CONSIDERATIONS CONCERNING THE POSSIBLE FAN EFFICIENCY IN REAL LIFE APPLICATIONS WITH LIMITED FAN SIZE

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SUMMARY

The minimization of energy consumption of fans in real life applications with limited fan size is addressed. It is demonstrated where fan size limitations come from, often from installation space, weight, available motor torque or indirectly from the product cost. In combination with the required duty point (volume flow rate and static pressure rise), the maximum possible impeller diameter defines an application parameter δ^*. Depending on the value of δ^*, different fan types are necessary for a minimization of the energy consumption. Fan suppliers can thus cover a big range of δ^* with optimal efficiency only by providing fans with different designs. The considerations are extended specifically for cases with installation space limitation.

INTRODUCTION

The main interest of a fan operator concerning the fan efficiency is the minimization of energy consumption of the fan during its operation in a given environment. The fan has to assure the delivery of a required airflow at the presence of a pressure drop resulting from the air ducting system and the components mounted in it. For the choice of the fan, often constraints with regard to the fan size have to be taken into account. A limitation of the fan size may come from the available installation space, the maximum weight of the fan, the maximum motor torque or indirectly from the product cost. This is demonstrated with some typical application examples. Opposed to the size limitations, the choice of bigger fan diameters is often favorable for low energy consumption or low noise level of the fan. In consequence, it is assumed in this work that the fan size is pre-determined for a fan demand (that is, the biggest possible fan size with respect to installation space, weight, motor torque and product cost is chosen). On the other hand, due to the growing availability of motors with adjustable rotation speed, the prescription of a fan speed is often not essential for a fan demand. In combination, the required duty point (volume flow rate and static pressure rise) and the pre-determined fan diameter, a parameter δ^* can be defined, which is compared to the specific diameter δ known from the Cordier diagram [1, 2]. The achievable efficiency and the optimal fan geometry depend on this parameter δ^*. This is demonstrated by comparing the efficiency curves of different fans designs with different fan types as function of δ^*. For each value of the so defined application parameter δ^*, the optimal fan design out of the different considered fan designs can be determined directly. This gives arise to a strategy for a fan supplier to cover an entire region of values of δ^* with nearly optimal efficiency using different fan designs. In the last section, the considerations are extended to the case that the fan size is restricted due to a typical installation configuration.

FAN SIZE LIMITATION

The energy consumption of fans with prescribed fan impeller diameter in fan applications with prescribed duty point is addressed in this paper. In our experience, the size of a fan is in the majority of fan demands at least limited. Different issues arise as reason for the fan size limitation. The main issues are:

- Limited installation space
- Limited weight
- Limited drive torque
- Limited cost / energy content of a product

At a first glance, in some cases the fan size limitation seems to be a “weak” constraint meaning that one could also choose the fan diameter of the next size, because it still fits in, a somewhat bigger drive could be used, a little more weight could be accepted or more cost content could be invested in order to have a more efficient product. But this argumentation does not fundamentally change the fact, that the fan size is limited, it may just shift the quantitative limitation to somewhat higher values.

Considering the energy consumption, the choice of a smaller fan diameter is often (clearly not always) detrimental if the ideal technology available is chosen. So if the fan solution with lowest energy consumption is searched, a “fan size limitation” can often be interpreted as “fan size prescription”, because then the fan diameter has to be chosen to be “as big as possible”. So in reality, in the majority of fan demands, the fan size is fixed or at least restricted to a small range of sizes by the application itself. This case is addressed in this paper.

In the following subsections, the four different issues of fan size limitations are discussed.

Installation space

It is common that the installation space is limited. Fans are used within higher level devices, for example heat pumps, residential ventilation units, air handling units, chillers, condensers, air duct systems, air chimneys etc. which have themselves size, weight or cost limitations. Then, fans have to be applied within a pre-defined installation space.

The connection between “installation space” and “fan size” is vague. At this point, the question, which fan diameter can be used within which installation space with still acceptable installation losses, arises. So a fan device planner could be tented to choose a relatively big fan diameter for a narrow installation space. The consequence would be, clearly, that this choice causes high losses due to installation effects. Assuming that some fan designs are less sensitive to narrow installation conditions than other ones, the former have a clear advantage due to the possibility, to choose a bigger fan diameter without severe installation losses. This subject is discussed in some more details in the section "RELATIONSHIP OF INSTALLATION SPACE AND THE FAN DIAMETER D" page 9.

Some typical applications:

- Air handling units (AHUs)
  The most common configuration for AHUs is a rectangular air duct where, due to the required pressure rise, a centrifugal or mixed-flow fan is mounted in, see scheme in *Figure 1* left. Downstream the fan, the air-flow is redirected in axial direction. So the impeller outlet should not be too near to the duct wall in order to minimize the installation losses. Typically, centrifugal impeller diameters are not chosen to be bigger than 56 % of the smaller duct width, whereas mixed flow impeller diameters can easily be chosen up to 65 % of the duct width. On the market, more and more fan designs appear optimized for low installation effect enabling to choose relatively big fan diameters. As in other fan applications, a trend can be observed that increased duct cross sections are used in order to reduce the pressure drops in the duct system, the coils and filters by reducing the air velocity. This means then that bigger fans can be used also. However, the size of the "optimal fan" is also increased due to the lower required pressure rise at the same air volume flow.
- Residential ventilation units
  The installation space is typically strongly restricted, especially for low-airflow units. A CAD model of a unit is shown in *Figure 1*, right. The compactness of these units is very important as they are installed inside residential houses taking expensive room volume. Typically, the fans have to be chosen as big as possible in order to get the required air volume flow. Suction side and pressure side installation effects often have to be accepted.
- Heat pumps
  Similarly to residential ventilation units, the outer volume of heat pumps is minimized by the manufacturers. Reasons are the installation volume of the heat pumps, the transportability within buildings, and also the manufacturing cost of the heat pump. As a consequence, fan installation space is also strongly limited.
- Chillers
  For chillers, typically high volume flow rates are required at low pressure losses. Often, axial fans are used, which are as big in diameter as possible. The fan diameter is limited by the extensions of the heat exchangers.
- Ventilation
  The duct cross section of ventilation systems is determining for the possible fan size. When mixed-flow impellers are used as duct-fans, sometimes a local increase of the duct diameter is realized. However this takes costly installation space for the duct system.
- Air exhaust chimneys
  For example in agricultural applications, the diameter of an air duct chimney directly limits the diameter of the axial fan used inside.

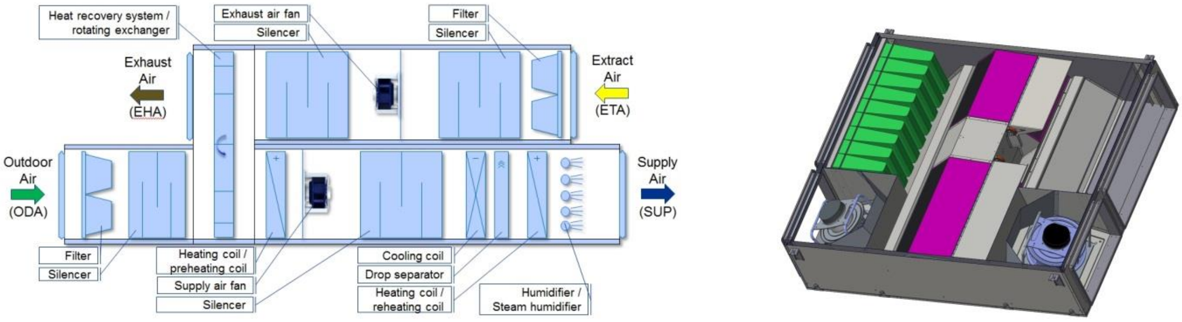

Figure 1: scheme of an air handling unit (left) and model of a residential ventilation unit (right)

Weight limitation

The maximum weight of a fan limits in some cases indirectly the fan diameter. First, the structure, which holds the fan during operation, could support only a limited weight. In combination with not perfectly balanced fans, also the excitation of vibrations could be the limiting factor.

When the fan system itself is moved within a higher-level system, as it is the case for example in trains, ships, cars, trucks, or airplanes, a high fan weight is in itself detrimental to the energy requirement of the drive of the higher-level system, and therefore fans with high weight have an important penalty and the fan size is limited.

The fabrication of a fan with lightweight materials may be a solution in some cases. But the higher-level system itself has a higher weight the more volume it occupies, so it is often constructed in a compact way leading to installation space limitations for the fan.

Motor torque limitation

The mechanical power transferred to a fan impeller is given as $P_{shaft} = 2\pi M \cdot n$. The trend when choosing bigger impeller diameters is that the same power is obtained with higher torque and lower rotation speed. One can counter that in a certain extent using fan designs with high specific speed σ, but the region of σ where efficient fan designs are possible is limited ([1], [2]). High torques at low rotation speeds lead to the necessity of big, costly and heavy drives, which cause cost and weight problems. So in some cases, the fan size is limited by the motor torque.

Cost limitation

In practice, an important factor in the choice of a fan is its production cost and other costs like lifecycle cost, the cost for the installation space, the cost for the drive or the cost for the structure that supports the fan weight. At a first glance, the choice of low-cost fans seems to be detrimental to the superordinate target of reduction of energy consumption or CO2 emissions. But it can also be considered that there is a proven relationship between cost of a product and the energy that was necessary in order to establish the product. This is described for example in [3]. So by choosing fans with a lower cost, the integral energy content of the fan product itself is limited. This is coherent with the general trend, that smaller fans are less expensive. In the end, the cost argument for the fan, drive and the installation space is an important factor limiting the fan size.

BASIC FAN REQUIREMENTS AND NON-DIMENSIONALIZATION

In this section, we define a non-dimensional parameter δ^* containing the duty point and a attributed prescribed fan diameter D. The parameter δ^* is compared to the specific diameter δ known from [1], [2]. The distribution of a large number of fan demands with respect to δ^* is shown.

Duty points

We consider uniquely duty point requirements for one fan (single stage) in the form of pairs of volume flow $\dot{V}$ and static pressure rise Δp_{stat} (total-to-static). In some ducted applications, a total pressure rise $\Delta p_{tot} = \Delta p_{stat} + p_{dyn,duct}$ is demanded, where the dynamic pressure $p_{dyn,duct} = 0.5 \cdot \rho \cdot \left(\dot{V} / A_{duct}\right)^2$ on the downstream side is considered only for the velocity component parallel to the duct axis. But, as the duct cross section is usually fixed, this can be transcribed to a total-to-static pressure rise without any impact on the choice of the most efficient fan, as $p_{dyn,duct}$ is fixed by the given volume flow and duct cross section. Over 200.000 fan demands have been analyzed, and the duty points are plotted in a $\dot{V}$-Δp_{stat} diagram in *Figure 2*, where the pre-determined fan diameter is represented by the dot colors. The fan demands correspond to the applications addressed in the section ***Installation space***.

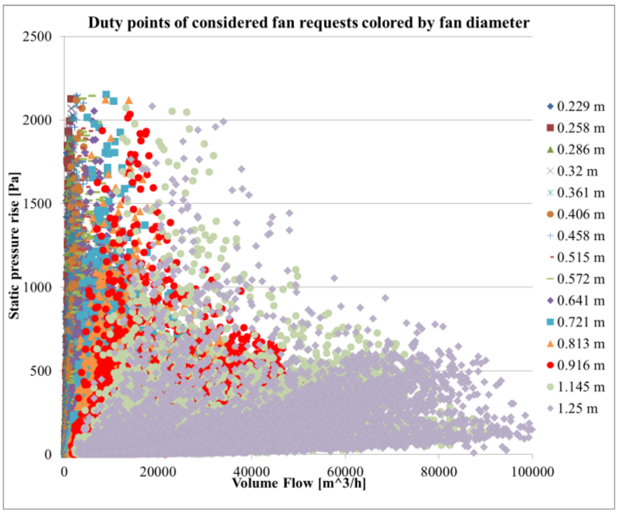

Figure 2: duty points for the analyzed fan demands

Definition of the parameter δ^*

In addition to the duty points, to each fan demand a fixed fan diameter D, also expressed by a reference radius $r_{ref} = \frac{D}{2}$, is attributed. With this, a non-dimensional parameter δ^* is defined:

$$\delta^* = \left(1 + \frac{\psi_{stat}}{\varphi^2}\right)^{0.25} = \left(1 + \frac{\Delta p_{stat} u_{ref}^2 A_{ref}^2}{\rho u_{ref}^2 \left(\dot{V}\right)^2}\right)^{0.25} = \left(1 + \frac{\pi^2 r_{ref}^4 \Delta p_{stat}}{\rho \left(\dot{V}\right)^2}\right)^{0.25} \tag{1}$$

The parameter δ^* has similarity to the specific diameter δ, as described for example in [5]:

$$\delta = \left(\frac{\psi}{\varphi^2}\right)^{0.25} \approx \left(\frac{\psi_{stat} + \varphi^2}{\varphi^2}\right)^{0.25} = \left(1 + \frac{\psi_{stat}}{\varphi^2}\right)^{0.25} = \delta^*. \tag{2}$$

In the definition of δ^*, the total pressure ψ occurring in the definition δ is approximated by $\psi_{stat} + \varphi^2$, meaning that for the dynamic part of the total pressure, only the air velocity component responsible for the air volume flow is taken into account, which is a common way for the estimation of the total pressure. This is technically reasonable because this part of the dynamic pressure is often the technically usable dynamic pressure in an air duct system. So, provided a fan request with given fan diameter, the value of δ^* is determined and the fan has to operate at this value of the application parameter δ^*.

From *Equation (1)*, it follows that $\delta^* = \delta^*\left(\frac{\psi_{stat}}{\varphi^2}\right)$ whereby $\left(\frac{\psi_{stat}}{\varphi^2}\right)$ is constant for a device curve of an air duct system (resistance parabola). So δ^* is constant for each resistance curve and is a non-dimensional, characterizing value of the application (fan device).

From $A_{ref} \propto D^2$ and $u_{ref} \propto D$, it follows that for a fixed duty point, $\delta = \left(\frac{\psi}{\varphi^2}\right)^{0.25} \propto D$, where its denomination "specific diameter" comes from. However, δ^* is, for a fixed duty point, not proportional to the fan diameter D, but $\delta^* \propto \left(1 + D^4\left(\frac{\pi}{4}\right)^2 \Delta p_{stat} / \left(\rho\left(\dot{V}\right)^2\right)\right)^{0.25}$. The explanation why δ^* is not proportional to D as δ is the fact, that the dynamic pressure part $p_{dyn,\overline{vx}} = \varphi^2 \frac{\rho}{2} u_{ref}^2 = \frac{\rho}{2}\left(\frac{\dot{V}}{A_{ref}}\right)^2$ used in the denominator of the definition of δ^* (*Equation(1)*) is, for a fixed duty point, not a constant but depends itself on D, namely $p_{dyn,\overline{vx}} \propto \frac{1}{D^4}$. Anyhow, δ^* is monotonically non-decreasing in D, meaning that for a growing D, always δ^* is growing, too, assumed that p_{stat} and $\dot{V}$ have positive values.

For a duty point with static pressure rise 0 Pa, δ^* takes the value of 1.

Number of fan demands attributed to δ^*-bands

For each of the fan demands presented in *Figure 2*, the value of δ^* was analyzed and attributed to a band of values of δ^* with bandwidth 0.1. The number of fan demands per δ^*-band is shown in *Figure 3*. The analyzed fan demands are distributed in this diagram for values of δ^* from 1.0 to about 4, whereas the number of fan demands with values of δ^* higher than 3.0 is less important. With the objective of covering all these fan demands and supply fans with high efficiency for the corresponding fan application, for the entire region of values of δ^*, fans with good static efficiency have to be available.

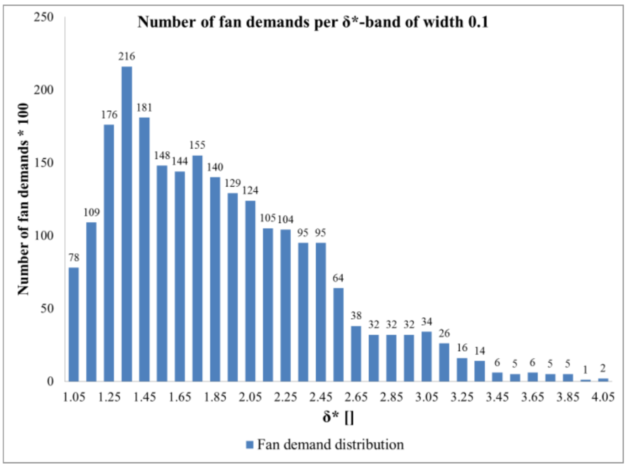

Figure 3: distribution of the analyzed fan demands to bands of δ^* of width 0.1

LOW FAN ENERGY CONSUMPTION FOR THE CONSIDERED RANGE OF δ^*

It is of basic interest of each fan operator, to minimize the fan energy consumption for the device in question with given application parameter δ^*. Thus it is the fan supplier's task to provide, for each value of δ^*, a fan with high efficiency. It is well known from the Cordier diagram [1] [2], *Figure 4*, that depending on the specific diameter δ, different non-dimensional fan designs are the ideal choice, especially in terms of efficiency and noise. For small values of δ, axial fans are the best choice; for higher values of δ, centrifugal fans are necessary. Inspired by Cordier's considerations, the efficiency of different recent fan designs is plotted together in one figure as function of δ^*.

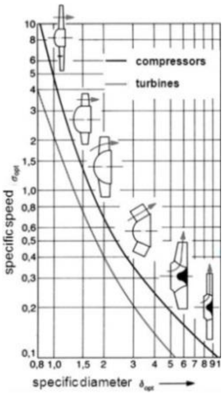

Figure 4: Cordier diagram for fans taken from [4]

Efficiency curves for different fan designs as function of δ^*

In *Figure 5*, the static efficiency curve for different particular fan designs as function of the parameter δ^* is shown. The range of δ^* was chosen to [1..3] as this was the range within which most of the fan requests considered in this work lie.

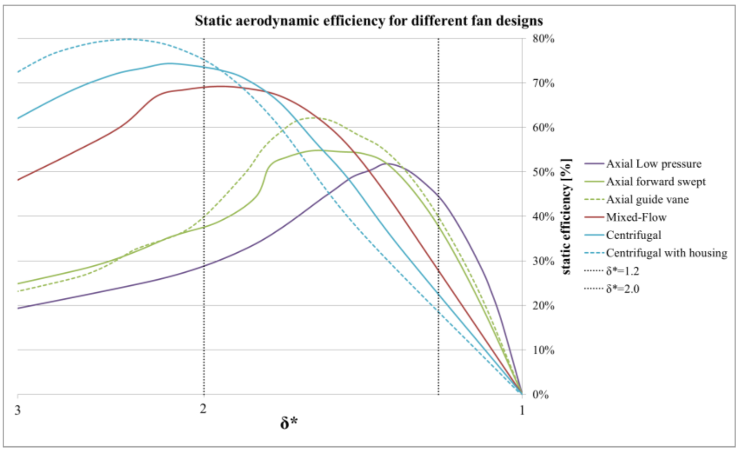

Figure 5: static efficiency curve of different fan designs

In order to reduce as much as possible upscaling- or downscaling effects, the data was generated for fans with the same diameter D = 400 mm and at the same rotation speed n = 2000 rpm. In order to evaluate the efficiency of an impeller design, one can now determine the value of the application parameter δ^* and see directly the resulting maximum value of the static efficiency and the corresponding fan design. In this view, the diagram in *Figure 5* can be seen as link of the fan requirements expressed in terms of δ^* and the fan's response in terms of efficiency. Note that the fan curves on the graph correspond to real, concrete designs and may be, by modification of the design, been optimized for higher efficiencies, shift to the left or the right concerning the value of δ^* at the best efficiency point, or the range of high efficiency could be enlarged in terms of values of δ^*. If, for example, a required fan has, from the application requirement, a value of $\delta^* = 2.0$, one can directly see that the best available efficiency can be obtained by the concrete centrifugal impeller design, with or without housing. If the choice of the housing would be worthwhile for the slight enhancement of efficiency is then a decision to take. On the other hand, none of the axial fans, even not the one with guide vane, could cover this duty point with so high efficiency, as the achievable efficiency with the centrifugal fans is beyond the maximum efficiency of the axial fans.

As another example, let's consider a duty point with a value of $\delta^* = 1.2$. In this case, the best efficiency is obtained using the low pressure axial fan design. It has still better efficiency than the forward-swept design for higher pressure buildup, and for this duty point, the guide vane for the axial fan has only a minor effect. Compared to the centrifugal fan design with a static efficiency of 23 %, the efficiency of the best fan design with 46% is double valued, meaning that the choice of the centrifugal fan would has as consequence the double amount of energy consumption in the application.

It is easy to see that depending on the value of δ^*, different choices of the fan design would be the best solution, at least in terms of static efficiency. A very similar kind of diagram could be shown concerning the specific noise radiation of the fans showing also that depending on the value of δ^*, different designs may be the most silent ones.

The efficiency curves for the fans designs shown in the graph in *Figure 5* suggest that, at least in a range of values for δ^* from 1 to 2.4, higher values of δ^* are favorable for the achievable static efficiency, confirming that in cases where just an upper limit of the fan diameter D is prescribed, the choice of the highest possible value for D is the most promising in terms of static efficiency at least as long as the value of δ^* does not surpass 2.4.

Strategy to cover a region of values of δ^* with a limited number of fan designs

Theoretically, for each given value of δ^* coming from a fan demand with specified duty point and fan diameter, an ideal fan design could be developed with optimal static efficiency and noise level. Anyhow, the plenty of fan demands shown in *Figure 3* indicate, that then fan designs for a huge number of fan requests would be required. It is usually not economic to design for each value of δ^* special dedicated fan geometry, unless a large turnover is associated to this request.

Note at this point that a fan design optimized for a certain value of δ^* does not determine the physical fan geometry, as the fan diameter is defined only in combination with the duty point. So, even if a number of fan requests would correspond to one value or a narrow band of values of δ^*, different scaled fan impellers of different diameters D are necessary. The needed effort in aerodynamic design would be low to realize a fan impeller with modified D for the same value of δ^*, however, the structural design, the drive design, support design, tooling design and so on require a high effort. Also the rotation speed depends on the duty point. If variable speed drives are available as it is the trend for the considered applications, the right rotation speed can be used for the fan operation, as far as this does not exceed limitations concerning the maximum rotation speed of an impeller.

The strategy to cover all, or at least a high portion, of the fan requests with a limited number of different fan designs is it, to introduce some clustering of important parameters which for the very ideal case would be chosen completely arbitrarily. The first clustering concerns the fan diameter D, which is quite common, for example corresponding to a standard range of diameters. The corresponding D for the fan requirement is therefore chosen from a finite number of available diameters. Secondly, a clustering of the range of values of δ^* is proposed which permits to limit the required number of fan designs and to cover the considered range of values of δ^* overall with good static efficiencies. The graph in *Figure 5* already shows a clustering. With four impeller-only fan designs (two axial, a mixed flow and a centrifugal one), the range of values of δ^* is covered with good static efficiencies.

The mixed flow fan design has a lower maximum efficiency compared to the centrifugal fan design. However, if the value of δ^* lies below about 1.8, the efficiency of the mixed flow design is higher in the operation and thus the energy consumption lower. At $\delta^* = 1.5$, the static efficiency of the centrifugal design is at 51 %, where the mixed flow design has 58 %, representing an advantage of about 14%. A similar consideration shows the usefulness of the low pressure axial fan design compared to the forward swept axial fan design for $\delta^* \leq 1.3$

The housing for the centrifugal fan extents the region of very good efficiency somewhat towards higher values of δ^*, and the maximum efficiency is increased. Note also that for lower values of δ^*, the efficiency is in the shown case decreased by the use of the housing.

The presented guide vane design for the forward swept axial fan also extends the region of good efficiency for the axial fan towards higher values of δ^*, and the maximum efficiency is increased. The positive effect of the guide vane is less important for lower values of δ^*. The mixed flow fan has still higher maximum static efficiency and the optimum region is placed at higher values of δ^* compared to the axial fan design with guide vane.

RELATIONSHIP OF INSTALLATION SPACE AND THE FAN DIAMETER D

Installation situation “Axial duct system”

In many cases with constrained fan size, the fan diameter is determined by the installation conditions. The relationship between the installation space available for the fan in an application and the fan diameter D is discussed in this section for a generic installation situation, which exists in many applications, for example air handling units (AHUs). For other installation situations, a different but somewhat analogous argumentation could be worked out.

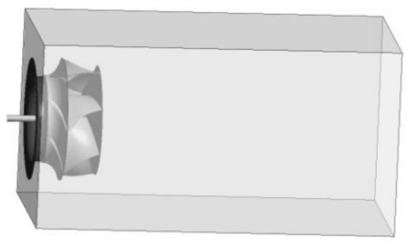

Figure 6: installation situation "axial duct"

In *Figure 6*, this installation situation is illustrated. A fan is installed within a duct system, which has in the shown case a rectangular cross section which can be characterized by the smaller side length s of the rectangle, and the flow direction upstream and downstream the fan is aligned parallel to the fan axis. For centrifugal impellers, this means, that the flow is redirected in axial direction after being ejected in radial direction from the impeller outlet. If then the fan impeller has too big diameter with respect to the duct cross section, installation losses occur.

Relationship between duct cross section and fan diameter

It is evident that for axial fans, a bigger fan diameter D can be chosen with still low installation loss compared to centrifugal fans. Thus, with respect to the given side length s, the picture presented in *Figure 6* , comparing different fan designs, would be distorted as now fans with different diameters would be compared.

With the aim to use the fan diameter as big as possible in order to maximize the static efficiency, we use the biggest possible fan diameter D, which ensures for this configuration an installation loss of still not more than 2 % in static efficiency. The biggest possible ratio D/s depends on the fan design and can be determined, for a given fan design, by experiments or numerical simulations.

In particular, s/D could be chosen to 1.25 for the axial fans (considering the outer diameter of an inlet cone). For the mixed flow fan, $s/D=1.6$, for the centrifugal fan, $s/D=1.8$ and for the centrifugal fan with housing, $s/D=2.1$

Modification of δ^* and the efficiency graph considering the installation situation

For the computation of δ^* (*Equation (1)*), the fan size constraint was given in form of the fan diameter D. Considering the installation situation outlined in *Figure 6*, we define now for this installation situation a modified parameter

$$\delta^*_{install} = \left(1 + \pi^2 \left(\frac{s}{2} \right)^4 \Delta p_{stat} \Big/ \left(\rho \left(\dot{V} \right)^2 \right) \right)^{0.25} \tag{3}$$

where, as reference length, the fan diameter D was replaced by the smaller side length of the cross sectional square s. The value of s is determined for each fan type in a way such that s/D is minimized under the constraint that the installation loss is not higher than 2% considering the static efficiency (values of s/D as given in previous subsection). In *Figure 7*, the static fan efficiency is presented for the same fan designs than in *Figure 5*. The difference is that here, on the abscissa, the modified parameter $\delta^*_{install}$ is plotted. This means, at constant $\delta^*_{install}$ and duty point, we do not compare the efficiency of different fan designs with constant diameter D, but with constant duct cross section of the installation situation as outlined in *Figure 6*, taking into account the installation effects of the different fan designs in the duct.

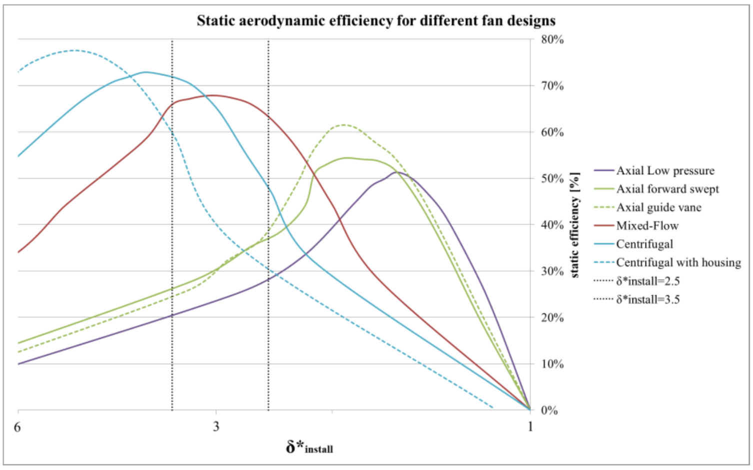

Figure 7: static efficiency curve of different fan designs

As expected, the results are distorted compared to the result shown in *Figure 5* especially in the direction of the abscissa. Looking at $\delta^*_{install}=2.5$, the mixed flow fan has an important benefit compared to the centrifugal fan design. The mixed flow fan design has for this installation effect a more pronounced region where it has significantly higher static efficiency compared to the centrifugal fan design. The reason is that mixed flow fans of bigger diameter can be used in the same installation space with low installation losses as a result of the improved behavior under this kind of installation condition.

At $\delta^*_{install}=3.5$, the centrifugal fan design has significantly superior properties without housing. The obvious reason is that a housing applied in this kind of device has particularly negative installation effect, or a comparatively small fan diameter has to be chosen

CONCLUDING REMARKS

Most of the fan requirements and demands for fan applications as for example air handling units, residential ventilation units, heat pumps, chillers, ventilation fans, or air chimney fans, come with an inherent fan size limitation, which is an additional constraint to the required fan duty point. This fan size limitation can arise from installation space restrictions, weight restrictions, motor torque restrictions or cost limits. In the majority of these cases, the choice of the biggest possible fan diameter has the potential to achieve the highest static fan efficiency under operating conditions, equivalent to the lowest energy consumption. Thereby it is assumed, that the rotation speed can be chosen freely, what is for these applications more and more the case due to the availability of electric drives with adjustable rotation speed.

A large number of fan demands have been analyzed and the corresponding size limitations were attributed. Assuming that the maximum possible size is the best choice for the achievable static efficiency, the fan size attributed to the demanded duty points was supposed to be fixed to this maximum fan diameter.

A non-dimensional application parameter δ^* was derived incorporating a prescribed duty point and the pre-determined fan diameter. The efficiency curves for different recent fan designs were analyzed, whereby the static efficiency was plotted as function of the application parameter δ^*. The value of δ^* is fixed by the fan demand itself, so this represents the basic requirements of a fan application. The efficiency (or low noise level) a fan design can achieve is then the “response” of a fan design to this basic requirement. It was demonstrated that depending on δ^*, different fan designs show the best efficiency. These different fan designs (low pressure axial, high pressure axial, mixed flow, centrifugal, axial with guide vane and centrifugal with housing) each cover a region of values of δ^* where they are the best choice. From this, the strategy arises, to cover an entire range of values of δ^* with different fan designs in order to be able to provide fans with good efficiency for all values of δ^* of interest.

This reflection was extended to the case that the size limitation comes from a particular installation effect, namely an axial duct. Due to installation effects, which depend in a strong way of the fan design, the size limitation for different fan designs in such a duct lies for a given duct size at different fan diameters. This effect distorts the picture obtained by just constraining the fan diameter to the benefit of fan designs optimized for low installation losses for the given installation condition.

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ANNEX: VARIABLE SYMBOLS, DEFINITIONS, UNITS

D	$[m]$	Inner diameter of the fan housing
$r_{ref} = \frac{D}{2}$	$[m]$	Inner radius of the housing, used as reference length
n	$[rev/s]$	rotation frequency of the impeller
ρ	$[kg/m^3]$	fluid density
$A_{ref} = \pi r_{ref}^2$	$[m^2]$	reference area
$u_{ref} = 2\pi r_{ref} \cdot n$	$[m/s]$	reference velocity
$\dot{V}$	$[m^3/s]$	volume flow
M	$[N \cdot m]$	shaft torque
$P_{shaft} = 2\pi M \cdot n$	$[W]$	shaft (input) power
Δp_{stat}	$[Pa]$	static pressure rise of the fan
$p_{dyn,x} = 0.5 \cdot \rho \cdot v_x^2$	$[Pa]$	dynamic pressure contribution of axial velocity v_x
$p_{dyn} = 0.5 \cdot \rho \cdot \vec{v}^{\,2}$	$[Pa]$	dynamic pressure
$\Delta p_{tot} = \Delta p_{stat} + p_{dyn}$	$[Pa]$	total pressure rise of the fan
$\varphi = \frac{\dot{V}}{A_{ref} u_{ref}}$	$[\ ]$	dimensionless volume flow
$\psi_{stat} = \frac{\Delta p_{stat}}{0.5 \cdot \rho \cdot u_{ref}^2}$	$[\ ]$	dimensionless static pressure rise
$\psi_{tot} = \frac{\Delta p_{tot}}{0.5 \cdot \rho \cdot u_{ref}^2}$	$[\ ]$	dimensionless total pressure rise
$\lambda_{shaft} = \frac{P_{shaft}}{0.5 \cdot \rho \cdot A_{ref} u_{ref}^3}$	$[\ ]$	dimensionless shaft power
$\eta_{stat} = \frac{\varphi \cdot \psi_{stat}}{\lambda_{shaft}}$	$[\ ]$	static efficiency
$\eta_{tot,x} = \frac{\varphi \cdot \left(\psi_{stat} + \varphi^2\right)}{\lambda_{shaft}}$	$[\ ]$	total efficiency with mean axial velocity
$\eta_{tot} = \frac{\varphi \cdot \psi_{tot}}{\lambda_{shaft}}$	$[\ ]$	total efficiency
$\delta^* = \left(1 + \psi_{stat} / \varphi^2\right)^{0.25}$	$[\ ]$	fan application parameter
$\delta = \left(\psi / \varphi^2\right)^{0.25}$	$[\ ]$	specific fan diameter
$\sigma = \left(\varphi^2 / \psi^3\right)^{0.25}$	$[\ ]$	specific fan speed