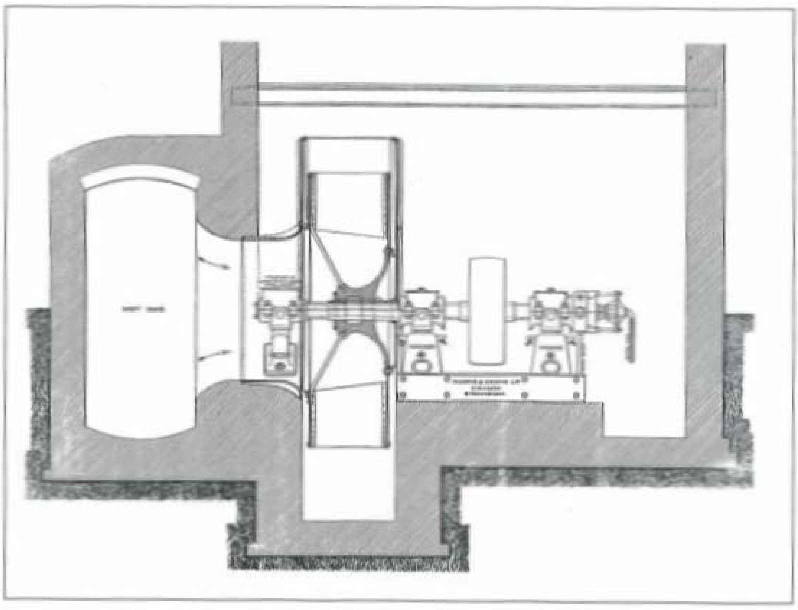
Suggested axial flow fan efficiency requirements

Introduction

The fan industry is characterized by a staggering diversity in fan sizes, applications and technology. In an analogy to the motor vehicle industry it spans from motor cycles (e.g. PC fans), cars (e.g. ventilation fans), trucks (e.g. industrial fans), heavy duty trucks (e.g. mining fans) and tractors/ fire trucks/ tanks/ mobile cranes (e.g. Atex fans, smoke extract fans, military fans and construction exhaust fans). The aim wanting to try and find just a few numbers to fit all should probably be reviewed. A large size axial fan for a clean room can and must be built with completely different design criteria compared to a mining fan. It is evident that the design criteria also have an impact on the fan efficiency. By restricting the number of categories will lead to efficiency limits having to be set at the lowest denominator instead of getting the maximum benefits.

The fan industry is a very mature industry which from the outset has designed fans with high efficiencies. Below is an example of a fan one of our companies built in 1905. Please note the bell mouth inlet, the aerodynamically very efficient shroud, the minimized air gap and the tight manufacturing tolerances. The fan efficiency will certainly have been more than 70%.



Alldays & Peacock Fan from 1905 for a steel work extracting hot gases

Contrary to other industries the room for improvement for industrial fans is very limited:

- Maximum efficiency is 100% (A fan is not a perpetual motion machine
- The available IE3 motors have efficiency between 80 % - 96%.
- There are unavoidable losses inside the fan
    - friction on the walls of the fan,
    - losses in the air gap,
    - impulse loses on the blades and the hub,

  - friction across the fan blade,
  - friction across the motor
  - impulse/ friction losses on the motor support
  - friction losses from the cables from the motor
  - losses from the rotational energy (perpendicular to the air stream)

- The tip speeds of the blade and therefore the maximum airspeed inside the fan are between 75 - 150 m/s, i.e. 250 km/h - 500 km/h perpendicular to the airstream! Even with the best available technology, any fluid mechanic professor can confirm, that aerodynamic losses for axial fans of less than 10% are not realistic, without turning the fan into a compressor/ turbine or jet engine machining every component at vast expense.

- At least 2-4 % design tolerances must be taken into account. Most industrial fan makers in Europe are small companies of 40 - 100 people and they make on average 50 - 100 fans of each type/ size per year. Demanding tolerances for each component such as are known from the car industry would not practically be feasible.

Conclusion: For "normal" industrial axial flow fans a total fan motor efficiency (FME) of ~ **9% - 83%** for axial fans**,** depending on motor size (0.75 kW to 500 kW) is the maximum efficiency that can be demanded and can be achieved for example in clean room fans. (See also the analysis in Appendix 1 with the data in appendix 2 & 3.)

Allowances

However further allowances must be taken into account:

- **Allowances for discrete working points**

  The EU directive 327 covers fans between 0 and ~10.000 Pa and between 0 and ~3.000.000 m³/h. The values required for industrial fans are set by the customer's applications. For a given design, besides the inlet fan diameter and the fan speed, the key variables the manufacturer can vary are the impeller diameter and blade width.

  Just like the fan speeds are in very discrete steps (2, 4, 6 ... poles), for practical purposes the geometric sizes of the fans are in discrete steps also. Typically they follow a logarithmic scale of sizes: 100, 112, 125, 140, 160, 180, 200, 224, 250, 280, 315, 355, 400, 450, 500, 560, 630, 710, 800, 900, 1000, 1120... in mm which have been standardised just like motor frame sizes have been.

  It follows that not every fan duty point can be met with exactly the optimal combination of speed, inlet size, impeller diameter and/or width. While it may be possible to get the optimal speed with variable speed drive this is not possible when it comes to casting of aluminium impellers. In practical terms this has in the past been overcome by making V-belt drive arrangement with a difference of 6 % between pulley sizes.

  Also axial fan technology has a typical volume flow/ pressure combination area where they work best and the flow is more or less laminar. However there are many application where a low flow must be combined with a slightly higher than optimal pressure. In those cases the flow becomes more disturbed and the total efficiency will suffer. In appendix 1 can be seen how for working point slightly above the optimal volume/ pressure combination the FME is reduced by around 10%.

When comparing axial fans to centrifugal fans one can see that there is less ability from a manufacturing point of view to match discrete points, since for centrifugal fans it is possible to reduce the width and the length of the blades, which is not really possible for cast aluminium blades. (see the example selections in the appendices 2 & 3.)

For axial fans therefore in order to be able to provide a solution for all low pressure applications and in order to do so economically with discrete working point, it is necessary to have a margin up to the theoretical maximum bench mark efficiency of at least 10%

- **Allowances for application specific design requirements**

It is difficult to list all the different design requirements which the customers and their applications make on fans, since even after many generations of fan making, we come across ever surprising uses of fans. It would severely limit the development of many other technologies if the use of fans were to be too severely curtailed. We doubt the whole computer development and transistor technology would have been possible, if from the outset the available fans would have been restricted to high efficiency fans only. (It is only an example, since PC fans are outside of scope, but remember that every car will today at least have 3-4 fans built in.) These are some of the design modifications we as Witt, makers of bespoke fans, have to do to comply with customers wishes:

- In many applications we have to use an externally cooled motor in an axial fan. Apart from dust loads, toxic fans etc. which are exempted, other examples include the whole food industry where the air is not allowed to come into contact with electrical motors for fear of contamination from the lubricant (grease) etc. Also applications with 100% humidity, e.g. the pulp and paper industry, we have to encapsulate and seal the bearing to protect them from the conditions. We apply a correction factor of at least [redacted].

- For many centrifugal applications, rarely, but also occasionally for axial fans, for example when we have externally cooled motors as above, we need to provide gastight shaft seals, not only for toxic applications etc. excluded fans. There are many industrial, chemical or agricultural uses where it is unacceptable to have odours or smoke to escape from the area around the shaft. We include a [redacted] reduction caused by friction with a [redacted] depending on the type of shaft seal that has to be used.

- While it can be argued that V-belt driven centrifugal fans can be replaced with a directly driven fan with a speed control that is not the case for axial fans. We have a few hundred applications every year where it is not allowed to have even an encapsulated motor, but that the motor has to be completely outside the air stream. Sometimes it is because the motor is simply too large to fit inside the motor (or as will be the case in the future will by its size reduce efficiency too much). In many cases it is because the gases/ substances being moved by law or practical considerations exclude having a motor even indirectly in the airstream. Reasons can be to avoid the extra energy costs (and reduction in real efficiency) by having to build a second fan for the external cooling or too high humidity in the air or laws of hygiene etc. We apply a correction factor of [redacted]. Giving

an allowance would not create a loophole, since an axial fan with V-belt drive is 2-3 times dearer than a normal axial fan.

- In some cases we need to make axial fans with external motors on a shaft. Examples are dual drive fans which can be driven both by an electrical motor and a gas turbine or a diesel engine for emergency use. [redacted]. Giving an allowance would not create a loophole, since an axial fan with a shaft drive is 3-5 times dearer than a normal axial fan.

- Increased air gap is a must in all our rugged applications. Mining, large furnaces, steel works, glass works, oil platforms, fresh air supply to metro and tunnel construction workers or wind turbine towers are just some examples that spring to mind. The external vibration levels and the recurring needs of many of them to be moved mean that the air gap between the impeller and the stator has to be increased. There is no good data on that. Many, many studies have been done to find a clear relationship. Obviously the problem increases with increasing pressure rises. [redacted]. However that is only an average figure.

- Atex fans (currently excluded). These fans need to have brass lining around the impeller, a larger air gap, stronger motor supports; the motors are larger with much larger terminal boxes and for centrifugal fans certain minimum distances between the housing and the impeller. The actual losses are dependent on the dynamic pressure part of the total pressure rise. [redacted]

[redacted] In cold climates it is not uncommon to reverse the flow during summer and winter, e.g. for the steel production in northern Sweden. All axial fans can be reversed, however if "true reversibility (more than 80% flow rate in the reverse direction) must be achieved the fans needs to be modified for example with less efficient symmetrical impellers, straight guide vanes etc. [redacted]

[redacted] Regularly, maybe for 1-2 % of our business we have to free form axial fan impellers or centrifugal impellers. The reasons are legion: Ventilation fans for the Norilsk nickel mine where the steel had to be made of special alloys that could stand some knocks at -60°C during winter storage; tip speeds exceeding some 150 m/s, which for many designs is the maximum possible with aluminium. Impellers made of titanium for special strength properties. So instead of using cast blades or hubs we have to make free formed, welded hollow bladed designs which have a much higher manufacturing deviation and tolerances than castings have. [redacted]

- Another example of having to design with a lower efficiency is fans for the military (bunkers), which have to have special shock and vibration properties to survive a 60 times gravitational bomb blast and therefore have much thicker motors supports,

impellers etc. which is of course detrimental to the efficiency. A recent example are fans we made (sponsored by the EU) for the clean up crews of the nuclear submarines on the Kola peninsular in Russia. They have to be very rugged inside and have larger air gaps to withstand the treatment they receive from the mobile Russian crews. We applied a reduction [redacted]

- <u>Dual use smoke fans</u> today are virtually all reversible fans; however they need additionally to have large air gaps. We reserve [redacted]. (The impulse and friction losses of air molecules hitting the guide vane can be influenced by the angle they have to the airstream. The further right of the fan curve you operate the flatter the guide vane angle relatively to the blade angle.)

- <u>Fans with multiple working points and anti-stall properties</u>. Technically a fan can be optimised for a single working point or for multiple working points. At the same time they must cope with relatively high pressure drops. For example all fans working with filters must be able to function with a changing resistance as the filters clog up. Also in arctic conditions the extreme changes in the climate make the use of variable resistance necessary. The solution is to add anti-stall rings which "remove" the stall from the blade tips of the axial fans. This can either be done with a bolt on device before or after the fan or is integrated in the stator of the fan. It costs [redacted] fans.

- There are unique combined designs where equipment is so directly integrated into the design of the fan as not being able to measure them without the influence of the equipment. An example is silencers, where for space reasons instead of a bolt on silencer the body of the fan, including the area where the impeller is operating is made with perforated steel, which will cause a reduction in efficiency of typically a few hundred Pascal depending on the flow speed.

<u>Conclusion:</u> Practical manufacturing considerations and the vast range of products covered by this regulation, which all are justified and cannot be eliminated (We can all be against war, however if we build bunkers they have to have fans that work even after a bomb blast.) Since the theoretical limit for fan technology is **69% - 83%** FME for fan above 0**.75 kW,** at least a further **9 % - 13 %** allowance must be included for discrete working points, so a maximum reasonable limit is **60 % - 70 %** total FME. In addition further allowances should be included in the regulation:

- [redacted] dynamic pressure allowance for axial fans with externally cooled motors, V-belt drives and external shaft drives.
- Increased air gap should get [redacted] total fan efficiency allowance.
- Shaft seal need [redacted] total fan efficiency allowance (optional, as long as fan can be tested without the seal, but that should be made explicit).
- Free form fabricated axial blades (Non cast blades) have an additional [redacted] manufacturing tolerance allowance.

- Atex fans to be excluded since otherwise there should be different limits for the different explosion classes and flammable gases are in any case excluded. Otherwise at least [redacted] total fan efficiency allowance should be applied.
- Dual use fans to be excluded since the influence of the temperature is different for F300, F400 and the increasingly used F600. Otherwise at least a [redacted] allowance should be applied.
- Reverse flow fans should have a [redacted] allowance.
- Make allowances for anti-stall rings or make it explicit that the values are without such integrated devices.
- Make completely explicit how non removable design changes to the fan itself can be dealt with in terms of the regulation.

Please note that dismissing the need for some of these allowances by writing into the directive that only the best possible values for the fan are applicable and the fans can be measured without efficiency reduction measures, would not be helpful. The main purpose of the label on the fan is to provide the customer with the data he requires. Practically to write efficiency numbers on data sheets and labels which do not correspond to the reality of the product and is different from the values found when being measured during a factory acceptance test (FAT) is at best confusing and at worst will result in endless legal disputes with customers and their lawyers who do not (want to?) understand the details of the directive. No customer would permit a performance test to be legally binding if for example the anti-stall ring, shaft seal or motor cover is removed.

Also in the industrial market, each fan being unique, using values from a type test is neither practical nor would it satisfy the legal requirements in the customers specifications. We need a directive that, maybe with the exception of external add on equipment such as protective grills or bolt on silencers, covers the fans as built.

Benchmarks and discrete working points

There has been a lot of discussion about the published benchmark figures in the 327/2011 regulation. The industry is convinced that for the larger fans the numbers where Total Fan Efficiencies not taking into account the motors. VHK seem to believe they were too low for larger fans, hence the proposed slopes which were to exceed the values laid down in the fan regulation.

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

In the following we have selected axial fans for the normal combinations of volume flow rates and pressures our customers require from us. For most working points there is the "standard" fan selection that would be sold today and an "optimal" fan selection with the highest possible efficiency for the given duty point. The graph below shows the Total Fan Motor Efficiency (FME) for both alternative selection strategies as a function of the electrical motor input power. Please note that the values used were measured in a laboratory under ideal conditions and minimized manufacturing tolerances.

[redacted]

The **red** lines are the "optimal" selections, the **blue** line the "standard" selections and the **black** line is the benchmark line from the EU directive 327/2011.

The individual fan selections and curves can be found in the appendix 2 & 3. Fans have been selected between 125 Pa - 2500 Pa and 4.000 m³/h - 500.000 m³/h with fan sizes between 250 mm - 3550 mm which is the Witt & Sohn axial fan range. The resulting motor electrical input powers were between 0.75 kW and 350 kW.

The selections reveal the following:

- For a given duty point there are many possible selections. In all cases, it is a compromise between size, noise level, power consumption and price.

- There is a narrow volume/ pressure band where there is little difference between the optimal and the normal fan selection. The further the fan has to be selected away from that band the worse is the performance.

- There is an area of low volume flow rate and higher pressures which have significantly lower efficiencies due to the less laminar flows inside the fans.

- The EU benchmark line, based on ISO 12759, is a surprisingly good fit, more or less between the red and blue lines. As was to be expected a bit too optimistic for small fans as the motor efficiencies were not sufficiently taken into account, as we have stated before.

- The effect of fans with larger motors having higher efficiencies is a truth with modifications as these are often fans with higher pressure rises which negates the small positive effect of the larger motors, i.e. the larger motors are need because a higher pressure rise is required making the fan work in a pressure/ volume combination which has a lower efficiency. These can especially be seen as the **blue** and **red** dots below the curves. The individual fan curves can easily be recognized when drawn in a volume - pressure diagram as they have increasingly pronounced stall areas in the fan curve. (See graph below)

- If the **red** dots would be disallowed certain volume/ pressure combinations would not be possible or lead to suboptimal designs. (For example selecting a fan with a higher volume flow rate and discard some of the air flow through additional outlets.)

- The "optimal" efficiency selection is always the most expensive option and nearly always resulting in a larger fan.

- The difference in efficiencies between the optimal and the usually selected fan is around 5 %. However, the difference in the power consumption in the specified duty point is only 3%., because the best efficiency point for larger fans often is further away from the duty point. (See appendix 1).

- In Appendix 1 the figures show that to get an average **5 %** improvement in the power consumption for Witt fans an average of ~40% higher price must be paid! **No contractor is willing to pay that for a real gain of only ~3%!**

- This large price difference makes it almost certain that if the limits are set too close to the benchmark, massive non-compliance will occur in the attempt to win orders with a little "creative" selection or documentation.

The key point is that: Although one may argue that the **red** "optimal" selections should set the basis for the regulation, one cannot live with a standard that "outlaw" the blue points. The red dots below the red line are not "bad fans", but are fans that have to work at somewhat lower or higher pressures compared to the optimal volume/ pressure combinations. Banning these types of combinations would effectively mean that fans with blade angles with some stall are being banned.

Conclusion: Although the benchmark found in the directive is generally correct, it would be scientifically incorrect to conclude that this is the goal towards which all fans can be pushed. Such a high limit would cripple the application of fans to a narrow volume/ pressure band of fans. The

economic incentive to "cheat", because the price difference is more often than not over 30%, especially in the near total absence of market surveillance would make the whole regulation become grossly unfair and at worst absurd. For extraterritorial supplier, where fines for non compliance would be hard to collect, the incentive to dump products accompanied by fake efficiency values would be irresistible.

In general: The steeper the fan curve, the lower is the maximum efficiency. (Same impeller, but with different blade angles.)

Suggesting a new limit

In finding a proper minimum efficiency limit an important factor to take into account is the cost element of the different types of fan designs. One of the important differences when it comes to axial fans is the difference between vane axial and tube axial flow fans. All the fan selections analysed in the previous chapter are vane axial fans, i.e. there is a regain of dynamic rotational pressure. Clearly the benchmark values in the regulation were determined by looking at vane axial flow fans.

Since Witt & Sohn doesn't make tube axial fans we can't provide too much information as to their performance. However the study published by the DOE (US Department of Energy) showed how at the lower level of efficiencies the tube axial dominate with 67% of the market, while at the top 30% of efficiencies vane axial dominate and would be 76 % of the market. In other words, setting the

limits only based on vane axial fans would mean that almost 60% of all fans produced today would be eliminated from the market, almost all of which are tube axial fans!

To me that sounds a bit too drastic, since that is what many manufacturers make exclusively and they would have to develop completely new fan ranges, competing with well established competitors in these ranges.

We would suggest the following (again only talking about industrial fans above 0.75 kW):

- Base the regulation on ISO 12759 and but set a higher Efficiency Grade

- Take the old benchmark limit with an efficiency grade of 75 and subtract:
    - 10% taking into account the difference in standard & optimal selection and the allowance for discrete working points at higher and lower pressures (which is basically the same).

    - Add another 3% allowance to allow for tube axial fans.

- That would mean a new minimum efficiency grade of **62** for a total increase in 12 efficiency points since 2013. That may not sound like much, however if 75% (at 11 kW) is the maximum achievable total fan efficiency it takes fans from a minimum total efficiency of 50/75 = **67%** to 62/ 75= **83%** an improvement of **24%**. I doubt the washing machine industry or the ventilation industry is so close to their theoretical limits. (And certainly not the kitchen hood manufacturers.) To compare it to other related industries I want to point out that the difference for the motor manufactures minimum efficiency levels between IE2 and IE4 at 11 kW is only from ~92 % to ~93%. So if the motors only make a gain of around 1 %, why must the fan industry make more of a gain than 24% within the same timeframe? What evidence exists that the motor industry is more mature than the fan industry? They started at the same time in the 1860s, one of the pioneers being our company Alldays & Peacock.

- Give allowance for V-belt and shaft driven axial fans, fans with encapsulated motors, reversible fans, dual use fans, fans with larger air gaps, fans with multiple working points (anti-stall) and fans with shaft seals.

- To push the limit higher now would make many industrial fan producers take the easier way out. Claiming compliance without actually doing so. Because the average industrial customer only really cares about the power consumption in his working point and there is very obviously no market surveillance, it would be the more sensible strategy and for many the only route to survival.

Summary

The fan industry, particularly the industrial side of the industry operating typically above 0.75 kW is a very mature industry with well know technologies. Low efficiencies are mainly driven by commercial aspects rather than technological. Whereas other products, especially as they incorporate electronics or other control systems may have technological improvement possibilities, no such new technologies for fans are even on the drawing boards in any university laboratories.

The stated goal of wanting to reduce fan categories and unify the slopes is in our opinion heading the wrong direction leading to lower then possible efficiency targets as there is a market need for most fan types and we risk ending up with the lowest common denominator. We recommend to: increase the sub categories, increase the efficiency levels, decrease the exemptions and provide more allowances for special designs or applications. That way we get closer to the real energy efficiency potential of the fan industry.

The theoretical FME limit for fans with existing motor technology is somewhere between **69% - 83%** depending on the motor size. The suggested new FMEG of **62** would set the new limit at ~**85%** of that limit. To push that limit, especially since the majority of fans do not operate continuously, is economically not to be sensible for axial fans as the additional costs of achieving the additional efficiency is higher then the benefits over the life time of the product. For all intends a purposes it should be accepted that the practical limit is ~85% total fan motor efficiency for medium sized and large fans.

The problem with fans is that they span such a wide range of application from clean room fans working in a very controlled and stable environment to heavy duty mining fans that have to withstand severe mechanical shocks. Also fans with higher the pressures are normally have lower efficiencies (Hub ratios change, flow speeds increase, blade angles are steeper etc.). To reduce pressure losses would obviously be a way to reduce losses and improve efficiency; however the trend is the opposite. With ever increasing demands for heat recovery, system efficiencies in the equipment and lower noise requirements the pressures needed are continuously increasing.

To accommodate the discrete working points over the whole economically sensible volume flow/ pressure range we recommend a margin to the maximal possible benchmark of 13 % using the ISO 12759 curves. In addition special allowances for special designs must be included in the revised regulation.

Sticking to the ISO 12759 relationship between motor size and total efficiency, developed specifically for use by the 327-2011 regulation, is a sine qua non. Not only is it a reasonable fit as the analysis has shown it has become the basis for the rest of the world's efficiency measures. Coming up with new slopes would put unnecessary stumbling blocks in the way of this very successful industry. The European fan industry is one of very few European industries where we actually technologically and commercially are world market leaders and most firms operate to some degree globally, with many having the majority of the business outside the European Union. Since many countries have adopted the ISO 12759 as their local standard to base their efficiency regulations upon, it would mean that the European industry suddenly would have to develop different fan ranges for different markets and put them at an unfair disadvantage with local competitors around the world.

An increase to an FMEG grade of 62, forcing the industry to having to make all fans to around ~85% of the practical efficiency limit, is a very large step and the impact on the industrial fan market, eliminating, we would estimate, based on the US DOE study, probably another 25 -30% of the fans sold today, mainly tube axial fans. The savings in terra watt is probably a factor 10 higher than the electrical motors in the same time frame.

Appendix 1: Axial fan selection

Using the Witt selection programme a representative number of volume flow rates (from 4.000 - 500.000 m³) and total pressure increases (125 Pa - 2500 Pa) typically for axial flow fans were selected. In total 84 duty points were selected, representing ~ 150 different axial flow fans, all being selected from Witt & Sohn high performance designs. For each working point was noted down the fan selection with the highest Best Efficiency Point (BEP) and the selection that commercially most fan manufacturers would offer their clients and the customers would normally want to buy.

The fan data was derived directly from prototype fans measured in the Witt & Sohn AMCA certified laboratory. Note the values represent ideal conditions with manufacturing tolerances minimized.

For each selection was noted down the fan size, fan speed, the shaft power consumption in the duty point, the best efficiency point and the normal sales price. Using the default values for the motor efficiencies the combined Fan Motor Efficiency (FEM) at the best efficiency point was calculated. (See appendix 2 for all the individual data. In appendix 3 are put together all the technical data and fan curves of each selection.) From the results the following questions were answered:

1) What is the relationship between fan sizes and efficiency?

2) How much efficiency is gained by selecting only according to efficiency and how much more does that cost compared to the current practice?

3) What impact does the pressure rise have on achievable maximum efficiencies?

The following results were found:

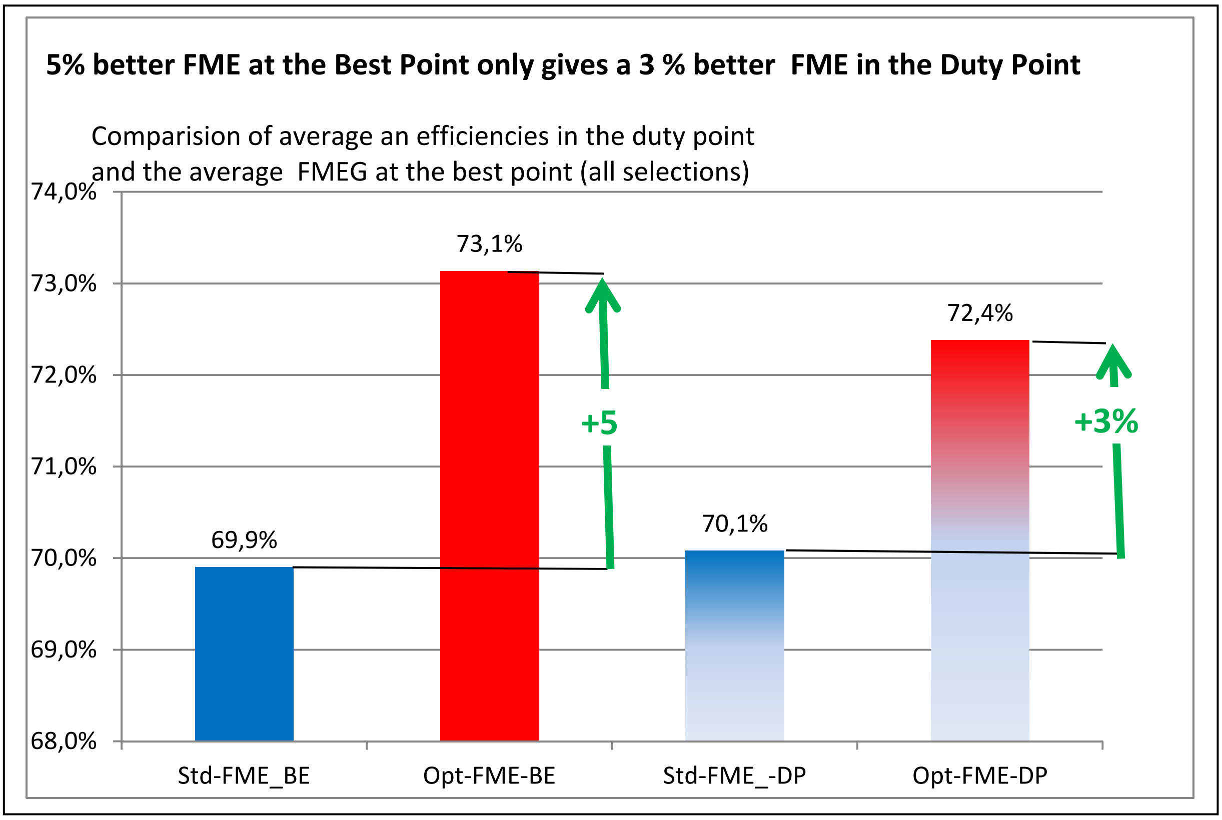
1) Correlation between efficiency and fan size

In the above graph is shown in **red** the most efficient selection for each data point and in **blue** the fan selection that today would have been offered to the customers. As can be seen there is a fairly clear correlation between fan size and efficiency. From size 355 to 1000 mm diameter the efficiency increases by 5%. From diameter 1000 mm to 3550 a further increase of 3 % is achieved. (When

looking at the individual data points, the logarithmic trend line seems to underestimate the increase in the small sizes and overestimate the increase for larger fans.)

2) Price difference optimal and standard selection

The question is whether the best efficiency selection yield concrete benefits. Below can be seen the comparison of the difference of the FME's for the best selection (on the right) with the actual efficiency in the best efficiency point and the actual duty points for all the selections made for this report.



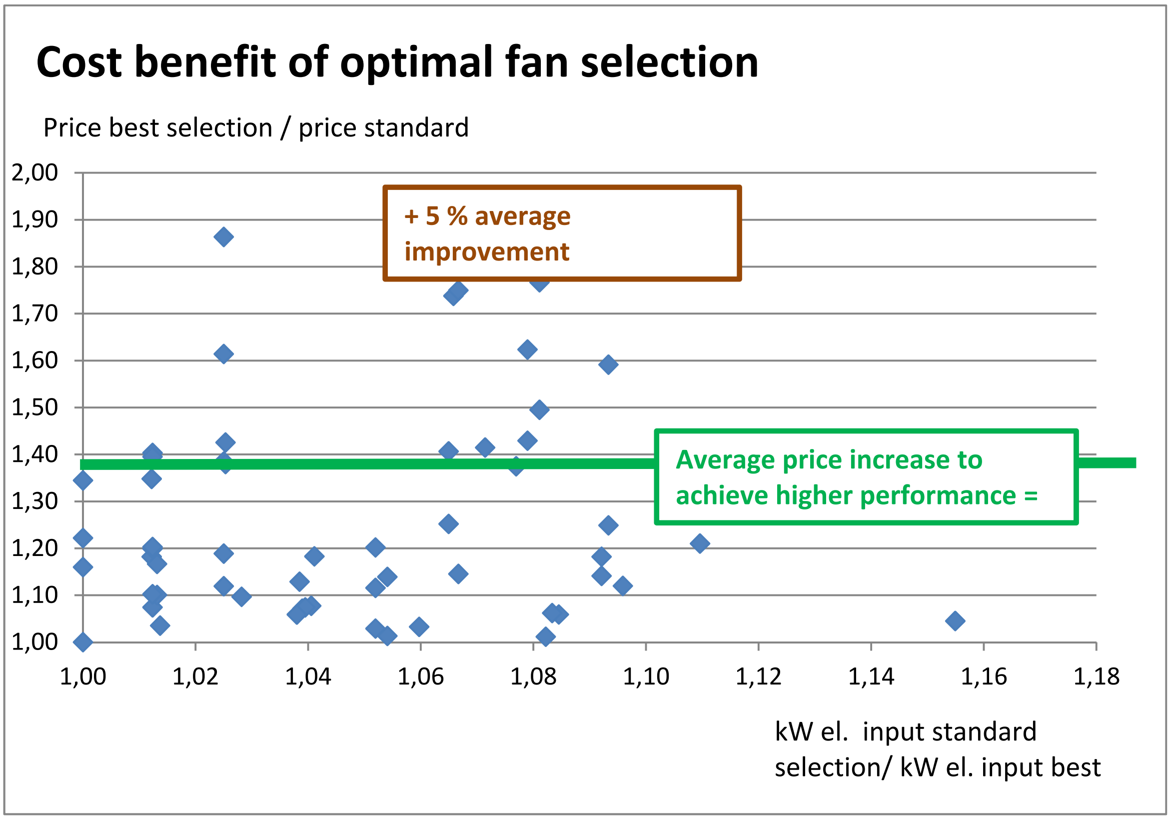
The optimal selections are again shown in **red** and the standard selections in **blue**. As could be seen from the other graph the difference between the 2 FME curves is approximately 5 %. However the gain to the end user is only 3 %! So 1/3 of the seemingly better selection is not real. The reason is that a fan curve which has a higher efficiency somewhere on the fan curve in many cases is not technically best fit for the job, because the best efficiency point is too far away from the operating point.

Below is shown on the Y-axis the difference in price as a factor between the best efficiency selection and the today standard commercial fan selection. On the X- axis is found the corresponding improvement in the power consumption in the required duty point. The **green** line shows the average difference in price between the two selection strategies, while the **brown** line shows the average improvement in electrical power consumption.

The figures show that to get an average 3 **%** improvement in the real power consumption and a 5 % higher efficiency rating an average ~40 **%** higher price must be paid!

<u>No contractor is willing to pay that! The incentive to cheat is massive, especially since the contractor and end user gain so little in real terms.</u>

Even in terms of energy savings that may be a very dubious investment. The average price for the standard selection was [redacted] with an electrical input power in the duty of [redacted] and for the comparable values for the best fan selection was [redacted] and [redacted]. Assuming electricity cost of 0,1 €/kWh it would take more than 21.000 hours to recoup the difference in investment. Typically an industrial fan runs ~8 hours a day during working hours i.e. around 2000 h per year. It would take more than 10 years just to break even. For emergency fans or fans only being used seasonally that break even will never be reached!



<u>The best selection i.e. the benchmark should not be the guiding principle for setting the efficiency limits.</u>

3) Pressure/ volume flow ratios

The market specifies the duty point that the fans have to achieve. As an industrial manufacturer, where each project is in general unique, there is in reality no influence on that. As stated in the main text, the trend is for pressures requirements to increase, because many processes have more stringent emission and/or noise requirements.

[redacted]

The key point is that for axial fans there is a large region of duty points, especial medium volume and higher pressures, where physics dictate that the benchmark efficiency values cannot be met.

Also it can clearly be seen that the effect of fan size is quite small. A benchmark around 80% is a realistic figure. With motor efficiencies for these tables between 86.5% and 96% this means a maximum benchmark value for small fans of 69 % (=80 % x 86.5%) and up to 79 % (= 82% x 96%) for very large fans.