Discussion paper

Maximum achievable fan motor efficiency taking into account application specific losses

Introduction

Fans are a little recognized factor in everyday life. They are found everywhere from hairdryers to kitchen hoods, air conditioning units and cars. When your average person thinks about fans, these are the typical examples that they will give you. In the fan industry these types of fans are thought of as commercial or residential fans.

What few people realize or think about is that there is a whole range of fans, which perform vital tasks in our society, but which few people ever see. Fans are in windmills, power plants, cement & steel plants, on transformers, in tunnels and metros, providing air to underground mining and to bomb shelters. These are collectively known as industrial fans. Whereas residential and commercial fans in general are manufactured by large companies in large quantities, industrial fans are in general individual units made by Small and Medium-sized Enterprises (SMEs) with on average 50 - 100 employees and are family owned. In terms of market size the industrial fans probably make up only 10% of the market in terms of units produced (if that?), but 80 -90 % of all companies in the industry.

In all the fan efficiency discussions we have had, the impression is that the main focus has been on the more readily available and well known commercial and residential fans. The special considerations that industrial manufacturers have to make to fulfill specific legislation (for example in terms of noise, pollution, and odor and smoke emission and / or safety rules) and customer specifications seem not to have carried so much weight.

It is the purpose of this paper to

- Detail the specific losses that industrial fans must take into account when they are designed
- Give specific examples of what order of magnitude those efficiency reduction amount to
- Show what maximum energy efficiency industrial fans can achieve, unless compensation factors for these losses are allowed

All examples are using vane axial flow fans, but the same arguments apply for centrifugal fans and many other fan types. The same industries use centrifugal fans as well. Some of the compensation factors are not required for centrifugal fans; however there are some additional ones, such as double inlet fan losses.

In the appendix are shown pictures from specific projects to illustrate these special fan designs. These are very proprietary and not to be shown publicly.

Design specific losses

A typical axial fan has a dynamic pressure share of the total pressure of 10 - 20 % in its Best Efficiency Point (BEP) as shown in figure 1. However some application demands a higher dynamic pressure figure due to space restriction. In the following is assumed a dynamic pressure of 15% of total pressure rise.

The design specific losses are in two categories. Mainly they are dynamic pressure losses because the air flow is in some way obstructed by necessary modifications to the fan (such as an increased air gap) or they are indirect losses which are unrelated to the dynamic pressure (such as shaft seal losses).

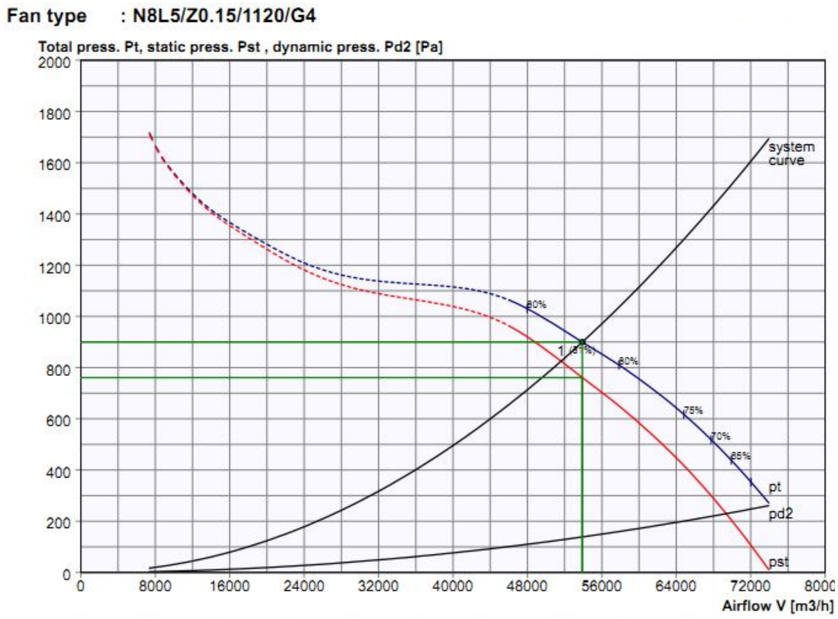

Figure 1: Standard axial flow fan curve

Typically the airflow is obstructed leading to dynamic pressure losses of 15% - 60%. Using these values in figure 2 one can calculate the resulting reduction in aerodynamic efficiency. Since the fan already has its own inherent aerodynamic losses (e.g. an aerodynamic efficiency of 75 %) the actual compensation factor is larger.

Design specific losses	Delta Pd loss		Efficiency reduction*	Compensation factor**
Anti stall	15%	30%	**2% - 4%**	**0,95 - 0,97**
External shaft motor	40%	70%	**6% - 10%**	**0,87 - 0,92**
Externally cooled motor	20%	40%	**3% - 5%**	**0,93 - 0,96**
Free form fan blades	15%	40%	**2% - 5%**	**0,93 - 0,97**
Heavy duty design	15%	40%	**2% - 6%**	**0,92 - 0,97**
Increased air gap	20%	50%	**3% - 7%**	**0,91 - 0,96**
Large terminal box/ extra thick cables	15%	40%	**2% - 5%**	**0,93 - 0,97**
Light/ occasional dust load (> exclusion)	10%	60%	**2% - 8%**	**0,89 - 0,97**
Reverse flow (100%)	30%	50%	**4% - 7%**	**0,91 - 0,95**
Reverse flow (80%)	20%	40%	**3% - 5%**	**0,93 - 0,96**
Shock/ earth quake resistant	35%	60%	**5% - 8%**	**0,89 - 0,93**
Star- delta motor / 2 speed motor (cables)	10%	15%	**1% - 2%**	**0,97 - 0,99**
Temperature motor	15%	20%	**2% - 3%**	**0,96 - 0,97**
V-belt drive	35%	60%	**5% - 8%**	**0,89 - 0,93**
Swing out type/ Access door/ 2 part fan	10%	20%	**2% - 3%**	**0,96 - 0,97**
High pressure	40%	60%	**6% - 8%**	**0,87 - 0,93**
Decontaminable			**1% - 3%**	**0,96 - 0,99**
High pressure blower (high hub ratio)			**2% - 6%**	**0,92 - 0,97**
Shaft seals			**2% - 3%**	**0,96 - 0,97**

*Assuming 15% dynamic pressure at BEP
**Assuming a 75% aerodynamic efficiency

Figure 2: Examples of application specific losses and compensation factors

In general, as can be seen from the figure 2, a compensation factor of 0.90 - 0.93 is the average. Please note that this is probably not an exhaustive list. Other manufacturers would need to be asked to complete the list. However, as is shown below, these design measures do not exist in isolation. Normally a fan design will have to incorporate a combination of these design elements.

Actual application losses

Below and in the appendix 1 are shown examples (and pictures) of actual industrial fan applications. They are a random selection of some of the fans that Witt has in their production at the moment. The efficiency reduction values are the ones we have included in our selection data, compensating for the various design features. They are obviously estimates, however are based on 100 years of experience with these types of fans.

Application	Efficiency red.	Comp. Factor	Application	Efficiency red.	Comp. Factor
Wind mill generator cooling	**11%**	**0,85**	**Military fan (bomb shelter)**	**15%**	**0,80**
- heavy duty design	4%		- shock/ earth quake resistant	8%	
- increased air gap	7%		- increased air gap	7%	
Mining fan/ steel mill	**14%**	**0,81**	**Smoke extract fans**	**12%**	**0,84**
- heavy duty design	4%		- increased air gap	5%	
- shaft seal	2%		- reverse flow	4%	
- V-belt drive	8%		- high temperature motor	3%	
Pulp & paper/ cement (small)	**10%**	**0,87**	**Pulp & paper/ cement (large)**	**13%**	**0,83**
- externally cooled motor	5%		- V-belt	8%	
- shaft seal	2%		- shaft seal	2%	
- heavy duty construction	2%		- heavy duty construction	2%	
- star/ delta motor (cables)	1%		- star/ delta motor (cables)	1%	
Wind tunnel	**12%**	**0,84**	**Atex blower (zone 1, infrequent gas)**	**13%**	**0,83**
- reverse flow	4%		- increased air gap	7%	
- externally cooled motor	5%		- heavy duty design	2%	
- free form fan blades	3%		- large terminal box	4%	
Nuclear	**11%**	**0,85**	**Fan with stand by motor (gas turbine)**	**10%**	**0,87**
- shock/ earth quake resistant	8%		- heavy duty design	2%	
- decontaminable	3%		- external shaft	8%	
Very cold climate ventilation fan	**13%**	**0,83**	**Food industry**	**14%**	**0,81**
- free form fan blades (- 50°C)	3%		- externally cooled motor	5%	
- reverse flow	5%		- shaft seal	2%	
- heavy duty design	2%		- increased air gap	3%	
- anti stall	3%		- high pressure (filters)	4%	

Figure 3: Efficiency reduction & compensation factors for different applications

Typically 2-4 of the efficiency reduction measures have to be combined to satisfy the application specific requirements. Since their obstruction or performance loss is largely independent of each other, they can be added together to come up with a total required compensation factor. As can be seen this overall factor is typically 0.80 - 0.85. In individual cases this can become higher; as for example when fan sizes have to be made smaller for overall economic reasons (e.g. building cement ducts for bomb shelters is very costly). A smaller fan with the same total pressure increase will have a

higher dynamic pressure and consequently the losses and required compensation factor is increased. In practice this is a very common occurrence e.g. when old tunnels or metros are upgraded to meet new fire safety standards with higher air volume flow requirements and it is prohibitively expensive to rebuild all the ventilation systems in the Paris Metro or London Underground. The problem is solved by using fans with the same diameter but much higher dynamic pressure. The result is fans with less than optimal efficiency.

Other very common examples of projects with design constraints are:

- Use of V-belt drives or angular drives (see photos), because it is prohibitively expensive to dismantle the whole fan for the inevitable motor bearing replacement every 3-5 years (24 hour) operation. (e.g. fans placed on the roof of the mining complex in Kiruna, Sweden)

- Swing out type fans because the underground installation site does not provide room to dismantle the complete fan. (e.g. fans on oil rigs or wind mill transformer platforms in the North Sea)

- Fan casing built in 2 parts because the replacement fan has to be brought into existing structures and opening made for much smaller fans. (e.g. the Paris Metro)

- Two drive train solutions with an electrical motor and a gas turbine used in the chemical industry where a power outage would have disastrous consequences for the whole plant if the fan stopped in mid process. (e.g. Norsk Hydro in Larvik, Norway)

Introducing compensation factors for these additional losses in the regulation, with correct and water tight definitions of the design features, would in our view not create not loopholes, as the resulting fans are much more expensive to manufacture.

Maximum practical efficiency

When setting limits the key question is: What is the maximum possible limit, i.e. what is the bench mark for axial flow fans? Figure 4 summarizes the maximum practically achievable limit for industrial axial fans.

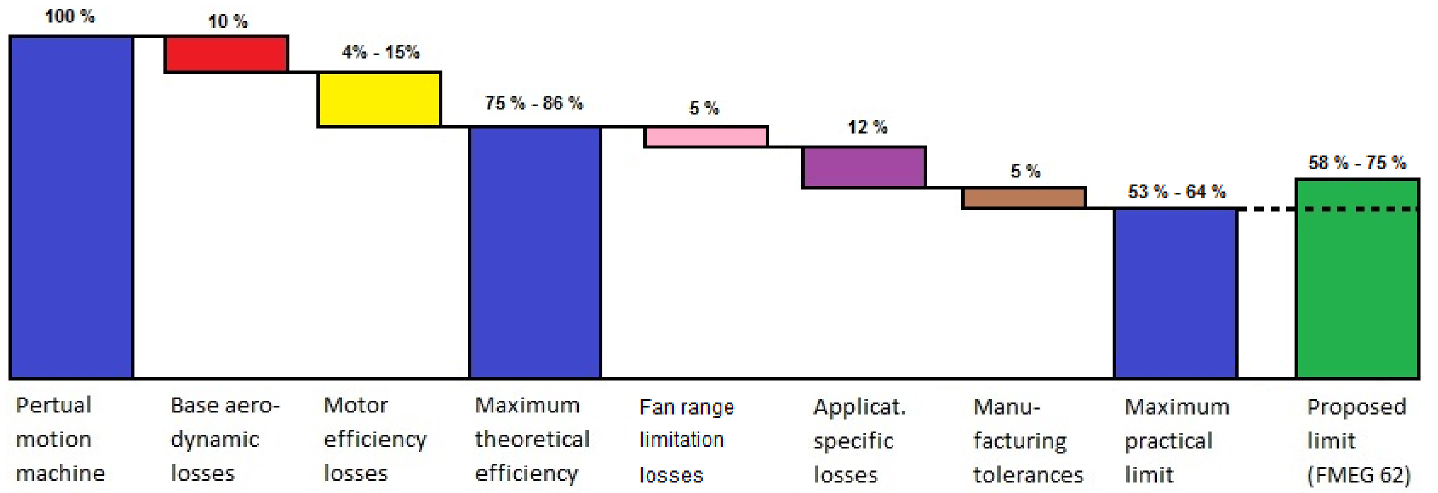

Figure 4: Cascade of losses for industrial axial flow fans (1 kW+)

The basis is the **100%** efficiency that a "pertual motion" machine would achieve. From that must be subtracted:

- The aerodynamic losses inside the fan, which are at least **10%**. These result from:
    - 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)
    - friction losses on the guide vane if the fan is a vane axial fan

  Extreme measures such as making miniscule air gaps, welding of blades to the hub etc. can maybe marginally improve on that, but the applications for which this is permissible, are normally outside the industrial fan applications and more used in commercial or residential applications. (For example in clean rooms or medium pressure ventilation units.)

- The losses from the motor as regulated in EU directive 640/2009. (IEC 6400034-31) and are **4% - 15 %** for motors larger than 1 kW.

This leads to a maximum achievable theoretical Fan Motor Efficiency (FME) of **75 - 86** %.As mentioned above, there are application specific losses that also have to be taken into account.

- Fans come in discrete sizes, fan speeds and blade angles. Every fan range has an optimal volume/ pressure area where near optimal fan efficiencies can be achieved. In practice, particularly blade angles, often have to be used which are slightly steeper (for higher pressure) or shallower (for lower pressure) to accommodate the customer's specific needs. Particularly the fan selection often has to be modified to have a steeper blade angle to take into account that the fan resistance is different from what the customer has calculated and a small mistake would lead to the fan stalling, if not corrected for. We have added a **5%** margin for that.

- If no compensation factors are allowed this will on average amount to **12 %** of losses. (See figure 3) However in individual cases it can be double that e.g. as mentioned above when for space restriction reasons a smaller than optimal fan size needs to be installed.

- Industrial axial fans are welded, one - off products, not mass produced. The ISO EN 13920 gives a tolerance for welded designs of for example 0.4m - 2 m (the size of a typical industrial fan) of ~+/- 6mm. That is a fact of the industry. When air travels at a maximum of 500 - 600 km/h inside an axial fan, at the tips of the blades, it should be obvious that even small variations in the dimensions or angles have a significant impact on the performance. For that reason the ISO 13448 allows a manufacturing tolerance of **5%.**

The above does not take into account the losses from specific fan designs. It should be clear that the above only really applies to vane axial fans. The large numbers of tube axial and propeller fans in the

market today, have additional losses, since there is no regain of the rotational energy in the guide vanes. Since Witt does not manufacture those types of fans we can give no verifiable values for the reduction in efficiency. In appendix 2 are shown some examples of propeller fans. We don't know whether they fulfill the directive today, but are certain they will not meet the more stringent requirements.

The means that without realistic compensation factors a maximum practical efficiency limit for FME is **53% - 64%**. The suggested limit of 62 (EVIA) is **58% - 75 %.** Even if the **5 %** manufacturing tolerance shown in figure 4 is disregarded, the suggested limit of FMEG 62 is at or above the maximum possible limit (i.e. **58% - 69%**), if extensive compensation factors are not introduced into the revised directive. Especially for the large axial fans over 50 kW the proposed limits are very challenging without compensation factors and can only be met by making use of the "permitted" tolerances. In other words, only with additional compensation factors, is the change from a FMEG 58 to FMEG 62 justifiable.

Conclusion

The nature of industrial fans is to deal with rugged and complicated applications, the designs of which have a direct impact on the achievable efficiency. In the current regulation there seem to be very little regard to those needs. This has been possible so far, since limits have been challenging, but for most high performance industrial fans just manageable.

In the discussions so far there seem to be a widespread reluctance to allow for compensation for special design features. It is our impression that this is because compensation factors are seen as potential loop holes which of course must be avoided.

Now to increase the limits without taking into account the design specific restrictions that industrial fan makers have in meeting their customers' specifications would create potential legal difficulties for the manufacturers. The customers (and their lawyers) will expect a certain efficiency grade as per the directive. It will be very difficult, unless specific compensation factors or allowances are given, to pass a factory acceptance test with a lower value, by simply arguing that the fan, without the special design features, would have passed. Building a "standard" fan is not an option given that industrial fans in general are one-off machines.

So whatever the limits are there should be no ambiguity in the regulation as to what can be excluded and what is not. What is obvious to engineers (and the people writing the ISO standards) may not be as clear to the lawyers.

In the absence of compensation factors the currently proposed FMEG of 62 is above the maximum possible limit. It pushes the rules over 100% of the practically possible Fan Motor Efficiencies achievable by the industrial fan industry for large fans. Since society needs those applications it would either drive the industry in illegality or force many of these industries outside the European Union if limits without compensation factors would be raised much higher. The impact on the SMEs working in the industrial fan industry would be dire. Therefore compensation factors such as indicated in figure 2 must be included in the regulation.