An analysis of the efficiency of centrifugal fans

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Foreword:

This document is intended to provide a technical assessment of the state of technological development of centrifugal fans, with particular attention to the fans intended for application in the HVAC&R business, and including some additional remarks extending to the use of common types of motors in other fan types. It is intended to identify the currently available technological margins and constraints for improvement of the fan efficiency.

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Fan definition – some consequences

A fan is generally defined as a power-driven device designed to move a continuous flow of gas between two separate spaces, where there is a difference of pressure in between.

To generate such a movement, the fan must impart some velocity to the otherwise stationary gas, and spend some energy to increase the kinetic energy of the gas. This energy is on top of the work done to move the gas volume against the pressure difference, existing between the suction and discharge environment.

A fan moving air between different spaces have a stationary part, besides the impeller, interacting with the impeller and the airflow, and connecting the fan with the partition which separates spaces at different pressure. By interacting with the impeller and the airflow, the stationary part (e.g. orifice nozzle, scroll case, tubular case etc.) contributes to the transfer of energy to the gas flow.

There are some special fan types which represent exceptions to this basic definition, because they just impart kinetic energy to the gas flow. Typical examples of these special fan types are Jet-fans, circulator fans and comfort fans. Some of these special fans may sometimes not have an aerodynamically active stationary part. This is never happening with Jet fans, so we can say that all the fans included within the scope of the current Regulation 327/11/EC do include an impeller and some kind of stationary aerodynamic part.

Total efficiency vs. static efficiency. – Fans with scroll and fans without scroll case.

Fans with case, either with backward-curved or with forward-curved impellers, are essentially intended to direct the flow toward and into a discharge duct. In this case of application, imparting kinetic energy to the flow is not just unavoidable, but useful, as the delivered airflow must move along the discharge duct, and this requires kinetic energy.

In this case, the energy balance of the fan is measured as “total system efficiency”: by multiplying the volume flow rate by the total pressure generated by the fan, the useful power becomes the sum of the power used to move a volume flow rate against an increase in static pressure, plus the flow of kinetic energy imparted to the airflow. This useful power is then divided by the electrical power input to the drive system, to compute the total system efficiency.

When the fan is used to deliver a gas flow into an open space, or into a large plenum, then the kinetic energy imparted to the flow is still unavoidable, but cannot be used effectively, as it is immediately dissipated, as soon as the discharge airflow is diffused into the open space. The useful power is then calculated multiplying the fan volume flow rate only by the static pressure generated by the fan, and so the flow of kinetic energy is not accounted as useful power output, which is consistent, as it is always immediately wasted. The useful power is then divided again by the electrical power input, to compute the “static system efficiency” of the fan.

Plenum fans and Plug fans are fans where the stationary parts do not extend downstream of the impeller, and so cannot guide the airflow into a discharge duct or channel. The airflow is immediately distributed into the discharge environment, and any remaining kinetic energy of the airflow is immediately dissipated. For this reasons, plug fans and plenum fans are always rated in terms of static efficiency, and are optimally used when the airflow downstream of the fan must be immediately distributed into a wide volume, like an open space, or the plenum upstream of a wide bank of filters, or a heat exchanger.

Using these fans to blow air into a duct is not an efficient approach, because this means, as a first step, dissipating the kinetic energy at the impeller discharge, and then using a part of the static pressure provided to the airflow, to convert it again into kinetic energy, and move the airflow along the duct.

The electrically-driven fan as a compound machine

When evaluating the technological limits of fans driven by electric motors, it is useful to have in mind the separate contributions of the aerodynamic parts ("Fan") and that of the drive system ("D/S", meaning the combination of an electric motor with an electronic speed controller, when applicable, plus a mechanical transmission, in some fans).

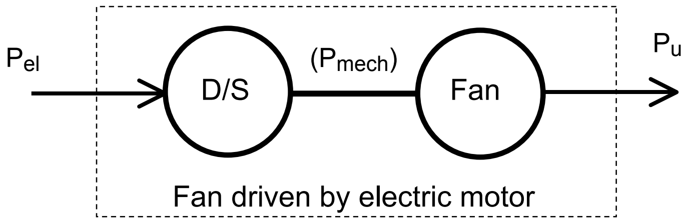


Fig. 1: Staged structure of the electrically-driven fan,

Sometimes the motor is deeply integrated inside the fan, and its size and shape impact either positively or negatively the aerodynamic efficiency of the assembly but, in any case, the system efficiency shall always be the product of the aerodynamic efficiency multiplied by the efficiency of the drive system:

$$\eta_s = \eta_{D/S} \cdot \eta_a$$

When the motor is integrated inside the fan, its shape and position contribute to guide the airflow through the fan, and may change, most often to the worse, the efficiency which the assembly of the same impeller and case may have, when run without the bulk of the motor.

Under such conditions, the aerodynamic efficiency of the fan may be assumed to be

$$\eta_a = \eta_{aB} \cdot C_I$$

i.e. the product of the efficiency of the basic fan η_{aB} (the same impeller and case without the motor or other interference-generating components, like bearing struts), multiplied by a coefficient C_I representing the effect of the aerodynamic interference of the motor bulk, or of any supporting structure for the motor or shaft bearings. Such a coefficient if not, strictly speaking, another multiplying efficiency, but a measure of the change in aerodynamic efficiency of the fan, and this is demonstrated by the fact that sometimes (actually, very seldom) it may be larger than unity.

Actual efficiencies of FC and BC fans

In a number of old but well-known publications, the difference between the aerodynamic efficiency (peak value, or Best Efficiency, or efficiency at the Best Efficiency duty Point) of backward-curved (BC) and forward-curved (FC) fans is significantly exaggerated, stating that the efficiency of an FC fan is in the order of 50% while a BC fan can reach 80%.

If state-of the art products are actually compared at the same duty point, or at least at the same level of power input, the difference is typically less than half as much.

The actual values can change significantly, obviously depending on the quality of design and construction, but are also significantly depending on the size and speed (i.e. on the Reynolds nr.) at the fan operating condition.

In physics, “small is beautiful” does not work, and smaller fans are less efficient than larger and/or faster ones, within the same general type.

For a reliable estimate of the difference in efficiency and power consumption, for a given duty, a comparison should be done between values which apply to products of similar power and equivalent performance.

The same applies to electric motors and to their driving electronics. Also the efficiency of the electric motors, for a chosen technology, is generally increasing with size and power.

As a result, inside the ISO 12759 standard, an attempt was done to define curves of efficiency vs. power input, in an attempt to define efficiency curves which might be equally challenging to achieve, across the power range, assuming that input power might be a good practical index of the fan Reynolds number, at the optimum operating point.

This was partly successful, but there are still problems, with fans belonging to the same broad category and of the same power input, but designed for different specific diameters and speeds, finding different efficiency technological limits. (Note: see also the technical paper from J. Anschuetz, published by EVIA, on this subject).

This is one of the reasons why fans designed for some “extreme” operating conditions, e.g. like the “high-pressure” BC centrifugal fans, otherwise called “narrow-impeller” BC fans, have sometimes real problems to achieve the efficiency targets set for 2015 by the current regulation 327/11/EC.

Looking at fans with a power input around 1 kW, the aerodynamic Total efficiency of a good-quality FC fan may be in the order of 65 % .

A PF fan (otherwise called, translating from a common German term, “BC centrifugal fan without case”) of similar power and up-to-date design, may have a peak Static aerodynamic efficiency of 72 %.

A BC fan with scroll, at the same power level, may have a Total aerodynamic efficiency of 76 %.

These figures are not absolute, but typical average values for benchmark-quality products, representative of the mass-produced fans for HVAC applications. The actual values may vary, because the efficiency resulting from the fan design process is not just a measure of the design quality, but also of a number of trade-offs with other design requirements which are different from optimum peak efficiency.

Fans may sometimes have to be designed to reduce their noise generation, even at the expense of their operating efficiency.

In many cases, the optimum size of fan, which should be adopted to geometrically scale a given fan design to get its best efficiency in the specified duty point, may not be fitted within given size constraints, which may also be constraints of cost-effectiveness of the overall machine incorporating the fan. The peak value of the fan efficiency may then have to be traded off, for an improvement of the fan efficiency in an operating condition with a larger flow coefficient, allowing a more acceptable power consumption while still using a smaller-sized fan for the same duty.

Another reason for compromising on the fan peak efficiency may be the need to extend the stable operating region of each fan size and model: the actual duty point of a fan, when operating in its real environment, may be known, at design or selection stage, with a margin of uncertainty, or the duty point may change with time, for any reason connected with the normal operation of the ventilation system, e.g. because air filters are progressively clogging, between installation and replacement.

Last but not least, stress levels and structural reasons may impose design solutions (including shape and dimensions of critical components like blades), which may interfere with the achievement of the best theoretical efficiency levels.

Stable operating conditions and best efficiency line

For each fan design and size, the performance in “cinematically similar duty points”, or duty points where the fan pressure is proportional to the volume flow rate squared,

$$P = Const \cdot Q^2$$

are connected by simple mathematical relations, based on the fact that, under such conditions, in the airflow fields, local velocity directions are the same and velocity modules are proportional to the fan rotational speed.

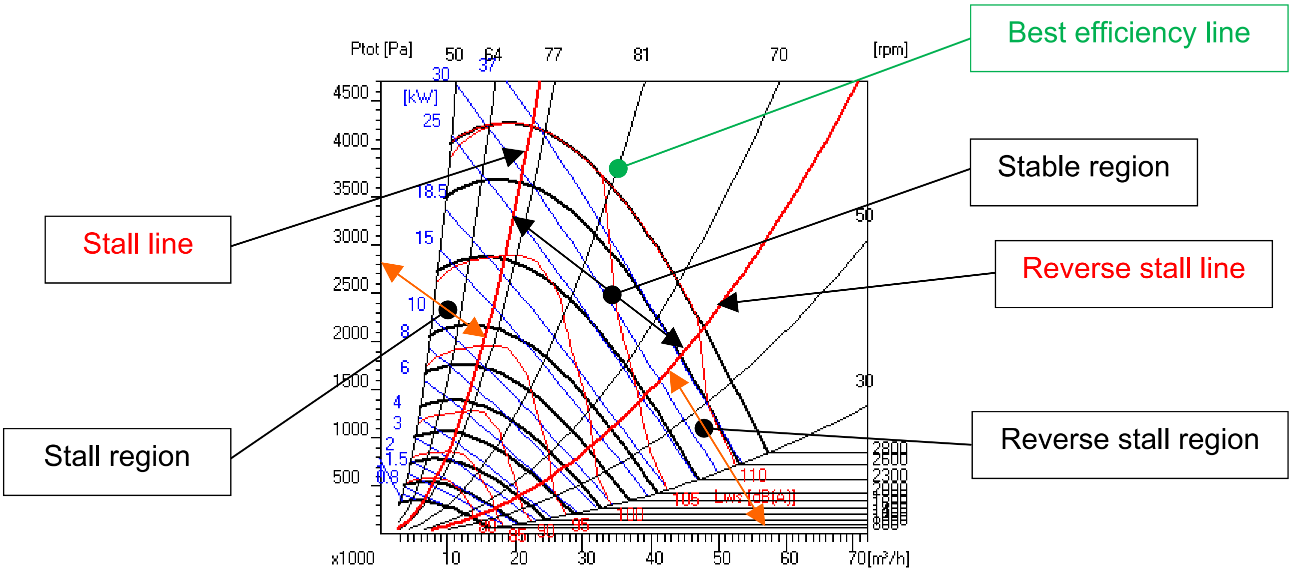


Fig. 2: Stable operating region and unstable areas for BC fan (linear diagram)

(See also the attached Nicotra Gebhardt Engineering Bulletin no. 003, for a more extensive, but still basic introduction to the fan cinematic similarity laws).

Each fan suffers from undesirable aerodynamic instabilities, when the duty point moves to the left or to the right of a stable band delimited by two parabolic lines, called the stall line (the left border of the stable region), and the reverse stall line (right-side border of the stable region).

Inside the stable region, there is a parabolic line where the efficiency of the fan is best, at each operating speed. As long as the evaluated efficiency is the efficiency referred to the mechanical power input to the fan impeller, without involving the drive system, the efficiency can be practically considered to be constant along the best efficiency parabolic line of each fan. There are anyway fans where the high mach number or just compressibility effects at high speed, or, more seldom, low Reynolds nr. effects at very low speed, may introduce measurable deviations in efficiency, along the best efficiency line.

The efficiency of the drive system, on the other side, is normally changing significantly with a change of fan speed, and thus changes the overall efficiency of the complete system.

The relative width of the stable-operation band, and how much its boundaries are far away from the best efficiency line, depend on the design of the impeller, and can make a fan design more or less easy to use effectively at its best efficiency, without meeting mechanical fatigue problems, which sometimes arise when the fan duty point is moved outside the stability boundaries.

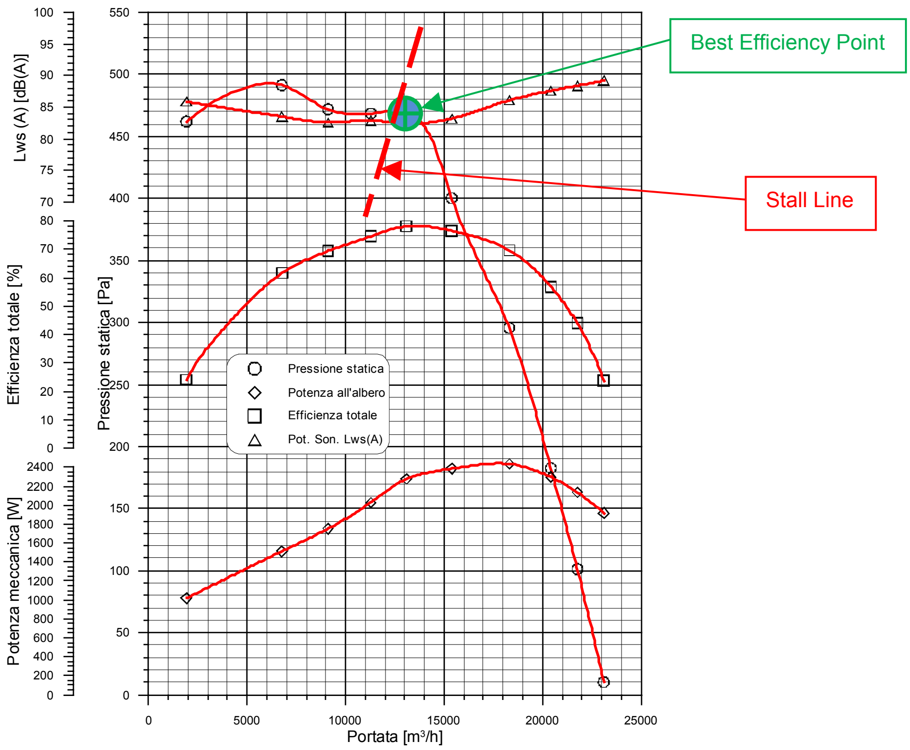


Fig. 3 Example of BC fan with deep stall and small margin between BEP and stall boundary

A fan with the stall line dangerously close to the B.E.P. (like the BC one shown above, having the stall point at 95% of the BEP volume flow rate at the same speed), may be difficult to use reasonably close to optimum efficiency, because of the high risk that any accidental deviation from the nominal design point may bring the actual duty point in a region of highly fluctuating pressure, which, in some critical conditions, like when the fan is run at a high speed and pressure (compared with the impeller structural limits), may lead to fatigue failures of the fan itself, or of other parts of the ventilation system.

The position of the stall line, relative to the B.E.P., is not related to the type of fan, and within a given fan type, it is not strictly correlated with the fan efficiency. Some of the easier tricks to improve fan efficiency, like improving three-dimensional alignment of the leading edge with the local airflow direction, when used without correction, may have a detrimental effect on the safety margin, between BEP and stall.

The claim that the FC fans have a stall point placed at a relatively high volume flow rate is based on the comparison with a BC fan of the same size; in this case, the optimum volume flow rate, at a given pressure, is also considerably smaller for the BC fan. When the FC fan is compared with a BC fan optimized for the same duty, the stall line is also approaching the design duty, and the margin between duty point and stall line depends once again on the fan design quality.

The stall is a condition in which the airflow through the fan is intrinsically unstable, resulting in fluctuating pressure. While the stall is sometimes producing a visible “dip” in the time-averaged pressure vs. volume curve, this is not always true. Typically, fans with airfoil blades have a more gentle stall, which often is not visible in the measured pressure curve. On the other side, the upward-rising portion of the characteristic curve of a forward-curved fan is not a “dip” due to the stall, but to the increase of the energy exchange between the blades and the airflow, with increasing volume flow rate (increase of Euler work), due to the blade geometry.

On a FC fan, most of the upward-rising part of the curve is actually inside the stable region. A properly designed FC fan may be stable between 25% and 85% or more, of the free delivery flow.

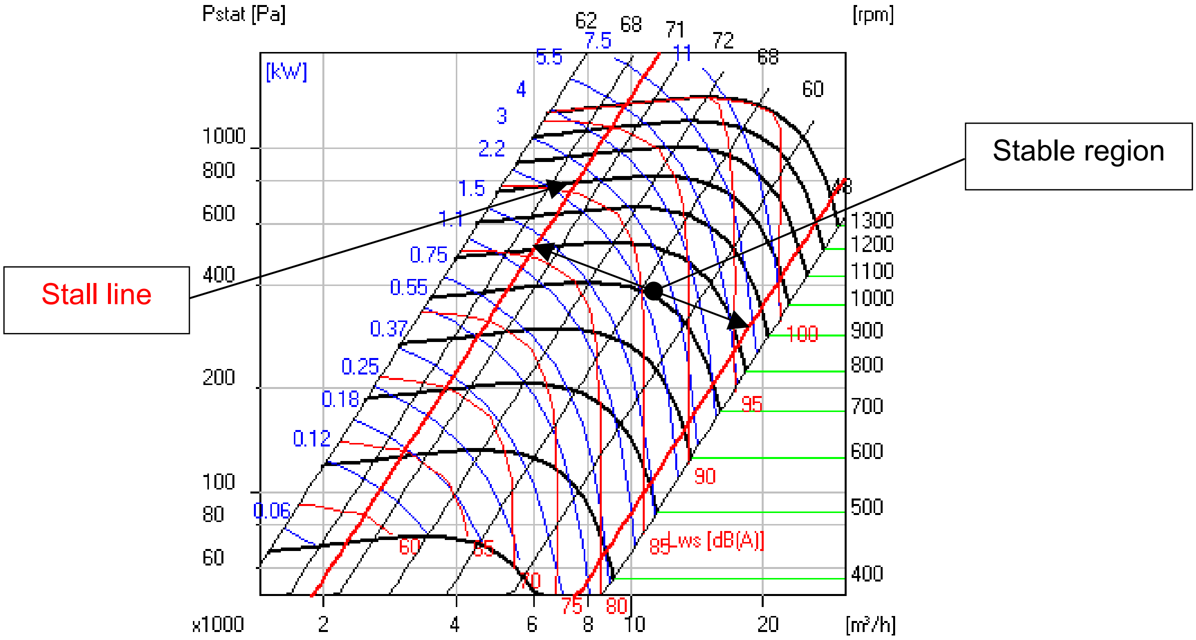


Fig. 4: Stall line on the variable-speed diagram of as FC fan (logarithmic diagram)

Relative size of optimally-sized fans: BC, FC and PF.

Fans which are geometrically similar, but have different sizes, have their bands of stable operation and their best efficiency lines placed at values of volume flow rate which, compared at the same fan pressure, are obviously increasing with the fan size.

If the fans of a range are in perfect geometrical similarity, the best efficiency of a larger-sized fan is achieved, at a given pressure, at a volume flow rate which is larger than that of the best efficiency point of the smaller fan, at the same pressure, by a ratio which is equal to the size-ratio squared.

Under the assumption that the speed of the larger fan is inversely proportional to the fan diameter

$$N_2 = N_1 \times \left(\frac{D_1}{D_2} \right)$$

which implies also that the fan peripheral speed values are the same

$$TS_2 = TS_1$$

and disregarding the small dependency of the fan aerodynamic efficiency with the fan size and Re number, if we assume the best efficiency unchanged:

$$\eta_{2Opt} = \eta_{1Opt}$$

we can calculate the new operating condition of the larger fan (2), from that of the smaller one (1):

$$P_2 = P_1$$

$$Q_2 = Q_1 \times \left(\frac{D_2}{D_1}\right)^2$$

$$W_2 = W_1 \times \left(\frac{D_2}{D_1}\right)^2$$

If a series of geometrically similar fans are scaled with the diameters according to the R20 series of normal numbers, i.e. in a constant ratio of 1.125, each size provides its best efficiency, at a given pressure, at a volume flow rate which is $1.125^2 = 1.265$ times larger than that of the immediately preceding size.

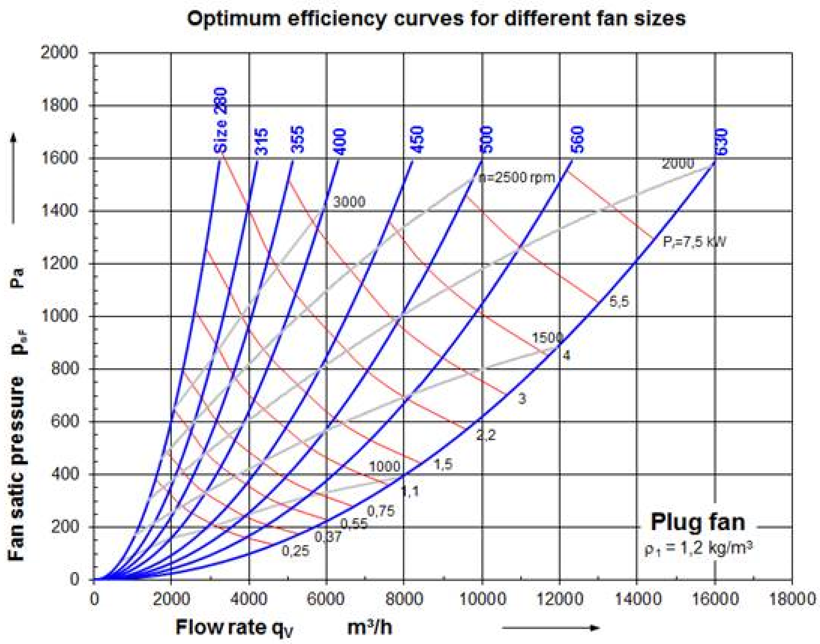
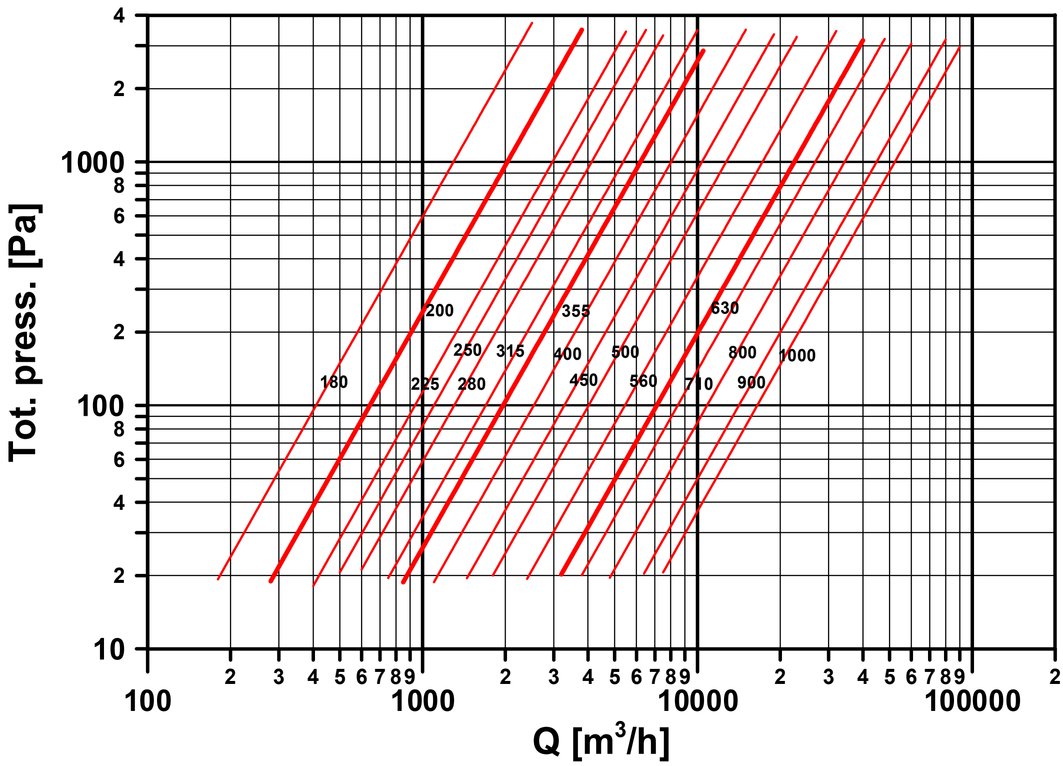


Fig. 5: Distribution of best-efficiency lines from a range of geometrically similar fans (linear diagram)

The parabolic curves of best efficiency for each fan size are transformed into straight, inclined lines, if the curves are plotted in a log-log diagram.

The diagram in Fig. 6, at the top of the following page, gives an example of how the best efficiency curves of a range of BC fans with scroll-case show up, when the impellers are not perfectly similar to each other. The principle that the curve of each size moves to higher volume flow-rate levels is still satisfied, but the curves are a bit less regularly spaced. The numbers written close to the middle of each line are the nominal fan sizes, i.e. the diameters of the different impellers.

Fig. 6: RDN series (BC with Housing) optimum efficiency lines



Something similar happens with a family of best-efficiency lines for a range of FC fans with scroll case.

Fig. 7: ADN series (FC with Housing) optimum efficiency lines

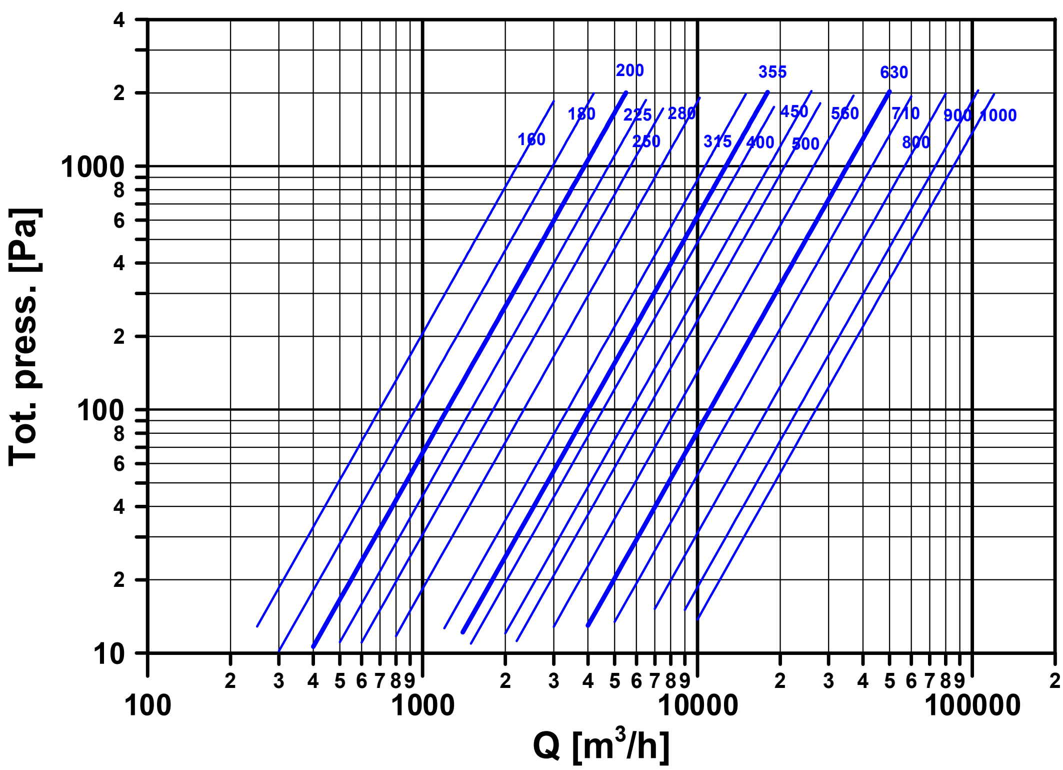
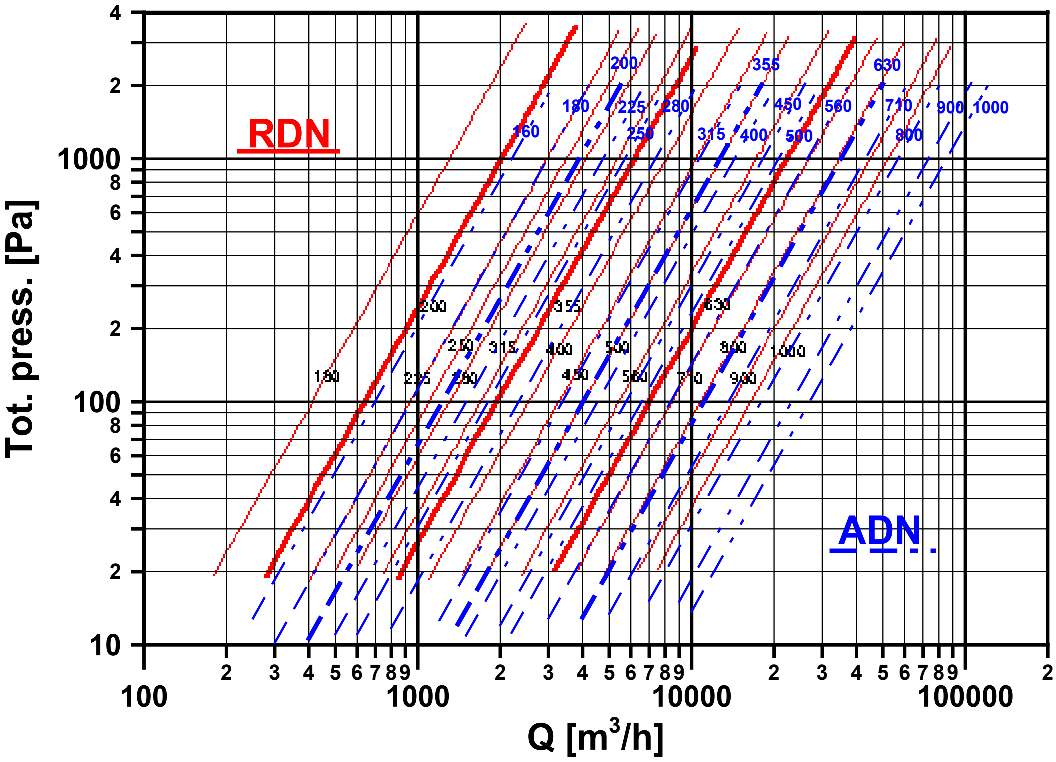


Fig. 8: Comparison between BC and FC optimum efficiency lines



If we try to overlap the two diagrams (Fig.8), as shown in the example above, we see that the FC fans provide their best efficiency at or close to the place of the best efficiency duty points of the BC fans being between two and three size steps larger: e.g. the best efficiency of a 355 mm FC fan size is achieved close to the best efficiency line of a BC fan with scroll of size 500 mm.

A similar result may be confirmed by looking into the results of selecting more modern product ranges, e.g. by looking at the product ranges now available from the Nicotra Gebhardt group.[1]

The same best efficiency line is achieved by a PF (BC without case) using a 630 mm impeller.

[1] See http://www.nicotra-gebhardt.com/

If we compare only the physical size of the three fans, without accounting for the additional space required around the suction or discharge openings, which is needed to avoid a significant performance penalty from the interaction with the surrounding walls, we have a picture like that visible in Fig. 9 .

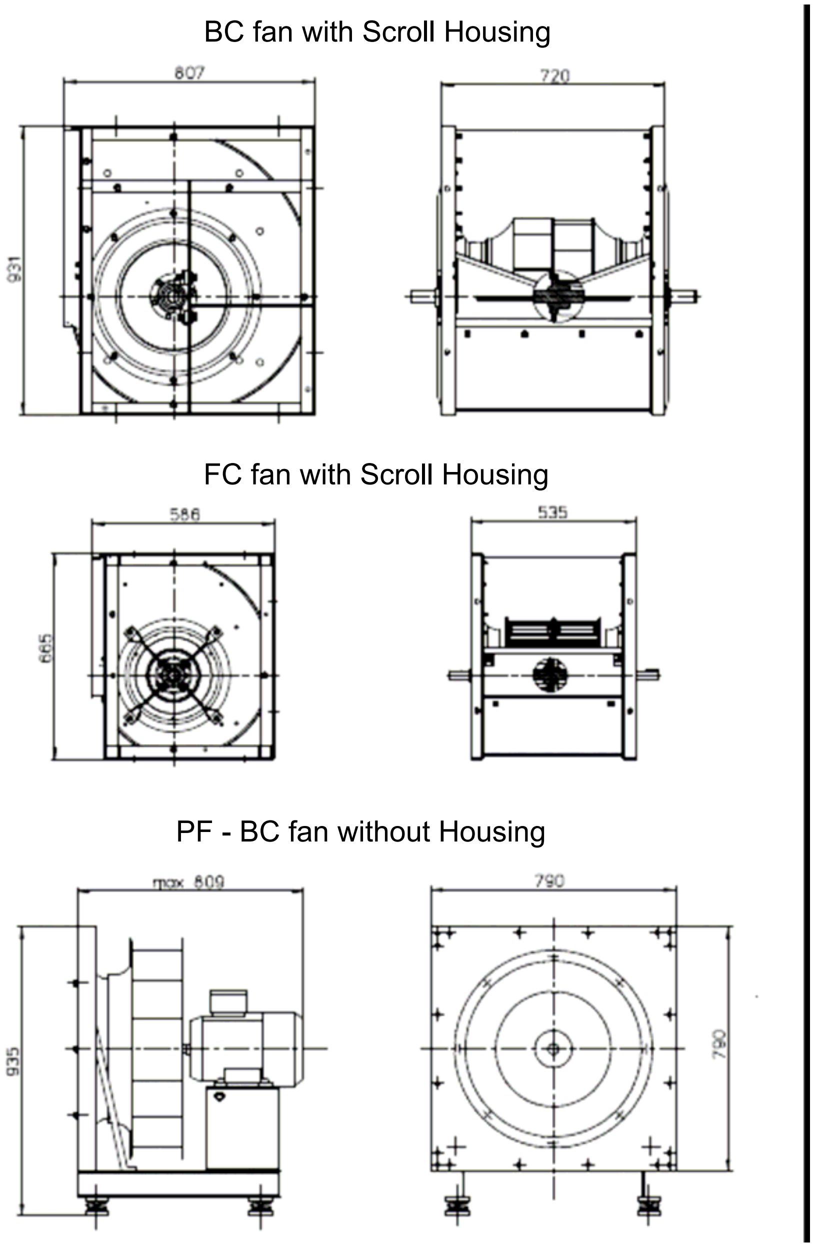


Fig. 9 : Comparison of fan size, between three different types, optimised for the same duty

For a proper comparison of the space required for installation and operation of the fan, the motor installation (whether integrated or connected with a transmission) of these centrifugal fan types should also be accounted for, together with the space, upstream of both the suction and discharge openings, which my be required to prevent unacceptable interference and performance worsening.

This would point out the only dimensional advantage of the plug fan, i.e. the lower length in the main direction of the airflow.

Only when the BC fan size can be optimised to operate the fan at its best efficiency at the specified duty, the potential improvement offered by the use of a BC fan can be achieved. If, e.g., any height constraint prevents the use of the optimal BC fan size, use of an undersized BC fan of size 355 would waste the majority if not all the advantage of the BC fan.

In many cases, the designer is not free to choose the absolute best fan size for each installation, and a compromised solution shows the advantage in compactness of the FC fan over other types.

The following exercise is done comparing the efficiency and performance of different fans, without accounting for the efficiency of the drive system, which shall be discussed in a later chapter.

For a duty point of

Q 7700 m3/h = 2.14 m3/s

Pt 500 Pa (Total pressure)

Installation type: B (ducted outlet)

the optimal selections, without constraints, would be:

Type	Model	Efficiency	LwiA Noise	Mech. Power input	
BC+H	RDA 450 R	78.3% EtaTa	79 dBwA	1366 Wa	(100% or best Wa input)
FC	AT 15-15 S	69.4 EtaTa	84 dBwA	1540 Wa	(113% of the best Wa input)
PF (=BC noH)	RLM E_-5056	77% EtaSr	77 dBwA	1390 Wr	(102% of the best Wa input)

If the overall height of the fan enclosure is restricted to 700 mm, the choice becomes restricted to smaller sizes of both the BC with Housing and the PF fans:

BC+H	RDA 355 R	67.0% EtaTa	89 dBwA	1597 Wa	(117% of the best Wa input)
PF (=BC noH)	RLM E_-4045	66.0% EtaTr	85 dBwA	1610 Wr	(118% of the best Wa input)

Note:

The noise levels and the efficiencies of the PF above, like in the following example, are still not corrected for interference from the nearby walls.

With a useful height of 700 mm and a width of 1200 mm, the estimated efficiency loss is 1.03 3 = 1.09, the power input becomes 1759 Wr (129% of the best mechanical power input) and the static efficiency referred to the power provided to the PF impeller becomes 60.4%, with a similarly significant worsening of the noise levels.

The performance of the PF is estimated in terms of static pressure and static efficiency, because no use can be done of the kinetic energy at discharge, and any kinetic energy in the duct must be built-up at the expense of static pressure at the fan discharge.

The advantage of the PF fan is visible only when the fans are installed in a condition closer to the standardized test installation type A, i.e. with the fans blowing into a plenum, without any kind of ducting, intended to make the best use of the available kinetic energy at the fan discharge.

If we try and specify a similar duty point, but specify a static pressure in installation type A, e.g. because the application is to pressurize a plenum, we have:

Q 7700 m3/h = 2.14 m3/s

Ps 500 Pa (Static pressure)

Installation type: A (free discharge)

Then the optimal choices would be the following fans:

Type	Model	Efficiency	LwiA Noise	Mech. Power input	
BC+H	RDA 450 R	73.0% EtaSa	76 dBwA	1464 Wa	(105% of the best Wa input)
FC	AT 15-15 S	51.7 EtaTa	85 dBwA	2068 Wa	(149% of the best Wa input)
PF (=BC noH)	RLM E_-5056	77% EtaSr	77 dBwA	1390 Wr	(100% or best W input)

If the overall height of the fan enclosure is restricted to 700 mm, then the choice for BC fans becomes restricted to smaller sizes, like the following

BC+H	RDA 355 R	58.70% EtaTa	89 dBwA	1823 Wa	(131% of the best Wa input)
PF (=BC noH)	RLM E6-4045	66.0% EtaTr	85 dBwA	1610 Wr	(118% of the best Wa input)

The noise values are total sound power level values, A-weighted, which provide only a partial indication of the acceptability of such noise emissions, as they do not provide information of the frequency distribution of such power, and of the consequent effectiveness of the silencers, which are often required to bring the noise level in the ventilated spaces within acceptable levels.

Comparison of noise spectrum between a typical FC fan and BC fans

The acoustical behaviour of the three main types of centrifugal fans resembles what was already seen for their operating efficiency.

If we compare the sound power level (inlet-side) spectrum of the three fans selected for the first example, i.e. a double-inlet FC fan, a double-inlet BC fan with scroll-case and a PF (BC centrifugal fan without housing), plus the two BC fans (BC fan with housing and PF or BC fan without housing) undersized to satisfy the limited height requirement, we have the following numerical values, for the inlet side sound power levels:

	FC fan	BC+Housing	BC No Housing	BC+Housing Undersized	BC No Housing Undersized
	AT 15-15 SC	RDA 500 R	RLM-Ex-5056	RDA 355 R	RLM-Ex-4045
LwiA dB					
AW	82	74	72	88	80
63 Hz	77	76	68	88	72
125 Hz	81	72	74	88	78
250 Hz	79	79	71	83	80
500 Hz	77	71	68	89	74
1 kHz	76	66	66	80	72
2 kHz	76	63	64	77	73
4 kHz	74	59	62	70	72
8 kHz	69	55	59	63	67

The comparison between the FC fan and the other two optimally-sized fans seems to be showing a considerable disadvantage of the FC fan.

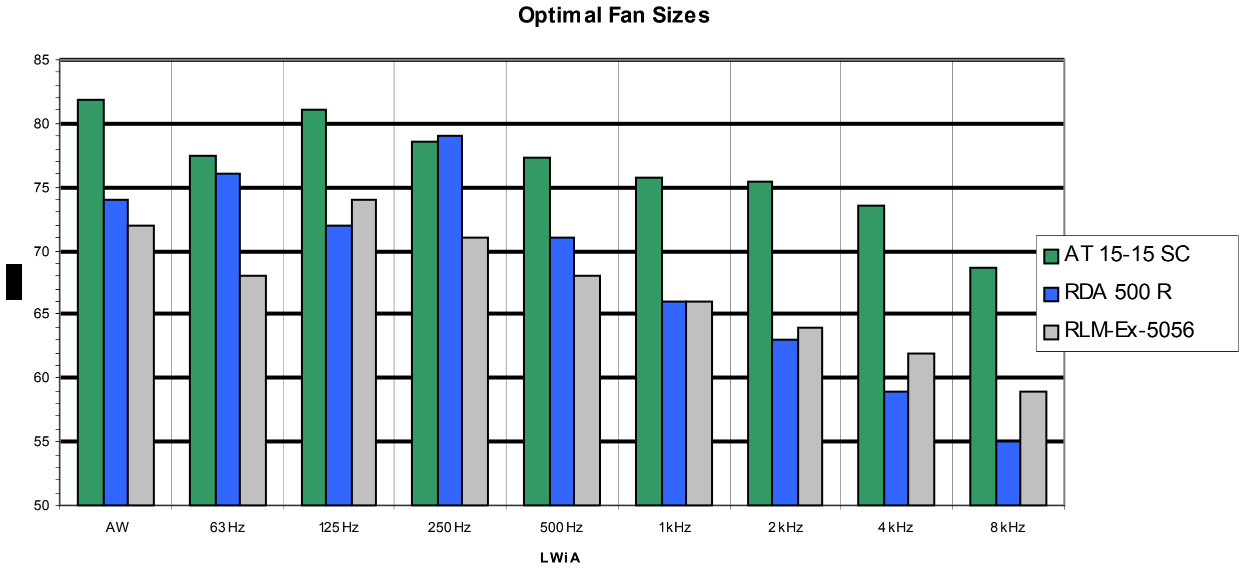


Fig. 10 : Sound spectra of FC, BH+H and PF fans when optimally sized for the same duty

but if we compare the FC fan with the two BC fan models, undersized to fit within a constrained space, we have a striking difference:

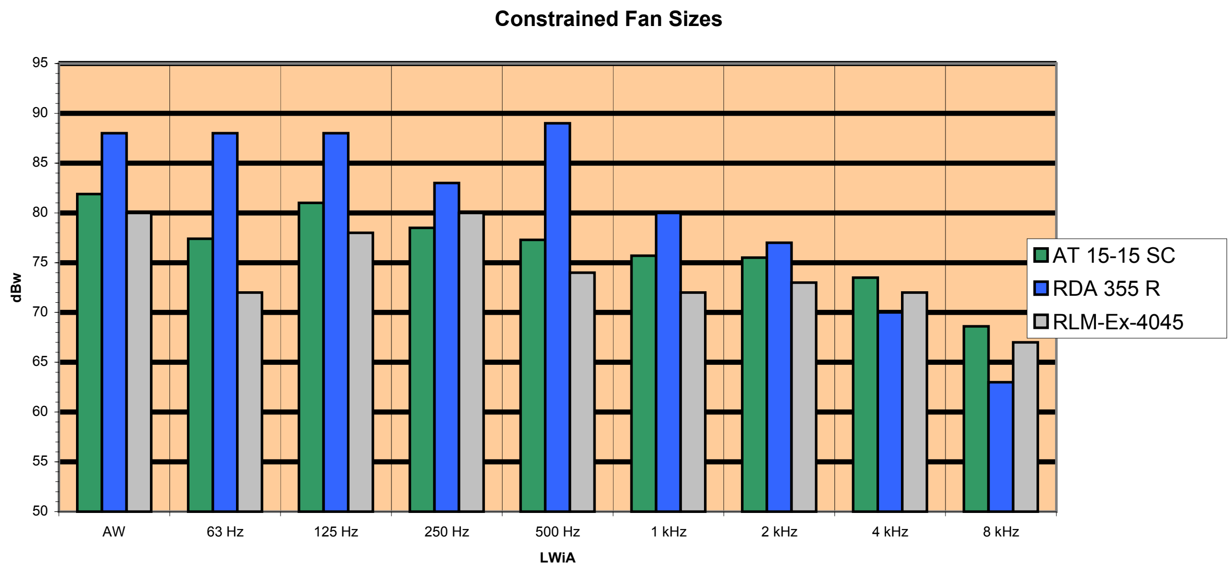


Fig. 11 : Sound spectra of FC, BH+H and PF fans when sized according to space constraints

Here the disadvantage of the BC fan with scroll, when deliberately undersized, is significant, while the undersized PF behaves better, but it must be noted that these sound spectra are still the values referred to the smaller fan operating at the specified duty, but still in laboratory conditions, I.e. with an unobstructed flow. Even without accounting for this additional penalty of the PF, when the space constraints restrict the size choice, the sound spectra of the PF ad FC fans are very close, with an efficiency advantage for the FC fan.

With a significant difference from fans with housing, the performance of PF fans is sensitive to the interference with the nearby walls, but the prediction of the SWL from these fans under such disturbed conditions is not easily achievable with the computational methods available in public catalogues, and should actually be measured. We may reasonably assume that the catalogue noise ratings represent an underestimate of the real values.

The effect of the passive silencer on the fan noise spectrum

In ventilation systems, the noise produced in the ventilated spaces, or in the surroundings of the ventilated building, are normally of the outmost importance, because high noise levels toward the inside can make uncomfortable the internal spaces, and toward the outside be unacceptable to neighbours, or even exceed legally enforced maximum levels for inhabited areas.

For obvious reasons, the fan noise cannot be easily contained by enclosing the fan into a sound-insulated box, because the inlet and the outlet would let the sound pass, as much as the airflow.

To comply with the design or legal noise limits, the system designer can rely upon the natural adsorption of noise as an effect of the viscosity of the air, along the ducts, or may rely on the sound adsorbing liners which may be added to the walls of the same ducts. Most often, when these solutions are not enough, the system must include passive silencers along the air path, i.e devices which attenuate the sound power transmitted along the device, introducing at the same time as low a pressure loss as possible.

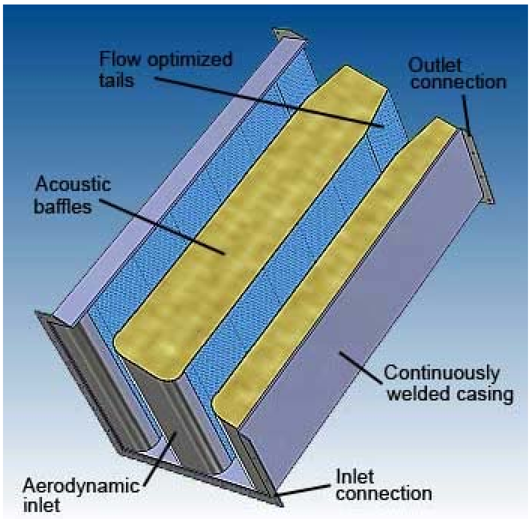


Fig. 12 : Example of passive acoustic silencer construction

As good as a silencer may be, its effectiveness is not evenly distributed across the acoustic frequency range, but is rather concentrated on the medium frequencies (500 Hz to 2 kHz) and introduces an additional pressure loss, in the system, with the result of increasing the pressure demand from the fan, and the power consumption from the system. It is also not widely known, but silencer do not only reduce noise: due to the air friction on their splitter surfaces, they generate noise as well, at the expense of the airflow energy, and this is actually limiting the pratical amount of noise reduction which can be practically achieved.

If we just try to calculate what can be the effect of adding a reasonably matched silencer with our reference fans we have these results:

	Insertion loss	Flow noise	FC fan	BC+Housing	BC No Housing	BC+Housing Undersized	BC No Housing Undersized
			Silenced	Silenced	Silenced	Silenced	Silenced
LwiA dB							
AW			55	54	53	58	54
63 Hz	14	57	70	65	64	75	69
125 Hz	23	55	61	60	57	67	59
250 Hz	34	43	51	47	48	56	50
500 Hz	45	42	45	45	43	46	45
1 kHz	50	48	49	48	48	51	49
2 kHz	51	48	49	48	48	49	48
4 kHz	46	42	44	42	43	44	43
8 kHz	30	32	46	37	38	43	44

If a silencer is added along the system, the difference of noise propagating upstream of the fan is significantly reduced, when the fan sizes can be optimised,

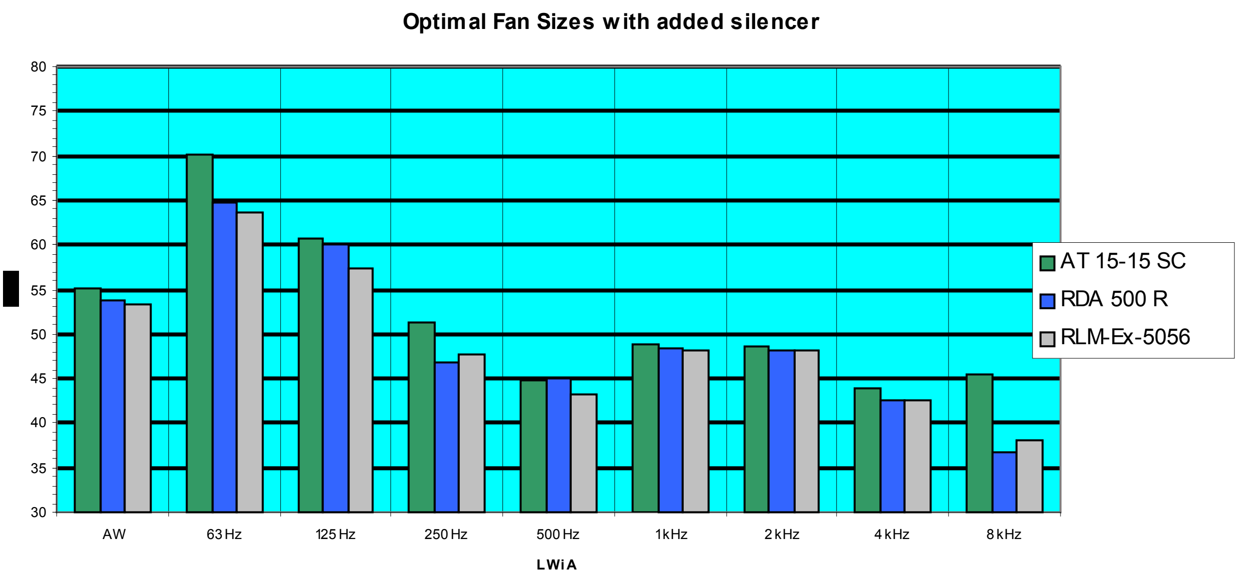


Fig. 13 : Sound spectra of FC, BH+H and PF fans optimally sized, across silencer

The difference is further reduced if the BC fans have to be undersized:

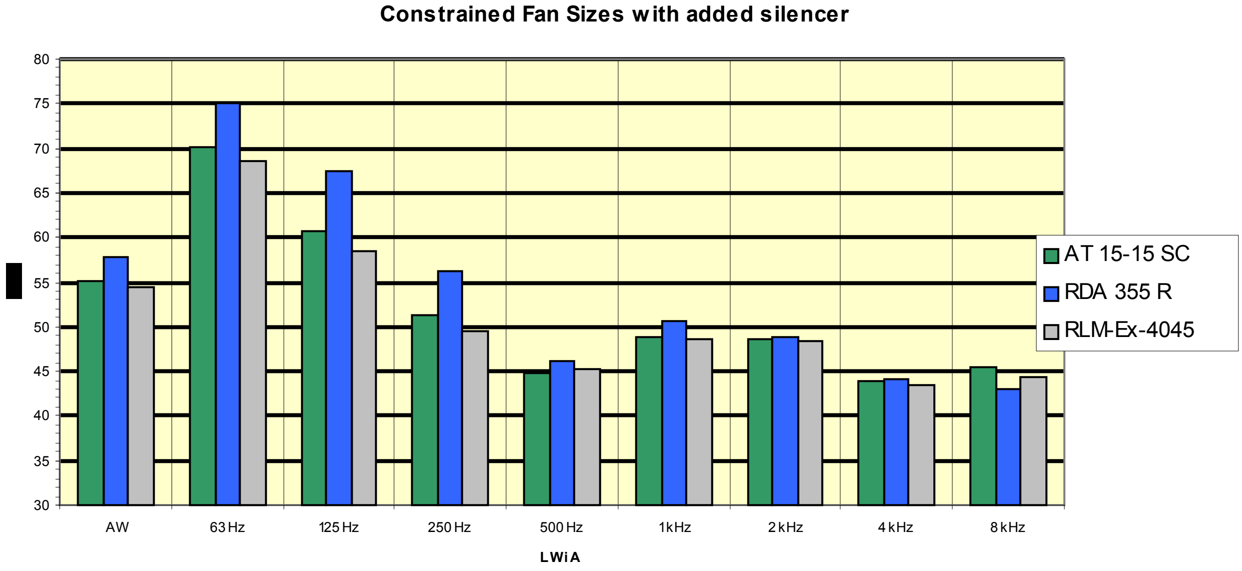


Fig. 14 : Sound spectra of FC, BH+H and PF fans optimally sized, across silencer

In this case, the difference between the total, A-weighted sound power level, propagating upstream from the FC fan is within 1 dB from that of the PF fan, which is generally credited of the best acoustic performance, but the efficiency advantage shown by the PF in an ideal installation has been replaced by a considerable advantage for the FC fan.

Drive system arrangement

All the considerations done so far, regarded just the aerodynamic efficiency of the fan (also called "mechanical efficiency", in US technical literature), i.e. the ratio between the useful aerodynamic power output, and the mechanical power delivered to the impeller, or to the shaft supporting the impeller.

The electrically-driven fan, as anticipated, is the result of some degree of integration, between a driven machine, and a drive system including at least an electric motor, and possibly a mechanical transmission and/or an electronic control unit (VSD or controller for PM motors or switched-reluctance motors).

A significant part of the variability of the efficiency, achievable with an electrically-driven fan, is due to the variability of efficiency of the drive system.

As will be clear from the numerical values, for fans having a power input below 5kW, the choice of motor technology and the motor shape and installation are at least as important, on the overall efficiency of the electrically-driven fan, as the choice and design of the aerodynamic parts.

The choice of the motor, together with its mechanical arrangement and layout, has an additional effect through the aerodynamic interference coefficient C_I , representing the effect of the aerodynamic impact of the motor on the efficiency of the assembly. This clearly has the major effect whenever the motor is directly coupled to the impeller, in a "direct drive" arrangement.

Fig. 15: Examples of Centrifugal fan with motor outside the airstream

The value of the Coefficient of Interference may be 1.0, when the motor is placed outside of the airstream, and has consequently no effect at all on the aerodynamic fan performance, but may reach values significantly below unity, when the assembly is far from being aerodynamically optimised, and the motor blocks the airflow.

Fig. 16: Examples of Centrifugal fan with motor inside the airstream and high Coefficient of Interference

This is typically the result of an arrangement conceived to use motors which may be more easily available on the market, but which were not specifically conceived for use in fans, resulting in a shape which is not ideal for this application.

Otherwise, motor may be specifically conceived for use in fans, looking not just at their electromechanical performance, but also at their shape and position of installation inside the fan structure.

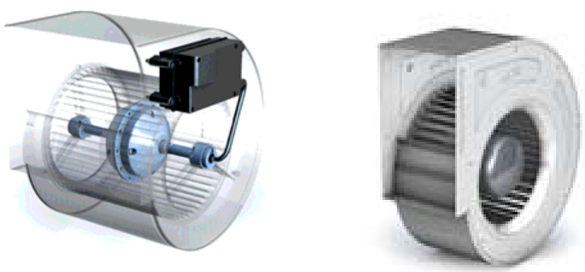


Fig. 17: Examples of Centrifugal fans with motor inside the airstream and very low Coefficient of Interference

Purpose-built motors for use in fans have the drawback of offering little or no flexibility when developing special-purpose fans or just fans having their performance adjusted to specific applications, like most high-power ventilation systems.

Developing and qualifying special-purpose fan motors for “speciality” applications, like ATEX or Smoke Ventilation (PSHV) would also be prohibitively expensive, given the relatively small size of these fan market niches.

With such products, the use of standardised motors is the only practically available option, and the only choice left is between designing the impeller to achieve the specified duty at the available motor speed, adjusting the motor speed to that needed by the fan to achieve the specified duty, by using a VSD, or fitting a mechanical transmission between the motor and the fan shafts, to de-couple the motor and the fan shaft speeds.

Fig. 18: Examples of modular assembly of Centrifugal fans with standard motor

For direct-driven single-inlet (SWSI, meaning: single-width, single-inlet) centrifugal fans, the motor may be placed outside the scroll, on the closed side of the scroll-shaped housing. This has the advantage of avoiding any obstruction to the airflow entering the fan, as well as that of allowing separation of the fan main airflow from the motor cooling flow, thus avoiding contamination, corrosion or abrasion of the motor by any substance which may dispersed into the main gas flow through the fan. On the other side, this arrangement increases the overall length of the fan arrangement, even more when, for any reason, the impeller is supported on an auxiliary shaft, itself supported on dedicated bearings, and connected with a coupling to the driving motor. This arrangement is more common in centrifugal fans for industrial process applications, where simplifies maintenance and may also preserve the motor from high temperatures if the fan handles a hot-gas flow.

On single-inlet fans, the solution of fitting a motor having a conventional, internal-rotor motor, supported by brackets at the centre of the inlet, and supporting the impeller on the motor shaft, is progressively disappearing, as this arrangement may help to reduce the overall length of the fan, but is leading to an efficiency reduction by a C_I factor which may be as low as 0.65.

A better solution is the use of an external-rotor motor, so far most frequently of the AC type, placed in the centre of the impeller disk, and with the motor rotor acting as an enlarged hub, and with the stator supported from the closed side of the scroll case. This solution may improve the C_I factor to values of 0.90-0.95 for backward curved impellers, and 0.95-0.98 for forward-curved impellers. The advantage is due to the fact that the motor body is placed in a region of relatively low airspeed, but the interference factor is still significantly depending on the shape and size (diameter and length) of the motor, compared with the space available to the airflow inside the impeller.

For double-inlet fans (DWDI meaning double-width, double-inlet), the options are similar.

The motor may be placed completely outside the scroll, with the impeller supported on a long shaft. This is generally done with FC impeller, because the low operating speed and low mechanical stress help in keeping the operating speed below the critical speed of the rotor.

A more common solution is to have an internal-rotor motor placed at the centre of one of the inlets, supporting the impeller on the shaft. This allows the removal of the dedicated motor cooling impeller but is leading to a significantly low interference coefficient, often ranging, for DWDI FC fans, in the 0.75-0.85 interval, once again depending on the shape and relative size of the motor, and generally getting higher (less interference) when the fan size increases.

By far, the best arrangement for the driving motor of a direct-driven DWDI fan, is to use an external-rotor motor, placed in the geometrical centre of the rotor. The interference factor may be as close to unity as 0.98, but depends on the arrangement chosen to support the motor and withstand the reaction torque. Again, the effect of the motor on the fan efficiency is generally different between FC and BC fans, but little experimental data is available, in publicly available literature, on the case of DWDI BC fans driven with external-rotor motors.

Belt drives are otherwise popular with the more powerful fans, or when the matching of the fan and motor must be done case-by case, depending on the specified duty.

With a proper design of the drive-guards, the belt-driven arrangement, or an in-line arrangement with a coupling, have no aerodynamical interference at all on the fan aerodynamic efficiency, but the mechanical transmission introduces an additional loss of power along the chain.

In principle, the use of belts could be replaced with an in-line coupling, combined with the use of a VSD, being used not just to reduce the fan duty according to demand, but also to achieve the speed required at nominal duty. This is increasingly done for direct-driven fans using standard motors, like industrial-process fans, but is less advantageous when continuous speed adjustment is not required. In this case, the efficiency loss due to the insertion of an otherwise unnecessary VSD, and due to the operation of the motor at other than design frequency, might not always be compensated by the removal of the energy loss in the belts.

Drive system technology

As far as the electromechanical technology chosen for the drive motor, fans are built using a number of widely different motor types and drive-system arrangement technologies.

The electrical motors alone, excluding all the non-electrical types of prime mover, may be:

I. Standardized according to IEC 60034 standards (or NEMA standards)

II. Purpose-built, and these may be have a mechanical arrangement as

  a. Internal-Rotor motors

  b. External-Rotor motors

They may be designed to be:

1. Powered from a Single-phase AC network
2. Powered from a DC network
3. Powered from a Three-phase AC network

Their main technology may be:

A. High-slip, voltage controllable AC motors

B. Low-slip, high efficiency AC motors

C. Permanent-magnet motors (PMMs, EC motors etc.)

D. Other less-common types of high efficiency motors (switched-reluctance motors etc.)

Technologies B and now also C are available in a IEC standard mechanical layout, while A is more common of the purpose-built motors, while D is still in the early stages of pre-production.

The increasing availability of power-control electronic devices is making the picture more complex, as they allow (at some cost) a degree of disconnection between the electrical design of the motor, and the kind of power-supply network available on site: e.g. three phase-motors can now be driven by an inverter taking power from a single-phase network.

Driving motors: "Standard motors"

There is a significant difference in motor technology, between smaller fans, series-produced and mostly intended for HVAC&R applications, and those fans which are built-to-order, or at least combined with a motor and mechanical transmission to best suit the application.

The fans of this last group include the larger-sized fans, or highly-specialised fan designs, which are actually built on small batches or single examples, but may include fans which are actually made in series, but where the final combination of the series-produced fan with the most appropriately sized motor and transmission is done case-by-case, in accordance with each application requirement.

The manufacturers of these fans cannot take advantage of large production runs to adopt highly customised motors. For these reasons, such manufacturers have to rely on the large range of commercially available "standard motors" i.e. motors built to the modular standard set up in the IEC 60034 series of standards.

The wide extension of the "standard range" is coming from the combination of different frame shapes and mechanical details, with different values for pole number and nominal power/size, and even of the number of phases, at the lower end of the power scale, where single-phase motors can make sense.

These motors are built according to a modular concept, leading to a theoretically huge number of design variants, but their general design is essentially intended to provide power to industrial machines and mechanical equipment. Showing some difference from the roughly-equivalent American NEMA standard, European IEC-standard motors are near invariably built with closed water-proof and dust-proof enclosures, and are generally cooled with their own integral but external, co-axial auxiliary fan. Other designs, like open-frame or independently cooled motors are included in the standardised modular system, but are seldom or never built, and hardly available, unless for very large orders.

Notwithstanding the practical restrictions, of the commercially available motors, to a relatively small sub-range, compared with what is theoretically available in accordance to the IEC standard, the standardization process makes these motors commercially available for small orders or even individual-piece orders.

Standardised motors are thus the common choice to drive the larger fans, including custom-built bespoke fans, and also the more frequent industrial-process fans, still built to order, but out of wide, modular fan ranges, and also to drive those series-produced fans where the drive system is chosen according to the specified duty for each installation.

Besides some centrifugal fans, axial fans are another fan type where the modularity of the product range allows a high degree of fan optimisation for any arbitrarily-chosen duty point, but leads to a wide range of different motor specifications. The use of standard motors is thus the only practical choice to cope with the wildly different motor specifications, combined with the small size of the individual fan production batches.

An advantage of this solution, beyond the flexibility is that different suppliers can produce motors to equivalent dimensional standards, and with similar performance specifications, providing a degree of competition.

The odd side of the deal is made up of a number of shortcomings.

First of all, only the most popular combinations of power, frame shape and pole-number, are actually mass-produced by the major manufacturers (usually outside Europe). Of all the other combinations, some less popular mechanical shapes of the common power levels and pole numbers can be built by warehouse-conversion of some mass-produced motors (e.g. converting B3 to B5 shapes by removing feet and fitting mounting flanges) but the less popular sizes and speeds, or shapes, have to be built-to order, still using standardised basic components, but reverting to a lot of hand-work. As a consequence, motor types which are not off-the-shelf may be slow to procure, and rather expensive.

The high cost is partly contributed by the fact that standard motors are built also for mechanical applications which may require motors having a considerably sturdier structure than is needed in many fan applications, and are often over-engineered for such use.

It must be noted that standard motors are not designed with aerodynamics in mind, beyond that of the cooling air-path, and their use in fans is straightforward only when they are placed outside the main airstream, like in single-inlet fans with the motor opposite to the inlet (Arrangement 5 or 7, according to ISO 13349), or belt-driven double-inlet fans.

When the motors are placed inside the airstream, (and this is done sometimes in centrifugal fans, as well as in many axial fans), they can easily influence the aerodynamic efficiency of the fan (effectively reducing the interference factor C_I) e.g. with the bulk of their physical size, with the interference produced by the cooling fan flow, or with that of the connection box, if this cannot be replaced with a less intrusive arrangement (e.g. in the case of ATEX motors, which cannot be freely altered, without compromising their explosion-safety certification).

The main problem with these standard motors, in relation to the European legislation on fan efficiency, concerns precisely the efficiency of such motors. A specific regulation (Reg. 640/09/EC) set minimum efficiency levels for three-phase motors above 0.75 kW, and a revision of such regulation may extend such requirements to single phase motors and to smaller motors as well.

The problem is that these motors are rather easily available only with efficiency levels which are barely exceeding the minimum mandatory levels. Even if motor technology is available to produce motors, with standardised shapes, dimensions and power-levels, but with efficiency levels significantly exceeding the current IE3 class, their availability to the general market is extremely limited.

Fan manufacturers are also fully aware that single-phase motors (for physical reasons), and three phase motors below the 0.75 kW level (for commercial reasons), still have efficiency levels which are still considerably below the mandatory minimum for 0.75 kW 3-Phase motors.

This is one of the reasons why manufacturers of the smaller-sized industrial-process centrifugal fans, and even more, manufacturers of axial fans, are worried by the shape and slope of the minimum efficiency curves of the existing and of the new proposed fan regulation, and of their mismatching with the minimum efficiency required by EU legislation for smaller-sized motors. The current curves are already assuming that IE4+ motors, in the power range below a boundary set somewhere in between 5 and 10 kW, may be available and are a viable solution to drive such fans.

This power range is where the spread in efficiency, between the minimum mandatory level, and the best available technology on the market (PM motors or switched-reluctance motors), is wider.

Unfortunately, those firms which have to work using standard motors, for flexibility reasons, are still now having serious problems procuring “better than market standard” motors.

For a few years, standardised-frame motors with permanent-magnet design, providing very high efficiency (IE4 or more) have been theoretically available, but they have been, so far, practically available only for special orders and large production runs.

When these motors are available, they have been PM motors, meaning that they can never be powered directly from the mains, but always require special drivers, providing also speed control functionalities (like a high-end inverter for AC motors), and such drivers are still commercially available, at least in small quantities, only from a limited number of players, rather obviously at premium prices.

It must be noted that the increase in cost to adopt such motor technology is not just due to the higher cost of the PM motor and its dedicated driver, compared with that of a more conventional AC motor and its inverter, but also to the fact that an electronic driver is needed also for applications where speed control with an inverter was not required or justified by the application itself.

The gloomy picture concerning “premium efficiency” motors may improve in the near future, under the pressure of a revised EU motor regulation, with an increasing number of manufacturers beginning to offer “standard-frame, permanent-magnet motors” (SF-PM), and suitable drivers, with competition beginning to improve availability and prices of both motor and driving electronics.

This is far from being certain, anyway: It’s worth noting that the revised EU motor regulation, together with the IEC 60034-30-1 standard, tentatively set the new IE4 efficiency class at such level that it can still be met by some improved-technology AC motors, thus discouraging, to some extent, those manufacturers which were attempting to use technologies significantly overshooting the IE4 level. On the other side, fan manufacturers, to achieve the current and possibly the future requirements for small fans, in applications where the aerodynamic efficiency cannot reach benchmark levels, may discover that motors of the IE4 efficiency level are not enough, and the switch to PM motors and IE5 or more may be unavoidable.

Driving motors: “Custom-built motors”

While manufacturers of more or less “custom made” fans are struggling with the availability of standard motors, the larger manufacturers of series-produced fans, mostly for HVAC&R applications, are making fans overlap with special-purpose designs, in the power range below 5-10 kW, but using purpose-built motors, often internally produced by the same fan manufacturers, and a few of them also introduced, since a few years, high-efficiency motors using permanent-magnet technology which, if properly built, at 0.75 kW (output) or below, may already exceed by a significant margin the efficiency limit proposed for the IE5 class.

This is by far the major technological breakthrough available in technology, leading to the improvement of electrically-driven fans of low power.

Above 10 kW, on the contrary, the use of such motors can provide only a marginal improvement over what can be achieved with conventional AC motors of the improved IEC efficiency classes.

Fans built with these high-efficiency motors were already part of the panel of products used by the Fraunhofer Institute in its preliminary research, which supported the definition of the efficiency limits contained in the existing Reg. 327/11/EC.

As the analysis, which led to these limits, did distinguish between fans driven by AC motors, and fans driven by PM-motors or any other high-efficiency technology, the minimum-efficiency curves for axial fans and for different types of centrifugal fans where drawn relatively high in their final leftmost part, where the PM-driven fans, already a few years ago, where considerably raising the efficiency of the population of fans, both centrifugal and axial, with a nominal power input of few hundred watts.

This is already giving troubles, with Tier 2 of Reg. 327/11/EC, to those manufacturers who have to rely on standard AC motors for their fans, because custom-made motors are either unavailable or technically

unsuitable (e.g. cannot withstand high temperature, even for a short emergency run), which cannot achieve the specified efficiency levels set for Tier 2 or further targets.

Custom-built motors are still nowadays mainly of the high-slip type, a kind of induction motor where the downward slope of the torque vs. speed curve is kept low, by appropriate design of the squirrel-cage rotor. These motors have two interesting characteristics which, before the availability of power-control electronics made them particularly interesting in combination with the FC fans.

The first one is that when these motors are subject to a “high-load” meaning that the driven machine applies a relatively steep input-torque-vs.-speed curve, crossing the curve of the torque provided by the motor close to its maximum peak, the effect is that the running speed of the motor and of the driven load drops significantly.

In a “low-slip” motor, the motor speed may be allowed to drop by only a few percent, before the overload current and the consequent overheating become too high to be safe. The high-slip motor is designed to loose as much as 30 % of the zero-load speed, without being subject to unacceptable current and overheating.

This characteristic was interesting in combination with the forward-curved fans, which have a torque input, at impeller constant speed, steeply increasing with the volume flow rate, with the result that, without a speed change, the torque input at free delivery may be 3.5-4.5 times as much as the torque input at best efficiency.

Without any speed adjustment, this would mean that an electrically-driven FC fan, which must be able to be run at free-delivery without danger, should be driven by a motor capable of a maximum torque or power output roughly four times the power output which is necessary to drive the fan at its B.E.P. . Most low-slip motors would have to be oversized, to cope with such diverse loading conditions in a safe way, and would not provide their best efficiency when the impeller efficiency is at its best.

Some fans are driven with motors which are properly sized for the B.E.P. condition, but the fan cannot safely operate at free-delivery: the maximum volume flow rate, and consequently the minimum pressure as well as the maximum input current and power must be limited, by acting on the air system and / or by switching-off the fan when the duty point becomes unsafe.

Backward-curved fans do not suffer from the same problem, because they often have their maximum input torque, at fixed speed, at B.E.P. , or anyway the maximum torque is only marginally higher than that at B.E.P., so there is no serious risk of overloading the motor, by changing the system pressure-loss line, once the motor was correctly sized for fan operation at B.E.P. and at the specified speed.

The high-slip motors provide a simple and reliable solution to solve the problem: they simply slow down when they are overloaded. By loosing as much as 30% of their design speed they slow down the impeller. Under cinematic similarity conditions, the power input to the impeller is proportional to the third power of the speed, and so by reducing the impeller speed, say, to 70% of the speed of the impeller at design duty, at free delivery the power input to the impeller may drop from 3.5-4.5 times the power at best efficiency, down to 1.2-1.6 times. This amount of overload can easily be accepted by the motor, without dangerously overheating.

Another advantage of the high-slip motor, is that it can be slowed down, when connected with a load steeply increasing with speed, like a fan, just by weakening its magnetic field by reducing the supply voltage. Low-slop motor can accept a voltage reduction only within a limited range, its torque output is not very much sensitive to the input voltage, and the current changes nearly inversely to the voltage, increasing when the voltage drops. If the supply voltage to most low-slip motors is reduced by more than 10% the motor is likely to become overheating in a dangerous way.

High-slip motors can accept voltage reductions down to 10 % of the nominal voltage, restricting the maximum current and heat generation within acceptable limits, while at the same time their torque vs speed

curves flatten down. In combination with a fan, the effect of the change of the motor output-torque curve is that of slowing down the fan.

High-slip motors are thus suitable for speed adjustment by changing the input voltage to the motor, using a transformer, or, since many years, using cheap “triac” phase-cutting controllers, which provide continuous adjustment of the r.m.s. voltage provided to the motor, and thus continuous speed adjustment at the turn of a knob. Transformers (or autotransformers) are also still used to control the speed of high-slip motors, as an alternative to phase-cutting devices: they provide only stepped changes, but do not distort the waveform of the voltage and current provided to the motor, and so avoid exciting mechanical vibrations which may generate unwanted noise in acoustically-sensitive applications.

Another way to produce the magnetic field weakening required to slow-down a fan driven by a high-slip motor is to change the inductance of the motor, by changing the number of loops in the stator winding, with the insertion of additional loops in series to the winding. Speed is then selected by choosing the number of loops connected to the power supply line, with a simple selector switch. This is by far the cheapest and most simple solution to provide speed selection for small forward-curved fans, usually in the power range between 10W and 1 kW, and is still the most reliable and silent in operation.

Multiple-entry windings, intended for field weakening, are normally used only for single-phase motors, and should not be confused with “multi-speed” windings which are based on the principles of changing the number of poles of the motor.

High-slip motors have a significant drawback: their forgiveness to overloading and undersupplying is achieved by restricting the current in the rotor, by increasing the resistance losses in the rotor loops, with the effect of reducing the efficiency of the motor, compared with that of a motor with a similar design output, but using a low-slip rotor.

The efficiency of a voltage-controllable (not multiple-connection) single-phase motor is typically between 0.8 and 0.9 times the efficiency of a low-slip motor of similar design power output. Furthermore, the multiple-speed winding design is providing an even lower efficiency, because the slot space in the stator must accommodate additional winging loops, forcing the designer to reduce the wire section and increase the current density per unit of wire section, with the result of increasing the power losses due to stator winding resistance, for a given output.

The introduction of advanced electronic control, even in combination with low-power motors, is opening new perspectives. With advanced controls it is possible to use low-slip motors, or even no-slip motors like permanent-magnet motors (PMMs), in combination with forward-curved impellers, and prevent the risk of overloading: these controls can monitor the current and power provided to the motor and simply slow down the motor and impeller if the maximum allowed current is exceeded, thus providing safe operation with motors which cannot provide a passive safety system like the high-slip types.

Internal-rotor motors have been popular in the past, because fan motors could be produced using parts and technology, and even on the same manufacturing lines, which were used to make other mass-produced motors, e.g. motors manufactured for other popular electrical appliances like washing machines or kitchen.-aid devices. Nowadays even the internal-rotor motors have a tendency to be designed specifically for use in fans.

One of the major advantages of the special-purpose motors, is that motors designed to drive fans may be designed with the External-Rotor arrangement, i.e. have the stator inside, and the squirrel-cage or PM rotor built as a shell surrounding the stator, and directly connected to the impeller structure. Some axial fans were even manufactured with the blades die-cast in a single piece together with the rotor, even if nowadays the preference is for blades being mechanically separate, to allow for improved manufacturing flexibility.

Whenever compactness is important, the external-rotor design provides for the best close-coupling of the motor and impeller, with the lower aerodynamic penalty.

The drawback of all these advantages is that the ER motors are hardly adaptable for any other use but for driving fans, and external-rotor motors are near invariably dedicated designs made by the fan manufacturers for their own use. This is creating a problem of accessibility to this really promising technology to all those manufacturers, and particularly SMEs, which do not have a production run large enough to support the design and production of a dedicated motor range.

On the other side, the use of PM motors, combined with “smart” power control and intrinsical safety features, provides the best option to improve the energy efficiency of the small centrifugal fans used in HVAC&R applications, below the 7.5 kW level.

The advantage is significant when the PM motor technology is combined with the external-rotor solution, and the improvement would be even larger if measured in terms of averaged efficiency on a representative range of duty points, for a fan which is actually adjusted during the daily or seasonal cycle.

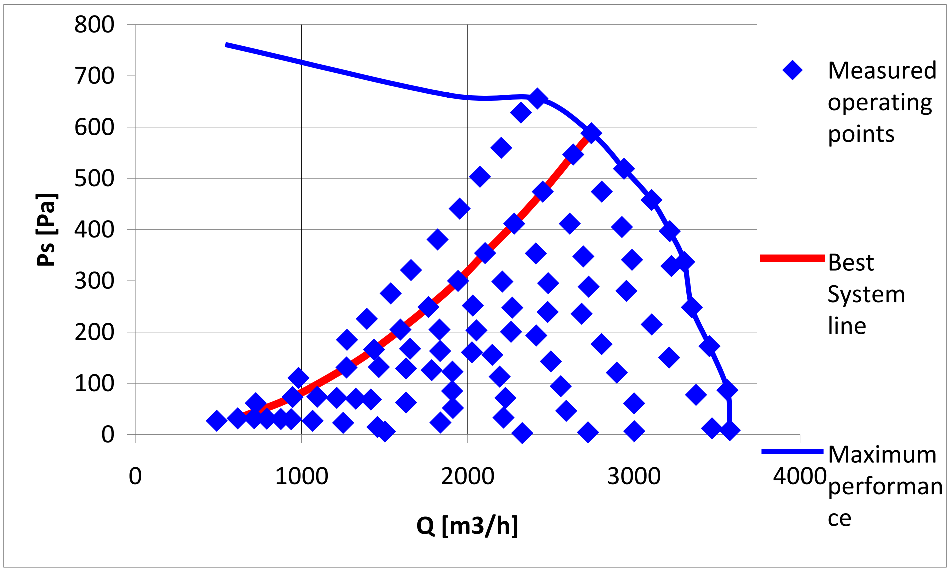


Fig. 19: Example of Forward-curved fan driven by ER-PM motor

The following diagram shows the operating range of a small, speed-controllable forward-curved fan, driven by an external-rotor (ER) PM motor, and the two following diagrams show the curves of efficiency (measured efficiency, without applying the compensation factor Cc), and electrical power consumption of the resulting fan, when the duty point is moving along the “best duty line” or the place of the best efficiency pints at different speed settings.

By comparison, the efficiency and power diagrams show the efficiency and the power input achievable from similarly-sized fans driven by more conventional solutions: ER-AC is an external-rotor AC motor, speed-controlled by voltage reduction; IR-AC is an internal-rotor AC motor, again speed-controlled by voltage reduction, while ER-AC+VSD is an external rotor AC motor, where the speed adjustment is achieved by changing frequency with a VSD.

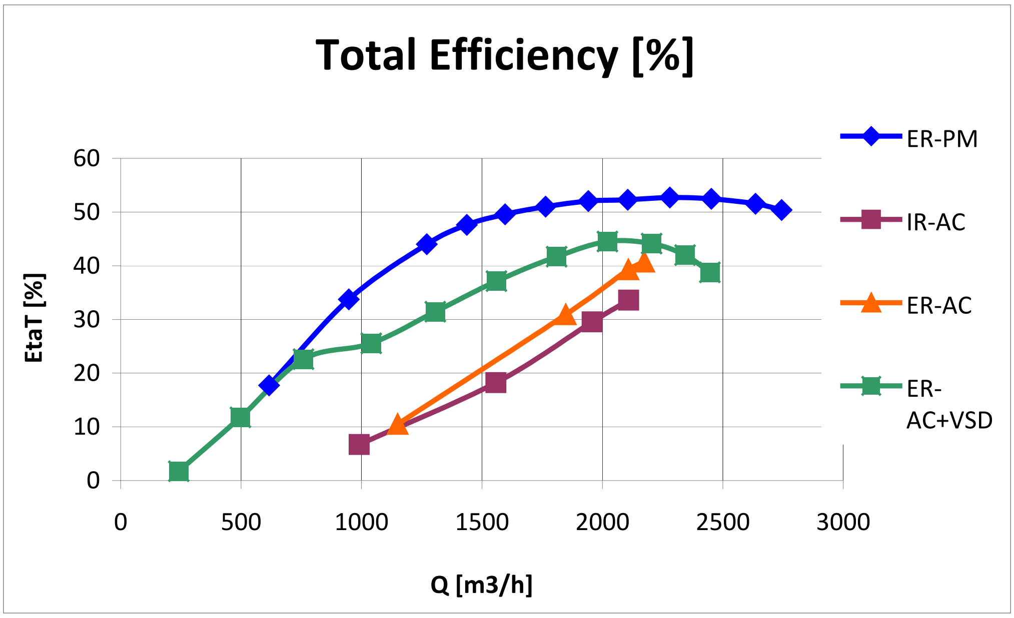


Fig. 20: Example of Forward-curved fan driven by ER-PM motor – Efficiency along the best-duty line

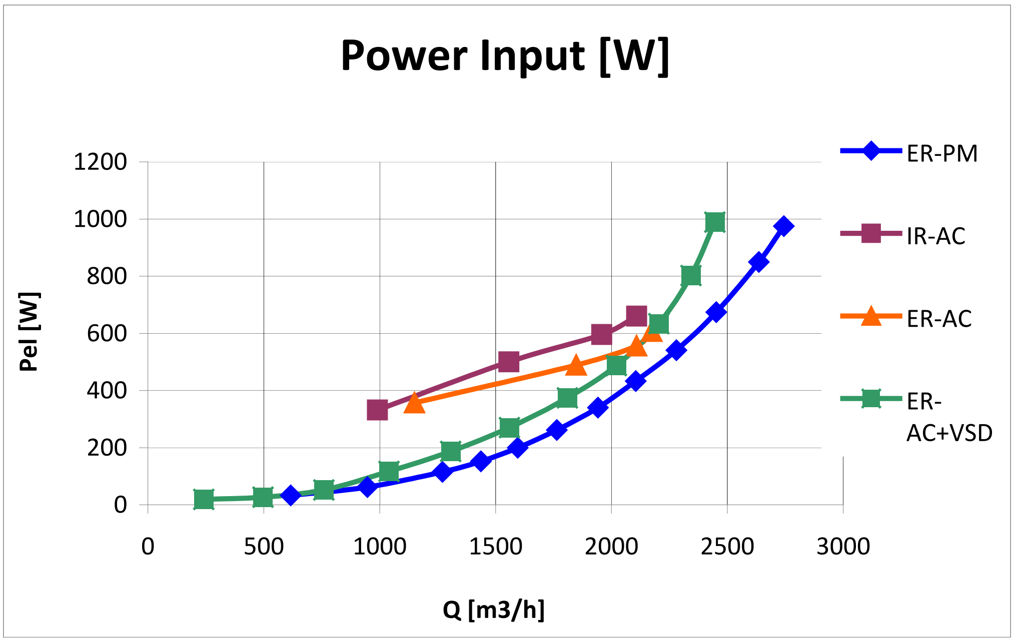


Fig. 21: Example of Forward-curved fan driven by ER-PM motor – Power input along the best-duty line

Fan Control and its impact on averaged operating efficiency

There are essentially four methods to adjust the volume-flow-rate of a fan:

a) Re-circulation or short-circuiting

b) Throttling with passive dampers

c) Adjusting rotational speed,

   1 by changing motor connection and pole number

   2 by changing motor connection and induction level

   3 by changing voltage to the motor

   4 by changing frequency to the motor

   5 by adjusting the transmission ratio

d) Adjusting the geometry of the fan

   6 Adjusting the angle of stator blades

   7 Adjusting the angle of rotor blades

Re-circulation

Re-circulation means opening, at least in part, a damper in a duct which is connecting the fan outlet back to the fan inlet, thus recirculating a part of the generated airflow.

It's a very energy-wasting solution, generally used only as a short-term, fast-response action, to adjust the performance of large fans, with high inertia, to rapid changes in demand, and this method is always combined with another, more energy-efficient but slower solution (most frequently involving speed adjustment) which is used for longer-term control.

Throttling with passive dampers

Throttling means adjusting a damper, located along the ducting system, to increase the system resistance, and thus modify the "system resistance curve" i.e. curve of the pressure needed to blow the airflow along the circuit, as a function of the volume flow rate along the circuit.

Raising the system curve moves the crossing of the system curve with the fan characteristic curve to a new crossing point. The crossing point is where the pressure provided by the fan equals the pressure needed to blow the airflow along the duct circuit, at the same volume flow rate, and represents the duty point under stationary conditions.

If the fan is operated at essentially constant speed, increasing the duct-system resistance reduces the volume flow rate.

This control method was popular in the past, as it required only a simple mechanical device like a damper, inserted anywhere along the ducts.

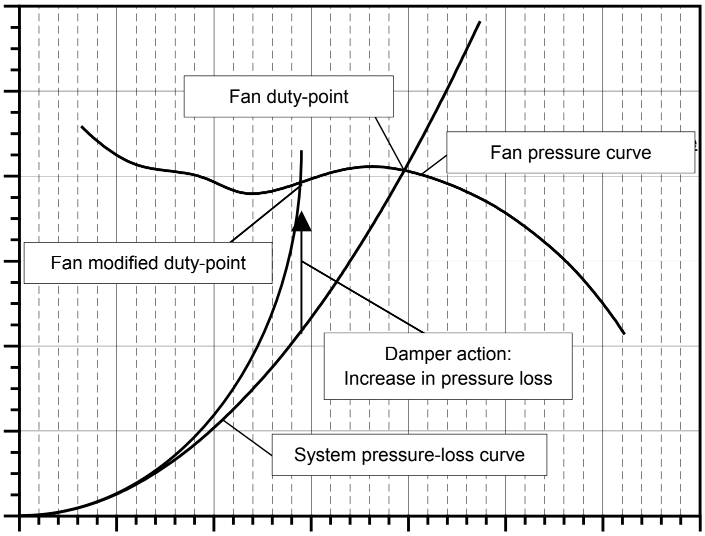


Fig. 14: Effect of damper closure on duty point

It is preferable to use damper control only when combined with a forward-curved fan, rather than with a backward curved one: The upward sloping power consumption curve of the forward-curved fan implies that a reduction of the volume flow-rate, without changing the fan speed, is enough to result in a reduction of the power input to the fan. The same result could not be achieved using damper control with most backward-curved fans, which have a far less-sloping power consumption curve.

See on this subject also the Fan Application Guide, jointly published in UK by the Chartered Institution of Building Services Engineers and by the Fan Manufacturers Association[2].

<u>Fan speed adjustment</u>

Instead of changing the system pressure loss, the other obvious way to change the duty point of a fan connected with a channel system is to change the characteristic curve of the fan representing the pressure provided by the fan with the changing volume flow rate.

The most immediate way to do this is by changing the fan speed. This modified the fan characteristic curve in accordance with the cinematic similarity laws, and moves the crossing of the new fan characteristic curve with the unmodified system curve, to a different flow rate.

The most common use of the speed change was to permanently adjust the performance of a fan, during system commissioning, by changing the pulleys of the belt drive connecting the electrical motor with the driven fan.

Speed adjustment using gearboxes or hydraulic couplings was used, in the past, and remains in use only for very large, high-power industrial fans, especially when driven by non-electric prime movers.

Nowadays the relative availability of inverters has made the speed adjustment by frequency control universally adopted in the control of the fans in central Air Handling Units, where log running hours combines with system operation with a volume flow-rate demand way below the maximum, which is needed only in extreme whether conditions.

[2] See “Fan Application Guide”, available from CIBSE at http://www.cibse.org/

Practically all the supply and extract fans in AHUs delivered in Europe, with power levels most frequently in the 3 to 30 kW bracket, are now provided with speed adjustment using an inverter, controlling an IEC standard AC motor.

The odd thing is that the motors are still normally connected to the shaft of the DWDI fan using a belt drive. This is partly due to low-practice of the designers in choosing motors having their design duty at a supply frequency far different from the nominal 50 Hz. This arrangement is also still preferred because the mechanical assembly using a belt drive makes easier the replacement of the originally-specified with an up-rated or down-rated motor, in the far from rare case when the pressure loss of the ventilation system is so far away from the design value (too much safety margin in the design, or too many last minute changes on site) to recommend an adjustment to the motor size and rating.

The alternative installation with the motor connected in-line to the fan shaft using a coupling requires a high-level of engineering to avoid mechanical problems related with vibrations, and requires more extensive re-working in case of a last-minute change of motor power and frame size.

This trend is slowly changing, thanks to the increasing popularity of plug fans / plenum fans, which are slowly convincing customers that the belt drives are made redundant by the adoption of VSDs, and demonstrating that the removal of the belts is both improving the energy efficiency of the fan, and eliminating a major need for maintenance.

The two main types of high-efficiency motors, which are now appearing on the market, PM motors, and (still in their infancy) switched-reluctance motors, have both the peculiarity that they MUST be operated with a dedicated closed-loop VSD, otherwise they cannot be started, and are unstable in operation.

The main advantage of variable-speed control is that it definitely reduces the power required by the impeller to provide the required volume flow rate and pressure.

On the other side, it is normally assumed that, by changing fan speed, the duty point shall move along a parabolic line of cinematic similarity, and thus the fan performance parameters shall change according to the fan similarity laws.

This is based on the more or less conscious assumption that the pressure loss of the ducting system is also proportional to the volume flow-rate squared, or to the dynamic pressure of the flow. This is true when the pressure loss is generated by jets into an open space, or by surface friction under fully-developed turbulent flow conditions. When these pressure loss sources are the only ones, the pressure-loss curve of the system is actually a square-law curve.

Laminar-flow friction, on the other side, produces a pressure loss proportional to the first power of the airflow speed, so devices where at least a significant part of the pressure loss happens in very narrow channels, or in slow moving flow, have a pressure-loss vs. volume curve which is effectively a second-order polynomial, sometimes approximated with a monomial power curve, having an exponent between 1 and 2.

In the end, transition effects can sometimes generate relatively sudden changes of pressure loss with relatively small changes of volume flow rate. Elbows without turning vanes are well know to create pressure-loss curves showing steps across critical speed thresholds.

Last but not least, hydrostatic pressure may generate a “pressure loss component” effectively independent from the volume flow rate, adding a constant term to the second order polynomial function. Typical cases of ducting system with some hydrostatic component are hot-gas slow in stacks or chimneys, or blowing a gas-flow through a liquid bath.

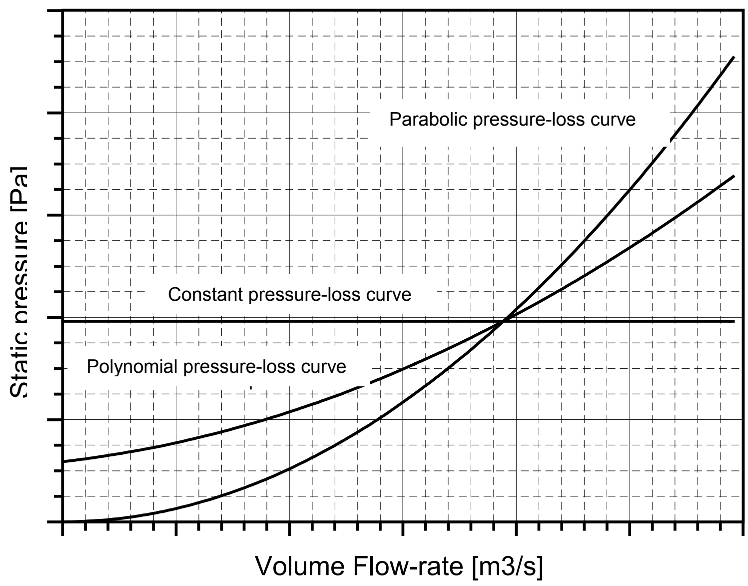


Fig. 15: Different shapes of system pressure-loss curve

In practice, the majority of the systems have a prevailing second-order term and a square-law function may be an acceptable approximation to the real function, at least in a limited air speed range.

As long as the system pressure loss can be considered quadratic, the duty point, at the crossing of the fan characteristic line with the system pressure-loss curve, moves along a parabolic line of cinematic similarity. Then the fan mechanical (or aerodynamic) efficiency is, at least approximately, invariant, the mechanical power input to the impeller shaft is proportional to the third power of the fan speed, and the only problem is the variability of the electro-mechanical efficiency of the drive system, with the change in its speed and power output.

When the actual pressure-loss curve of the system deviates significantly from the simple square-law function, the fan duty point does not move any longer along a cinematic similarity curve.

This implies a number of things:

a) The fan aerodynamic efficiency does not remain nearly constant any longer

b) The fan stability cannot be guaranteed along the .system pressure-loss line.

This second point may be dangerous for the larger and heavier duty fans. By moving to the left of a parabolic line, the duty point may enter into the stall region.

Smaller fans, with low centrifugal stress on the blades, like most direct-driven ventilation fans, may operate in stall for an indefinite time without risk, but the larger and faster ones, including most fans used in AHUs and in industrial process plants, generally do not have so much structural margins to be able to accept the airflow pressure fluctuations of a stall for an indefinite time, without starting to develop fatigue structural damage.

If the fan must be operated along a significantly non-parabolic system line, special aerodynamic solutions must be devised, to restrict the pressure fluctuations generated by the stall to structurally acceptable levels, or the structure of the fa must be suitably reinforced. Sometimes the preferred solution may be to revert to some kind of fan geometry variation.

<u>Fan geometry adjustment</u>

He best known case of fan geometry variation is the change of blade pitch used in some types of axial fans. In centrifugal fans, a change of blade pitch is more complex, because of the structural connections of the blades at the shroud as well as to the impeller disk. For this reason, very few centrifugal fans were ever designed to adopt variable-pitch blades.

By far, the most frequent geometry adjustment in centrifugal fans is the use of stationary “distributor” blades, set radially in the inlet flow, and turned all together around radial axes, to impart an adjustable “swirl” to the airflow entering the impeller. This device is generally known as “Variable Inlet Guide Vanes”. Very roughly described, the effect is vaguely similar to changing the effective speed of the impeller, by turning the entering airflow. Turning the vanes changes the pressure-volume characteristic line of the fan, even without changing its actual speed, and reduces the power consumption, compared with the effect of a simple damper.

Before the availability if VSD, these mechanical devices were common in large BC fans with housing. Nowadays, they have been largely replaced by the VSD, except in those applications where the special shape of the system pressure-loss curve can lead the duty point inside the stall region of the centrifugal fan. A side-effect of the flow-distortion produced by turning the stator vanes, is to move the stall line to the left, when the vanes are partly closed.

For this property, VIGV devices are still in use, mainly as a mean to move the stall line to lower flow rate values and preserve stable operation of relatively large BC centrifugal fans, when reducing simply the fan speed would move the duty point of the resulting fan and system combination in a dangerous stall condition.

As the VIGV devices reduce the fan efficiency, due to the drag of the blades, even more when the bladed are turned from their neutral position, the VIGV devices are now frequently used in combination with VSD, to adjust the volume-flow of the system with the minimum necessary angle or deflection of the blades which is needed to preserve stability outside the natural stable operating range of the fan.