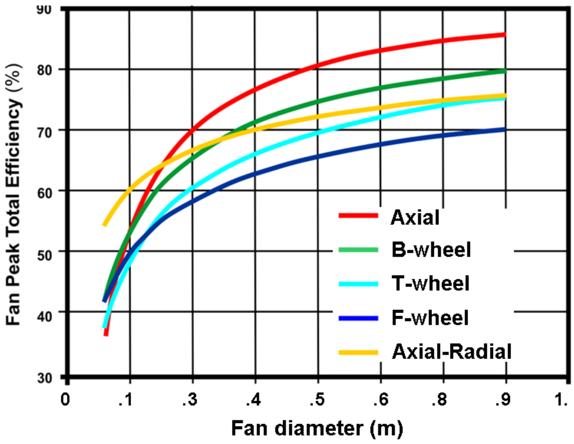


Figure 22: Fan peak efficiency curves. [Fig. © ACME Engineering & Manufacturing Corp]

3.2.1.4 Sizing for optimum performance & efficiency

The trick is to select the correct size of fan such that it normally operates at, or near, peak efficiency. This principle is illustrated in Figure 23 which shows pressure and efficiency curves for two alternative fan sizes. The larger fan operates at peak efficiency, and achieves the required flow rate at a slightly lower total pressure rise, because the lower outlet velocity reduces system losses at the fan outlet.

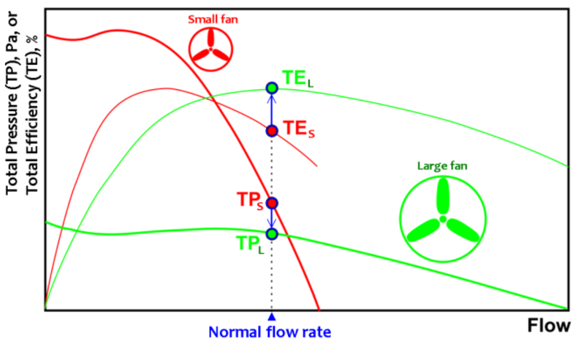
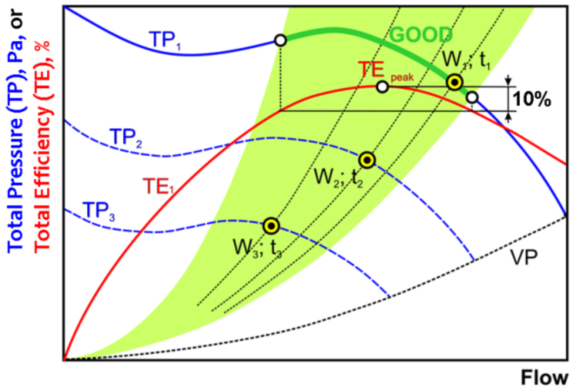


Figure 23: Fan curve illustrating optimum region for supply fan selection. [Fig. © ACME Engineering & Manufacturing Corp]

For systems with variable flow rate, one should consider the entire range of flow rates. The fan size may be optimized by taking into account the amount of time that it operates at different flow rates. This is illustrated in Figure 24 for a system with 3 different operation modes. The operating points should ideally not be more than 10% less than the peak efficiency at each fan speed. This is 'good' operating region is coloured green Figure 24. Figure 24 also shows fan pressure curves (blue) for 3 different fan speeds, and the efficiency curve (red) for the largest fan speed.



$$E_{year} = W_1\,t_1 + W_2\,t_2 + W_3\,t_3$$

Figure 24: Fan curves. [Fig. © ACME Engineering & Manufacturing Corp]

Unfortunately, fans are commonly oversized, such that their peak efficiency occurs at, or near, the 'worst-case' (design) flow rate. Such systems are troubled by low frequency noise (rumbling) when the fan operates at normal or reduced flow rates. This noise is difficult and costly to attenuate. It is better to select a fan size such that peak efficiency is achieved at the most common flow rate. During the short periods of maximum flow demand, the increased fan noise will have a higher frequency, which is much easier to attenuate, using cheaper silencers. Another key reason not to oversize fans is that they can stall or surge at minimum flow demand. Smaller fans have a narrower surge region. These issues are illustrated in Figure 25 which shows two alternative fan sizes, of which the smaller one is preferable with regards to acoustics & energy.

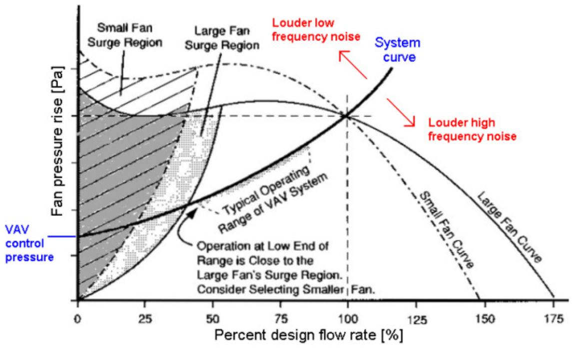


Figure 25: Fan curves two alternative fan sizes, showing the location of surge regions that should be avoided during part-load operation. [33]

3.2.1.5 Software for product selection & sizing

Many manufacturers offer software that can estimate fan performance for any given combination of flow rate and pressure differential, and select optimum components. Such software generally interpolates data from a limited number of measured operating points, and should ideally be certified (e.g. EUROVENT).

3.2.1.6 Fan performance rating

Large fans are generally rated in a laboratory according to ISO 5801 or ANSI/ASHRAE 51-07 (ANSI/AMCA 210-07). This can be done with or without a ducted inlet or outlet, giving 4 possible test configurations. Ducting modifies the flow into and out of the fan, and its design is extremely important. Most fans perform best with ducted inlet & outlet, except for axial fans. It is strongly recommended that designers ensure that data provided by a fan supplier is for a test configuration that matches the way in which the fan will be used in practice. If this information is not available then corrections should be made with AMCA's system-effect (SE) factors [16].

Standards ISO 12759 and AMCA 205 [4],[8] define a rating system called '*Fan Efficiency Grade*' (FEG) for the aerodynamic efficiency of bare-shaft fans of at least 125 mm. See Figure 26. This has enabled international harmonization of fan documentation. The FEG curves (Figure 26) were developed based on measurements of many types and sizes of fans, and accounts for the fact that efficiency is influenced by scale laws (also illustrated in Figure 22). All sizes of fan of a specific design generally achieve the same FEG rating. The FEG number for a specific fan is the peak aerodynamic efficiency[1] of a 1 m diameter fan of exactly the same design. For example, a fan of diameter 0.2 m and efficiency 50% is given a rating of FEG67 because, according to Figure 26, a large fan (with diameter≥1.0 m) of the same design can achieve 67% efficiency. The FEG curves are generic, i.e. the same curves apply to all types of fan, i.e. centrifugal, axial, and mixed flow. FEG numbers are a geometric series in steps of 6%, starting at 95%, i.e. 90%, 85%, 80%, 75%, 71%, 67%, 63%, etc.

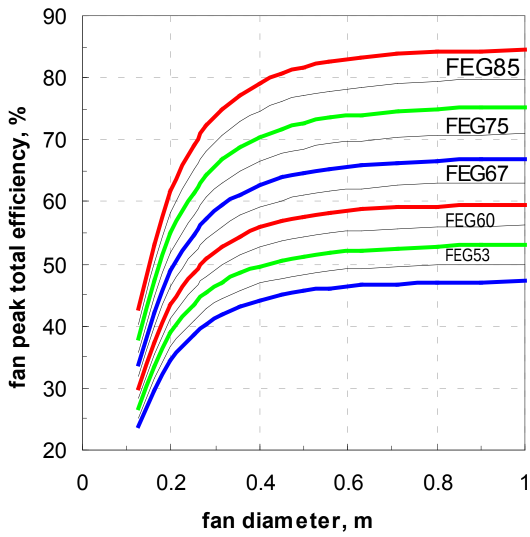


Figure 26: FEG classification system for bare-shaft fan total efficiency, $\eta_{fan,tot}$, at the fan's optimal operating point [ISO 12759]

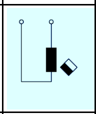
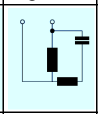
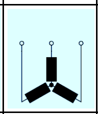
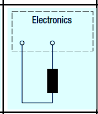
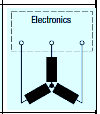
[1] Peak efficiency of a fan is the efficiency at the fan's optimal operating point, i.e. the combination of flow rate and pressure rise that gives the highest efficiency.

3.2.2 Motors and Motor efficiency

3.2.2.1 Motor types

Table 2 gives an overview of common types of fan motor.

Table 2: Comparison of features of main motor types used for fans. [Figs. © EBM]

	AC induction motors			EC motors [1]	
	Shaded pole	Permanent split capacitor	3-phase induction	Single core	3-core
Circuit design					
Power supply	1-phase AC	1-phase AC	3-phase AC	1-ph. AC or DC	any
Capacity	< 50 W	< 0.5 kW	> 1 kW	< 5 kW	< 5 kW
Integrated VSD	-	-	-	✓	✓
Rotor type	Squirrel cage	Squirrel cage	Squirrel cage	Magnetic rotor	Magnetic rotor
Efficiency	Poor	Medium	Good	Better	Best
Noise	Poor	Medium	Good	Medium+	Good+

Small residential ventilation fans have traditionally used shaded-pole AC motors. These are the least efficient motors but are cheap and reliable. Progressive manufacturers have switched to more efficient and expensive DC motors with electronic commutation (EC[1]). EC motors have integral speed control (VSD) with higher efficiency than VFDs for AC motors. Some small units even have a fan-specific sensorless control algorithm that can maintain constant flow rate or constant pressure rise. These motors have much lower losses than AC induction motors due in part to use of permanent magnets instead of electrical currents in the rotor. There is also no slip in rotor speed, unlike AC induction motors.

Larger motors are traditionally AC induction (asynchronous) motors, of which 3-phase motors with 4-poles are the most efficient. However large EC motors are gaining popularity for both axial and centrifugal plug fans. They cost more than the combined cost of AC motor with VFD, yet have lower LCC due to their higher efficiency, especially under part-load operation. They are also quieter and smaller, enabling shorter AHUs. All the largest motors (either AC or EC) run on 3-phase mains.

[1] Brushless DC Motors with permanent magnet rotor (full acronym BLDC), commonly known as 'EC-motors'

3.2.2.2 Factors affecting motor efficiency

The efficiency of electric motors depends on many factors. Figure 27 shows how peak motor efficiency depends on size. Larger motors are more efficient than smaller ones because losses do not scale up in proportion with power. For example, magnetic leakage at the ends of stators is scaled by a length-to-volume ratio per unit power, and heat loss is scaled by a surface-to-volume ratio per unit power. For the same reason, small motors have lower part-load efficiency (Table 3).

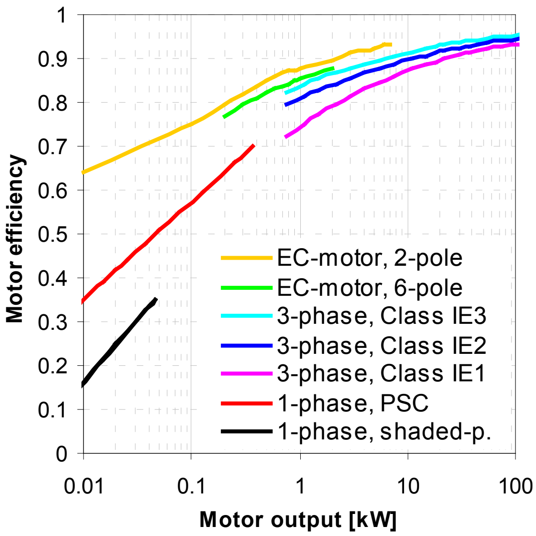


Figure 27: Peak motor efficiency depending on motor size, for different motor types. The three IE classes for 3-phase motors are for 4-pole induction motors at 50 Hz (60 Hz motors below 25 kW have approx 87% of the losses)

3.2.2.3 Sizing and part load performance

Table 3 shows the typical part load performances of different system components of different sizes. Figure 29 illustrates the overall efficiency of a 3-phase AC induction motor together with a VFD. It shows a clear drop below approx 50% of maximum load.

A motor is oversized if the nameplate power rating is significantly greater than needed. Oversized motors, especially small ones, are less efficient than motors that are suited to the load (Figure 29). The influence of size on efficiency depends on motor type, manufacturer, and must be checked in each specific case. Field measurements have indicated that many fan motors are vastly oversized, and thus very inefficient. Motors should be chosen to operate at peak efficiency at the particular duty point, not be oversized such that efficiency is reduced. For AC motors, the location of the peak efficiency depends on the motor size as shown in Figure 28. One should replace a motor with a smaller one if it operates at less than 40% of nameplate rating at design flow rate. It is therefore important to calculate the ventilation system's pressure drop before selecting fans and motors that are tailored to the load. If one is unsure of the operating point, or want flexibility in relation to future changes, or have a VAV system, then one should use a speed control strategy (e.g. VSD) that enables more optimal operation of the motor under all conditions, or if that is not possible then use staged fans). In such cases, it is important to use components with low losses (high efficiency) over the part-load range. Still, the motor should not be larger than necessary. If one is unsure whether the motor can meet future needs, one should rather size the electrical supply so that the motor can be changed later.

Table 3: Relationship between system size (kW) and part-load factor on efficiency of motors, belts and VFDs. (Typical~Best product values)

	kW	Part-load factor ($kW_{out}/kW_{out,max}$)			
		100 %	75 %	50 %	25 %
3-phase AC motor[29]	0.74	.76~.84	.74~.82	.70~.77	.58~.64
	7.4	.88~.91	.89~.92	.89~.92	.83~.86
	74	.93~.95	.93~.95	.93~.95	.90~.92
Flat belt [29]	0.74	.91~.94	.90~.92	.86~.89	.79~.81
	7.4	.95~.97	.94~.95	.90~.92	.82~.84
	74	.96~.97	.95~.96	.91~.92	.83~.84
3-phase VFD [30]	0.74	.89	.87	.84	.74
	7.4	.96	.95	.94	.90
	74	.97	.97	.96	.93
EC motor	0.74	.91	.90	.89	.85

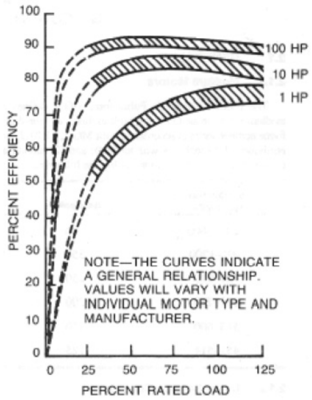


Figure 28: Part-load efficiency curves of 3-phase AC motors [Fig. © NEMA, Standard MG-10]

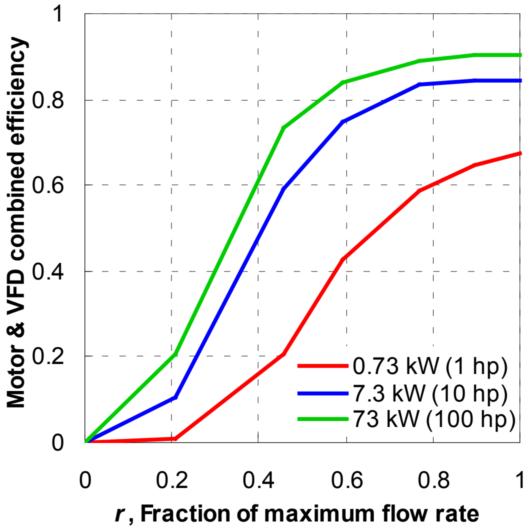


Figure 29: Approximate part-load efficiency curves of the combination of direct-drive AC fan motor and VFD, depending on size (kW). Valid for a typical ventilation system (flow exponent n=1.65 in Eq.4).

3.2.2.4 Motor performance rating

Different laboratory test methods are used, most notably IEC 60034-2-1 and IEEE 112-B. They measure losses in a slightly different way, which can give up to 1~2 % difference.

Motors can be given an efficiency rating based on the above measurements. Standard IEC 60034-30 defines International Efficiency (IE) classes for 3-phase induction motors over 0.75 kW. The 3 classes are: *IE1* (Standard), *IE2* (High efficiency) and *IE3* (Premium). See Figure 27. A fourth class, *IE4* (Super Premium) will be added in future to rate higher efficiency motors such as EC motors. The standard harmonizes earlier rating schemes such as

European CEMEP[1] (Their *EFF1* rating is equivalent to *IE2*) American NEMA[2] (Their '*NEMA Premium*' rating is equivalent to *IE3*) and the mandatory American Energy Policy Act (The *'EPAct'* rating is now equivalent to *IE2*).

Most industrial states have, or soon will, implement(ed) mandatory minimum energy performance standards (MEPS) for large electric motors. The first to do so was USA in 1997. The European Commission (EC) decided to phase in MEPS from 2011 as part of the Energy-using Products Directive (EuP). By 2011 these countries will generally require minimum class *IE2*. By 2017, both USA and EU will require *IE3* (Premium efficiency). The changes may potentially save 135 TWh/yr in EU in 20 years, equivalent to Sweden's annual electricity use. The global savings potential is a staggering 1850 TWh/yr, or 9.7% of global electricity production. Additional savings are achievable as a result of future MEPS for other components, including fans and VFDs.

3.2.3 Power transmission

Large fans in old ventilation systems are generally belt-driven, that is, the motor torque is transferred to the fan by a rotating belt (Figure 11). A disadvantage of belt operation is that it incurs energy loss of over 10 % if poorly designed or maintained, and the losses are substantially higher at low load. Flat belts have lower loss than V-belts. Moreover, particles from belt-wear can pollute the supply air. Because of these particles, there should be a fine air filter downstream in the supply air path. Belt operation makes it possible to change the fan speed by adjusting the exchange ratio between the motor and fan — however this function was made redundant by the advent of electronic speed control (VSD).

Modern fans are generally direct-driven, that is, the fan sits on the motor shaft. Direct-drive fans avoid transmission energy losses. Large direct-drive fans usually have VSD. For AC motors this is costly and incurs a similar loss to belt operation. However, VSD provides a number of advantages:

- The ability to regulate the amount of air to a minimum level, instead of shutting off air flow completely. This may reduce the risk of microbial growth and wear on the unit.
- Demand-controlled ventilation
- The possibility of intensive ventilation in periods with cooling demand
- Optimal efficiency of the fan motor

3.3 Summary of recommendations for fan system efficiency

- Accurately calculate system pressure drop with software (e.g. [40], [41], [42]). Correct for fan system effects [16], to enable proper fan/motor selection.
- Strive for a peak fan aerodynamic efficiency of over 80%. (FEG85) Select B-wheel centrifugal fans in conventional systems or efficient axial fans in low pressure-drop systems (or special centrifugal fans in hybrid systems).
- The fan type/size should be selected such that the efficiency at each operating point is within 10% of the fan peak total efficiency for the given fan speed (Figure 24).
- Select direct-drive fans. If belt operation is unavoidable, then use a flat belt.
- Aim for high fan motor efficiency, i.e. at least IE2 (IE3 after 2017), or ideally an EC motor.
- Do not oversize the fan motor. Size to run primarily in the 65% to 100% load range (the high end for smallest fans; see Figure 28).
- Use a VSD to control fan speed for demand-controlled ventilation. Large VAV systems should also have SPR control (Figure 4).
- Large fan systems are more efficient than small fan systems, due to physical scaling laws. This means that it might be more efficient to use one centralized fan system for a building rather than small decentralized fan systems for each zone in the building (e.g. apartments) . This is true *only* if the large system does not have higher pressure drop, and that there is an efficient control system (i.e. SPR) to regulate the airflow to each zone. Centralized systems have other advantages, such as lower noise and potentially better maintenance.

Correctly selected and sized products should give a total efficiency grade of FMEG60 (i.e. 60% efficiency for a 10 kW system).

[1] Committee of European Manufacturers of Electrical Machines and Power Electronics

[2] National Electrical Manufacturers Association

4 Fan inlet & outlet – Aerodynamic inefficiencies

4.1 General

Fans are tested in laboratories under optimal conditions that can be unachievable in real buildings. In practice, one can expect 10% higher pressure loss (30 Pa for each fan in a large system) due to fan inlet & outlet losses, but poor design can result in at least 75% higher losses (several hundred Pascals in a large system). These losses are collectively called 'system effects', and are caused by swirl in fan inlet, pressure drops at the fan inlet & outlet, and additional pressure loss downstream of the fan before the velocity profile has diffused. The final selection of fan and motor can only be done after the calculated pressure drop has been corrected for system effects.

System effects are calculated by the following formula:

$$\Delta p = C \cdot p_d = C \cdot \left(\frac{1}{2} \rho \cdot v^2 \right) \quad \text{[Pa]} \qquad (16)$$

where

Δp = system effect [Pa]

C = system effect coefficient [dimensionless]

p_d = velocity pressure at fan inlet or outlet [Pa]

ρ = air density [kg/m³]

v = nominal velocity (volume flow rate / area) [m/s]

4.2 Recommendations for fan inlets

- Should have a swirl-free (spin-free) symmetrical fully-developed velocity profile, without flow obstructions.
- For fans housed in a plenum, the minimum distance from the fan inlet to the nearest plenum wall should be greater than 0.75 times the fan inlet diameter.
- The inlet duct's cross-sectional area should be 92%~112% of the fan inlet area.
- There should be a straight inlet duct of at least 3 times the length of its hydraulic diameter (depending on speed). If this is not possible, then square duct bends should have turning vanes to prevent spin. Circular duct bends should have an inner radius at least as large as the duct diameter. Flow straighteners such as filters, coils or heat exchanger reduce the inlet system effect.
- Flow obstacles such as dampers and tees should not occur near the fan inlet.
- Avoid air spin at the inlet. Equi-rotation (i.e. air spin in the same direction as fan rotation) can reduce fan pressure (Figure 30). Contra-rotation is also undesirable as it increases fan noise and energy use.

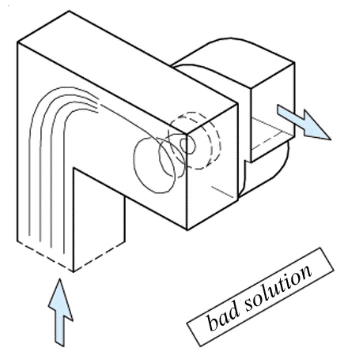


Figure 30: Example of poor design that generates spin in the same direction as fan rotation. Add turning vanes in the bend to reduce spin. [Source: Svenska inneklimatinstitutet & SINTEF]

4.3 Recommendations for fan outlets

- Aim to create a spin-free symmetrical fully-developed velocity profile, without flow obstructions.
- The connecting duct should ideally be straight, enabling the distorted velocity profile to diffuse, giving a gradual exchange of the high velocity at the fan outlet into useful static pressure[1] (See Figure 31). The duct length should be at least 2.5 times the hydraulic diameter (preferably > 6 diameters, depending on speed). See Table 4 for the effect of shorter lengths.
- If sizes of the fan outlet and connecting duct are different, then use a gradual transition. Reduction of the duct cross-section at the fan outlet generally creates less pressure loss than an increase (an evasé or diffuser). Reduction of the outlet duct cross-section should have an angle of max. 30° between the opposite duct walls (Figure 32). A divergence (evasé) should have a maximum angle of 15° between opposite duct walls. Avoid 90° bends; use rather 45° bends.
- If a bend is necessary, it should follow the direction of air flow out of the fan. See for example Figure 32 and Figure 33.

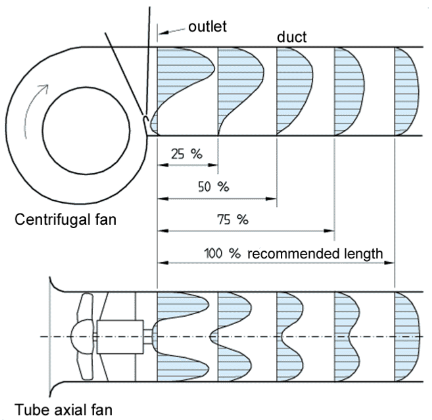


Figure 31: Velocity profile at outlets of centrifugal and axial fans. Outlet ducts should have a straight length of at least 2.5 times the hydraulic diameter to allow the velocity profile to diffuse, giving static regain, i.e. higher static pressure but lower velocity pressure. [Source: Svenska inneklimatinstitutet & SINTEF]

Table 4: Relation between effective duct length and static regain.

effective length	No duct	25%	50%	75%	100%
static regain	0%	50%	80%	90%	100%

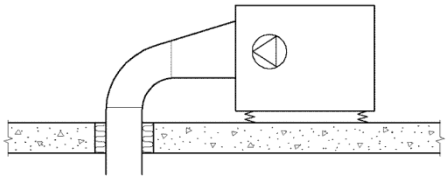


Figure 32: Example of a fan outlet with a reduction giving a satisfactory transition angle (<30°), and a bend that follows the air spin and the eccentric reduction. [Source: Svenska inneklimatinstitutet & SINTEF]

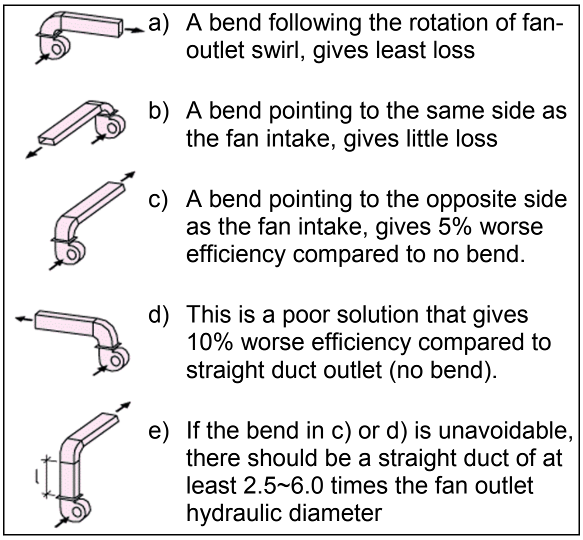


Figure 33: Impact of the efficiency of different outlets from centrifugal fans. [Source: Svenska inneklimatinstitutet & SINTEF]

[1] This is called *static pressure regain*

5 Ventilation system components – Pressure loss

5.1 Total system pressure drop – Rules of thumb

The single most important means of reducing fan power is by minimizing flow resistance.

Table 5 lists some rule-of-thumb component pressure drops in conventional mechanical ventilation systems in large buildings.

Table 5: Rule-of-thumb component pressure drops for large buildings

		Poor design	Typical design	Good design	
AHU	Face velocity	2.5	2.0	1.5	m/s
AHU	Filter EU3 bag	80	70	50	Pa
AHU	Filter EU5 bag	140	115	75	Pa
AHU	Filter EU9 bag	190-250	160	110	Pa
AHU	Rotary heat exchanger	200-250	150	90-100	Pa
AHU	Heating battery	120	80	40	Pa
AHU	Cooling battery	140	100	60	Pa
AHU	Humidifier	60	40	20	Pa
AHU	Fan silencer	80-235	50	30	Pa
AHU	Total AHU internal Δp	670	420	250	Pa
Air distribution	Ductwork (supply or exhaust)	300-490	200-230	100-115	Pa
Air distribution	Duct silencers	15	10	0	Pa
Air distribution	VAV box	112	112	25	Pa
Air distribution	Terminal reheat battery	105	50	0	Pa
Air distribution	Plenum box	100	50	30	Pa
Air distribution	ATD (supply or exhaust)	70	50	20	Pa
Air distribution	Exhaust stack/jet	175	175	175	Pa
	Fan system efficiency	40 %	50 %	63 %	%
	SFP	6	3	1.3	kPa

Table 6 shows the application of aggregating component pressure drops in a specific building with balanced ventilation and heat recovery. It illustrates that the total pressure loss though ventilation systems is roughly shared equally between the AHU and the distribution system. Much of the pressure drop can be avoided by astute aerodynamic design and by selecting an AHU with larger cross sectional area.

In hybrid ventilation systems, the contribution from natural driving forces (wind and stack effect) accounts for less than 1% of the energy savings in comparison to conventional ventilation systems with high pressure loss. The remaining 99% of the savings is actually a result of reduced flow resistance. Furthermore, all ventilation systems, irrespective of flow resistance, can potentially experience and exploit the same natural driving forces. Hybrid systems can therefore equally be called 'very low pressure loss' systems.

Table 6: Example of aggregating component pressure drops though a ventilation system in a large building, at design flow rate. [31][32]

				Poor design	Good design	Hybrid vent.
Supply	Distribution	Inlet louvre & duct		70	25	0
Supply	AHU	Filter section F5-F7*		250	50	27
Supply	AHU	Heat exchanger		250	100	13
Supply	AHU	Heating coil		100	40	0
Supply	AHU	System effect, →fan		30	0	0
Supply	AHU	Silencer/attenuator		200	0	0
Supply	Distribution	System effect, fan→		330	0	0
Supply	Distribution	Supply ductwork		150	100	1
Supply	Distribution	Terminals (ATD)		50	30	12
Exhaust	Distribution	Terminals (ATD)		30	20	0
Exhaust	Distribution	Extract ductwork		120	80	1
Exhaust	AHU	Silencer/attenuator		100	0	0
Exhaust	AHU	Filter section F5-F7*		250	50	0
Exhaust	AHU	Heat exchanger		250	100	13
Exhaust	AHU	System effect, →fan		30	0	0
Exhaust	Distribution	System effect, fan→		330	30	0
Exhaust	Distribution	Outlet duct+louvre		250	20	17
A: Sum distribution syst.			Pa	1330	305	31
B: Sum AHU			Pa	1460	340	53
C: Natural driving forces			Pa	ignored	ignored	-4
Sum Total (A+B+C)			Pa	**2790**	**645**	**84**
Fan system efficiency			%	**28 %**	**63 %**	**40 %**
SFP			kW/(m³/s)	**10**	**1**	**0.2**

* Final filter pressure drop before replacement

Systems in single family housing typically have lower pressure loss due to air distribution than indicated above. Good-practice systems typically have ~70 Pa external pressure drop in supply or exhaust duct systems. Thus the total pressure loss for residential balanced ventilation systems with heat recovery is typically 400 Pa. On the other hand, these small systems have lower fan system efficiency, typically 15% with traditional AC motors. This gives a typical SFP of approx. 2.7 kW/(m³/s) at normal flow rate.

5.2 Filters

5.2.1 Filter types and their aerodynamic properties

Air filters have two functions. They clean the supply air and protect the AHU components from soiling, especially the heat exchangers and fans. The pressure drop over filters gradually increases as they become dirty, and can exceed 250 Pa. Filters therefore constitute a large portion of the pressure drop in the ventilation system.

5.2.1.1 Bag filters

The most common filters are bag filters of glass fibre or plastic material. Some use an electrostatic material, which has a tendency to lose performance over time, and is not generally recommended. Others have an additional layer of active coal, which is recommended for areas with pollution from traffic etc. Bag filters come in different filtration grades:

Fine filters capture particles of less than 1 micrometer. The filters' particle removal efficiency is relatively independent of air speed. In the interests of energy-efficiency, the filter surface area should be as large as possible to reduce the pressure drop across filters. Bags with a tapered stitch form (i.e. cross-section 'VVV...' as opposed to 'UUU…') allow more uniform air flow through the filter bag area, and thus lower pressure drop.

Coarse filters capture particles larger than 1 micrometer. Momentum contributes to the filtration effect. A certain air velocity is therefore required to achieve the desired filtration effect. One can not simply increase the filter surface to reduce the pressure drop over a coarse filter, as this will reduce flow momentum and filtration effectiveness. For the same reason, coarse filters are not suitable in systems with considerable variation in air flow rate. Because of the different principle, it is rarely economical to place a coarse filter in front of a fine filter. Not only will this increase the total filter pressure drop, but it will usually only slightly increase the life of the fine filter. The need for a longer AHU, and the cost of having more filters to maintain, are other arguments against this solution.

5.2.1.2 Electrostatic precipitators (ESPs)

There are different types of EPS filters. They generally have a lower pressure drop than bag filters, and are therefore more energy-efficient. Another benefit is that they are highly effective, and are less likely to harbour bacteria than bag or HEPA filters. Typical problems with EPS filters are that they let large particles pass, and lumps of charged particles can separate from the electrode and enter the supply air. One possible solution to this is to allow air to pass through a chamber at low velocity after the filter, such that large particles & lumps fall to the floor of the regularly cleaned chamber. Another solution is to use a coarse filter before the electrostatic precipitator. Precipitators are also known to produce toxic ozone and NO_x to different degrees. Nevertheless, the indoor concentration of ozone can still be below outdoor concentrations.

5.2.2 Recommendations for air filters

- Consider the need for the filter carefully. In many cases it is sufficient with a fine filter on the supply side and on the exhaust side. Remember that coarse filters and fine filters in series is rarely a cost-effective solution.
- Consider using an electrostatic precipitator.
- Use bag filters with large surface area, i.e. low air velocity over the filters.
- Determine the filter changing frequency (i.e. final filter pressure) from economic considerations. Ask the filter vendor for assistance to carry out such calculations.
- Consider omitting the exhaust filter in buildings with best-practice routines for cleaning of the premises and ventilation system

5.3 Other AHU components — Heating/cooling coils and heat exchangers

5.3.1 Types and their aerodynamic properties

Heating/cooling coils (batteries) and heat exchangers account for a significant proportion of system resistance. Low air speed through these components is therefore important. It has been conventional practice to dimension an air velocity of 2.5 m/s through the AHU. However, a lower velocity, near 1.5 m/s, will in many cases give optimal life-cycle cost (LCC) . Lower air speed also increases the thermal efficiency of coils/heat exchangers.

For rotary heat exchangers, a positive pressure difference between the exhaust and supply sides of the AHU will result in leakage of contaminated exhaust air into the supply air stream via the brushes. This can be prevented by astute dimensioning of AHU components, or fitting a throttling damper (see Figure 34) such that the pressure difference $p_3–p_2$ is not positive. It is generally the responsibility of the AHU vendor to dimension this additional pressure drop. However, this is seldom done correctly, as it is sensitive to the external pressure drops in the four ducts. HVAC designers should double-check that this is properly calculated.

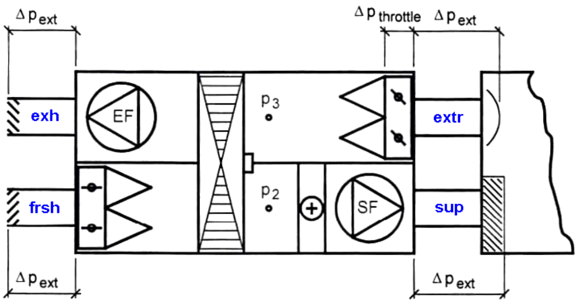


Figure 34: Illustration of an AHU with rotary heat exchanger and extra throttling ($\Delta p_{throttle}$) on the extract air side of the rotary heat exchanger to prevent wrong air leakage direction, $p_3 \rightarrow p_2$ [Fig. © Förening V]

5.3.2 Recommendations

- Determine unit size from an economic assessment where you take into account both investment and operating costs (LCC) . Ask the AHU vendor for assistance with such calculations. Include structural/building costs. Optimal units have a nominal air velocity of ≤1.5 m/s.
- Take into account the fact that reduced velocity increases the thermal efficiency of coils & heat exchangers. It may be possible to reduce the number of coil rows, thus further reducing pressure loss.
- If the heat recovery is very efficient (temperature ratio > 80%), you can consider omitting the heating coil and thereby further reduce the pressure loss and AHU length. This measure should be combined with robust automation that stops the unit and closes the air inlet damper in case of risk of frost.
- Consider the option of a bypass in parallel with heat exchangers. The bypass is opened when the heat exchanger is not in use, thus significantly reducing pressure loss. This is often impractical due to lack of space.
- For AHUs with rotary heat exchangers, check that the AHU's internal component pressure drops are properly sized to prevent leakage of extract air into the supply air stream. Additional throttling is calculated according to [2] or [36].

5.4 Air transport & distribution system (ductwork)

5.4.1 Distribution systems and their aerodynamic properties

The lowest pressure drops can be achieved by exploiting building-integrated air transport conduits such as corridors, stair wells for air transfer, or underfloor air supply plenums [25]. However, the remainder of this guide is dedicated to ductwork systems, which have become ubiquitous for other practical reasons.

The design of the duct system has a great impact on the ventilation function and fan energy. The duct path with the greatest flow resistance from the AHU to any terminal is called the '*critical path*' (shown by a dashed line in Figure 35). This path determines the pressure drop in the whole duct network. If one changes the ductwork design along the critical path to reduce its pressure drop, the critical path may move to another terminal. The new critical path ductwork can, in turn, be modified to further reduce the system pressure drop. Optimizing a duct system therefore involves a trial-and-error process whereby software is used to calculate the pressure drop for variations of the duct system. It is beneficial that the HVAC designer calculates pressure drop at an early stage in the project so that reasonable choices can be made about the location & size of technical rooms and the main service routes. The calculations should be updated in the detail design stage, and when later changes are made.

Figure 35 & Figure 36 show how the pressure drop in a duct network can be drastically reduced by small changes along the critical path.

Figure 35 is the work of a duct designer who was not fully aware of the impact of design on system pressure. The total pressure drop of 168 Pa is determined by the pressure drop along the critical path to the 'index terminal' (fully open, with a pressure drop of 10 Pa). Over half of the total pressure drop is the result of two main branches (4-way 'X'-junctions) with poor aerodynamic form. The total pressure can be more than halved by replacing these with 90° bends, and dimensioning the ductwork more generously using the *'⅓-Rule'*[1] (Figure 36). Fan power is more than halved as a result.

Straight main ducts should be sized so that the pressure drop is below 0.8 Pa/m. Bends and branches along the critical path should be chosen so that none exceeds 5 Pa loss. Moreover, the pressure drop for the index terminal should be selected to be as low as possible without compromising the needs for balancing and throw lengths. You should aim for a supply or return duct system total pressure drop of 150 to 250 Pa for large systems, and 70 Pa for dwellings (see Table 5 on page 23).

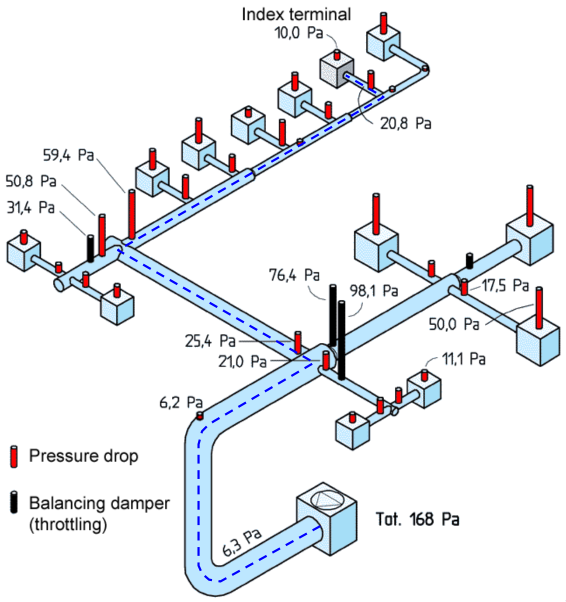


Figure 35: Pressures in a poorly designed system. Required fan pressure is 168 Pa. Unfortunately this type of solution is very common. Heavy throttling of 2 dampers may necessitate extra silencers. The critical path is shown by a dashed line [Fig. © SINTEF]

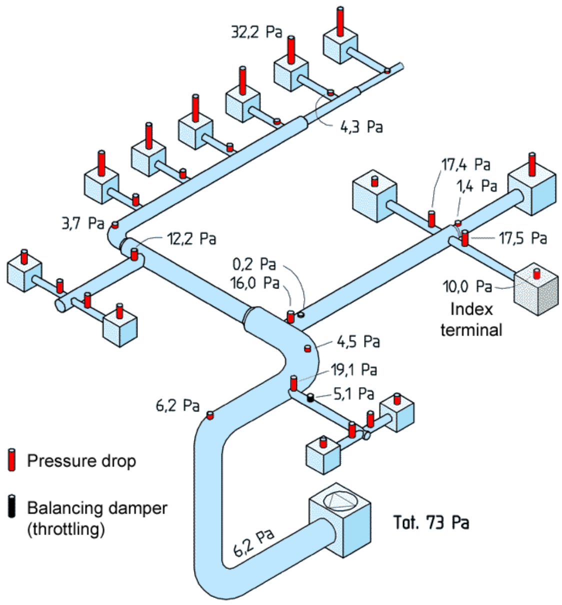


Figure 36: Optimal solution. Pressure drop is slashed from 168 Pa to 73 Pa. Observe that the pressure drop reduction measures have moved the critical path (Index ATD has Δp=10 Pa). The amount of throttling/ balancing is reduced, which further reduces noise. The system does not need balancing dampers, simplifying commissioning and further reducing noise [Fig. © SINTEF]

[1] The *⅓-Rule* for round ductwork: Duct diameter is kept constant along a duct until a branch (tee) flow rate exceeds ⅓ of the combined flow, in which case the duct dimension is reduced by 1 step downstream. The start of each duct branch is dimensioned for a given m/s or Pa/m. This duct dimensioning method gives results very similar to the static regain method. Static pressure is pretty constant along the duct, so little balancing is needed.

5.4.2 Principles of efficient duct system design

- Try to minimize duct length and the number of bends, when designing the duct system.
- Use rigid-walled round ductwork, ideally with rubber gaskets. This has many advantages over rectangular ductwork, including lower pressure drop for a given weight/cost of duct, and less air leakage. Avoid extensive use of flexible ducting.
- Use optimal components along the critical path. In particular, use bends instead of tees. If possible, use two 45° bends instead of a 90° elbow. Two 90° bends in series should have a distance of at least three times the duct diameter.
- Use ATDs with a pressure drop of ≤ 30 Pa. Low velocity supply terminals for displacement ventilation operate with lower pressure drop than high velocity jet supply terminals for mixing ventilation. Displacement ventilation has the added advantage of high ventilation efficiency [39].
- Any balancing dampers along the critical path should be fully open. Each duct run shall have at least one take-off (branch) that has a fully open damper/ATD, or no damper at all.
  → In the case of CAV systems, this is achieved by balancing using the proportional method [37].
  → In the case of VAV systems, this is achieved with SPR control.
- Dimension main ducts for a pressure loss of ≤ 0.8 Pa/m.
- Both supply and exhaust duct systems require regular inspection and cleaning [3],[11],[12],[38]. Failure to do so leads to both poor indoor air quality and higher flow resistance.

5.5 Exhaust outlets (incl. roof stacks)

5.5.1 Outlets and their aerodynamic properties

Jet roof stacks (jet hoods; Figure 37a) are often used as high-velocity exhaust air outlets to prevent the risk of the exhaust air mixing with fresh intake air. Jet stacks are often dimensioned with the pressure loss of over 100 Pa, which causes a significant rise in fan energy [SFP increases by +0.2 kW/(m³/s) assuming 50% fan system efficiency]. One should rather locate the outlet at a sufficient distance from the air intake so as to render a jet stack unnecessary, without using long ducts for air inlet and outlet (e.g. guidelines in Appendix A of EN 13379). Moreover, alternative stack designs are available with low pressure loss (e.g. Figure 37b).

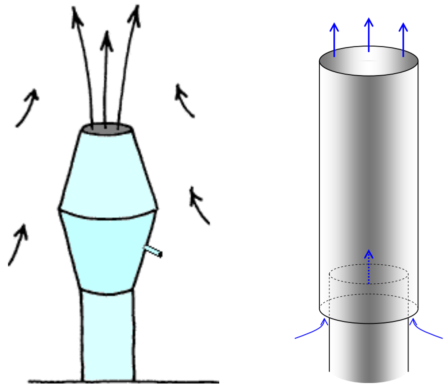


Figure 37: (a) jet stack with high pressure loss (b) exhaust stack with low pressure loss Both stacks have drainage for rain [Figs. © SINTEF]

If use of jet stack is unavoidable, such as for laboratory ventilation, they should be elected with as little outlet velocity as possible without risk of short circuiting to air intakes[35].

5.5.2 Recommendations for outlets

- Locate air intakes and exhausts so as to prevent short-circuiting from exhaust to inlet, while keeping the inlet & exhaust ducts as short as possible.
- If high-velocity jet stacts are unavoidable, try to design to keep the pressure drop as low as possible (maximum 50 Pa).

5.6 Silencers (attenuators)

5.6.1 Silencer types and their performance

The relationship between changes in air velocity and noise generation in ductwork is:

$$\Delta L_w = 10 \log \left(\frac{v_2}{v_1} \right)^6 \quad \text{[dB]} \qquad (17)$$

where

ΔL_w = change in sound power level [dB]
v_1 = original air speed [m/s]
v_2 = new air speed [m/s]

This means that a 25% reduction in velocity results in approx 7 dB less generated noise, which corresponds to the noise reduction in a 0.5 meter long silencer.

In practice, this means that systems with low SFP have less problems with generated noise and thus less need for silencers in the system (cross-talk can be more problematic, however, due to easier noise propagation in wider ducts).

Figure 38: (a) Rectangular silencer with baffles/splitters (b) Circular silencer without baffles [Fig. © Lindab]

Ventilation noise should ideally be attenuated/ avoided at source. For this reason, noise reduction can either be central at the AHU (fan noise), or local in the ductwork (flow generated noise):

5.6.1.1 AHU silencers (central noise reduction)

Standard AHU silencers with internal baffles can have a pressure drop of 50 to 100 Pa (Figure 38a). This is often unnecessarily energy-intensive. A better alternative is low pressure-drop silencers fitted immediately after the fan:

Inlet bell silencers and *outlet regain silencers* have optimal aerodynamic form (Figure 39). Outlet regain silencers also regain much of the fan dynamic pressure (velocity pressure) as static pressure. This reduces the system-effect pressure drop compared to AHUs without static pressure regain after the fan. For best performance, these attenuators should be custom designed for, and possibly integrated into, the AHU.

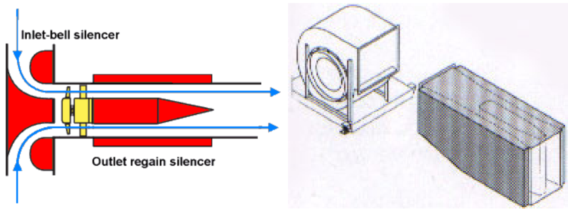


Figure 39: (a) axial fan with inlet-bell and outlet regain silencers (coloured red) [Fig. © M&I Air Systems Engineering]
(b) centrifugal fan with outlet regain silencer

Mass-produced silencers: There are also more conventional ductwork AHU silencers with minimal inlet and outlet loss because of their shape (e.g. Figure 38b). Such silencers can be sized for a maximum pressure drop of 20 Pa.

If there are duct bends near the fan inlet or outlet, it might be appropriate to use a circular elbow silencer or rectangular bend with curved baffles or acoustic turning vanes.

5.6.1.2 Duct silencers (local noise reduction)

A duct silencer is in principle a duct component with its inside surfaces covered with sound absorbing material. The silencer should have the same internal free area as the connecting ductwork, so as to minimise flow resistance. The silencer should be accessible for inspection and cleaning! Silencers with baffles obstruct cleaning (Figure 38b).

5.6.2 Recommendations for silencers

- Calculate noise levels, ideally with the same software that is used for duct system design and pressure drop calculations. Ventilation systems with low pressure-drop have little noise.
- AHU silencers should have maximum 20 Pa pressure drop. Ideally, try to select or custom-build aerodynamically shaped AHU regain silencers that give static pressure regain after the fan.
- Select duct silencers that are as long as possible.
- The need for noise attenuation is less stringent in energy-efficient ventilation systems with a low velocity and good aerodynamic solutions.

6 Placement of responsibility

In order to successfully implement SFP requirements in the building industry, the following stakeholders must be aware of their specific responsibilities:

- *Governments/building authorities* should implement maximum SFP limits in their building codes, and ideally implement mandatory building inspection/auditing schemes.
- *AHU manufacturers* should make available software for accurately calculating SFP_e and SFP_v for user defined operating points. Such software should ideally be certified (e.g. EUROVENT). They should also ensure that their AHUs enable easy & accurate measurement of airflow, fan power, pressure rise, and SFP.
- *HVAC designers* should set SFP targets for each building they design, to satisfy energy performance regulations. They should also calculate duct system pressure drops and flow rates and use this as input to calculate SFP with AHU vendors' software. They should take into account all ventilation fans in the building, also toilet extract fans, kitchen hoods, etc.
- *HVAC contractors* should properly commission AHUs after installation and report measured fan performance in accordance with agreed trade standards.
- *Auditors/inspectors* should check commissioning reports properly, and perform random spot-check measurements.

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