ZÁPADOČESKÁ UNIVERZITA V PLZNI

FAKULTA STROJNÍ

KATEDRA ENERGETICKÝCH STROJŮ A ZAŘÍZENÍ

ZÁPADOČESKÁ UNIVERZITA V PLZNI

POWER SYSTEM ENGINEERING, THERMODYNAMICS & FLUID FLOW

2013

JEDNOTLIVÝ PŘÍSPĚVEK ZE SBORNÍKU

esf evropský sociální fond v ČR

EVROPSKÁ UNIE

MINISTERSTVO ŠKOLSTVÍ, MLÁDEŽE A TĚLOVÝCHOVY

OP Vzdělávání pro konkurenceschopnost

INVESTICE DO ROZVOJE VZDĚLÁVÁNÍ

THE INFLUENCE OF THE STATIONARY PARTS GEOMETRY ON THE FAN PERFORMANCE PARAMETERS

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The aim of this paper is to summarise some of the experiences in the fan design particularly in the influences of the geometry of the main stationary parts on the fan performance in the aerodynamic and acoustic point of view. The scope of the following text is mainly to give some feedback from an industrial manufacturer to the scientific support.

Keywords: fan design, installation, performance, efficiency scaling.

Introduction

The last century ended up in the name of “the Environment”. Besides the fact that many other but environmental interests hide behind this magic word all the industrial branches are immensely pushed towards technologies as efficient and silent as possible. Regarding the fans not only for industrial use, this leads to highly advanced machines with enhanced blading assuming the cutting edge technologies in production.

However, even if the fan stage is “state-of-the-art” and its efficiency exceeds 90%, the final overall efficiency of the machine is obviously lower due to energy dissipation caused by stationary parts of the fan, which may also unwillingly cause unfavourable increase of the sound power. Unfortunately, these two aspects, namely aerodynamics and acoustics, often stand against each other and one has to compromise on the other.

The intention if this paper is to shortly summarise some of the followed aspects based on decades of experiences, development and cooperation with scientists, experts and universities, which hold ZVVZ, a. s. on the top of the fan production in the Czech Republic so far.

1. Inlet Geometry Impact

Considering the machine in the direction of the flow one should start the geometry discussion with the fan inlet. The influence of the inlet geometry has two parts. One is the geometry of the fan suction part itself (e.g. inlet chamber or inlet nozzle), which is designed by the manufacturer, while the other is a question of installation, for which the project architect is responsible. Apparently, both are equally important. A poor fan installation as e.g. a right angle turning of the inlet duct of a fan with supposed axial inlet causes an unpredictable fan performance reduction (e.g. Grönwoldt-Hesse 2012 [1]).

An example of such an installation is shown on Fig. 1. There are two installation faults from the aerodynamic point of view. It is very inconvenient to place a diffuser directly on the suction flange in any case (Fig. 1 on the left) as any flow disturbances develop in a decelerating flow. The flow disturbances on the fan suction side are sure to occur in this case since a right angle elbow precedes a rather short straight part of the duct (Fig. 1 on the right). Being asked to qualify the reason of the reduced air power of the competitor’s fan, one has to admit that the problem was at least partially on the side of the project. The quantification was obviously impossible.

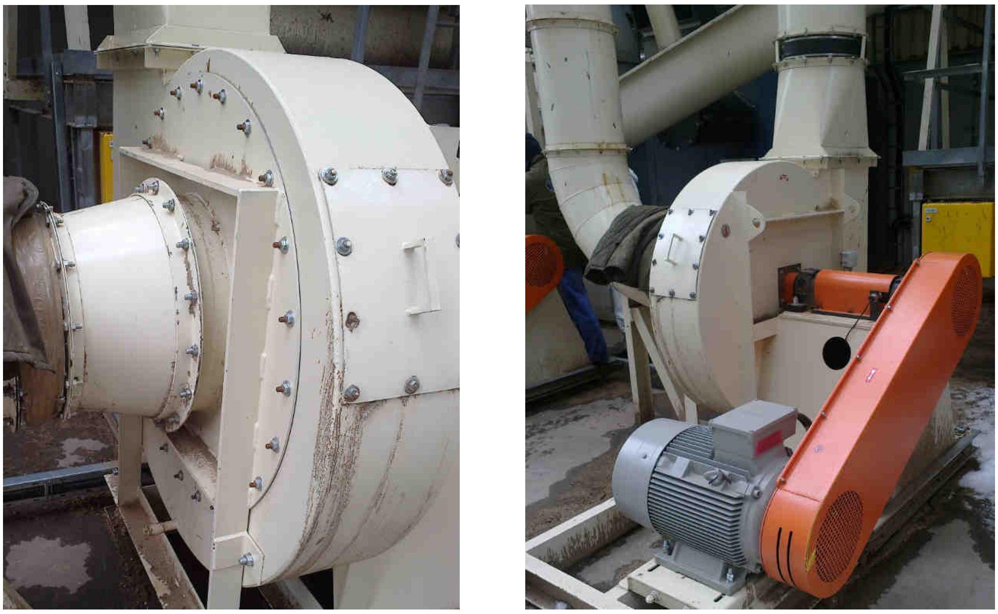


Fig. 1: Poor installation of a centrifugal fan.

The above situation may have been helped using a centrifugal fan equipped with a proper inlet chamber. However, the inlet chamber of poor design causes similar troubles. Fig. 2 shows a case of centrifugal fan, where the energy dissipation due to rather bad geometry of the inlet chamber prevails and being dependent on the velocity its influence on pressure coefficient and efficiency increases with the flow coefficient. Professor Cyrus has shown in his works that in the case of the centrifugal mid pressure fans the inlet chamber loss may be neglected for certain ratios of its inlet and outlet area.

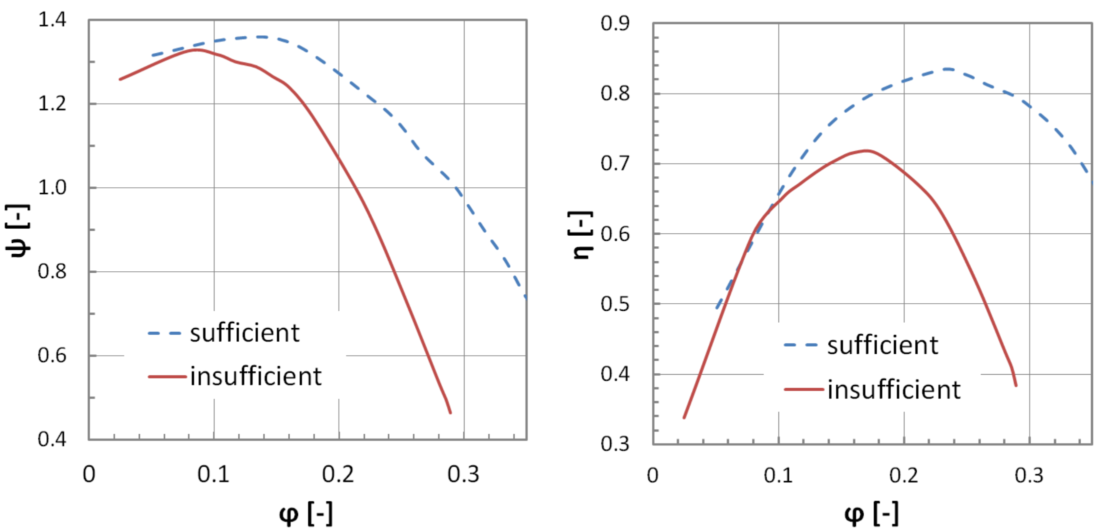


Fig. 2: Poor inlet chamber geometry influence on fan performance.

Generally all the fans are extremely sensitive on the inlet geometry regardless whether axial or radial. In the time ZVVZ started manufacturing mixed-flow fans (axial fans with meridional flow acceleration) their efficiency was about 75%. Mostly thanks to the optimisation of the inlet chamber these fans ended up with efficiencies about 85%.

Tab. 1 summarises the influence of the inlet chamber inlet area A_1 and of its aspect ratio a/b (all dimensionless with respect to the impeller diameter D) on the mixed-flow fans with smaller hub to tip ratio. Reaching the higher efficiency meant increasing the inlet chamber inlet area and its conversion from a square into a rather narrow rectangle.

a/D [-]	b/D [-]	A_1/D^2 [-]	η_{max} [-]
1.268	1.268	1.608	75%
0.887	1.577	1.399	83%
1.000	1.761	1.761	85%

Tab. 1: Influence of inlet chamber area and aspect ratio on efficiency.

Yet, the principal geometry of the inlet chamber itself had to be optimised as well. Fig. 3 shows the main difference being essentially the flexion of the flow (original on the left, optimised on the right).

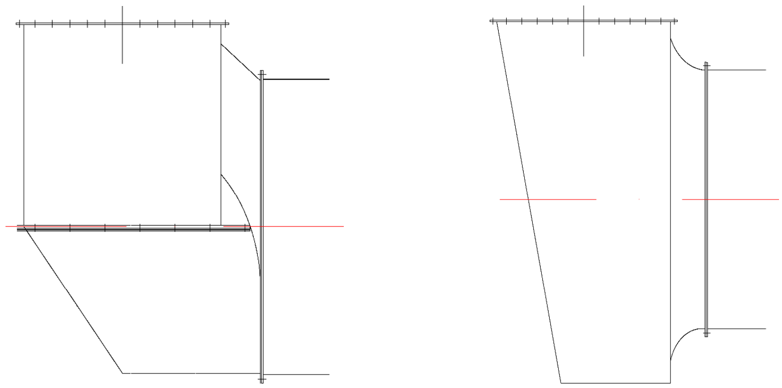


Fig. 3: Main difference in geometry of the mixed-flow fans inlet chamber.

The aerodynamic regulation of the mixed-flow fans has always been performed by generating an angular momentum in front of the rotor using inlet guide vanes (IGV) with mechanical flaps. This was historically common mostly in the former Czechoslovakia and East Germany according to Ušakov et al. 1962 [2]. However, according to Boehle et al. 2004 [3] the classical type of IGV consists of uncambered or slightly cambered profiles, the stagger of which can be varied, although he found in his research that IGV with mechanical flaps are more convenient from the aerodynamic point of view and induce less pressure losses especially if the knee is closer to the IGV leading edge. The largest turning angles can be than reached with small IGV loss coefficients in the case of the IGV cascade having the longest flap.

Modern power plants though require higher volume flow, which leads to axial-flow fans with high flow rates and large areas of high efficiency regulated by means of rotor blades variable pitch. Professor Cyrus has aerodynamically optimized the inlet chamber and the struts supporting the casing hubs (Cyrus et al. 2012 [4]) of these fans thanks to which the inlet chamber loss coefficient was decreased by approx. 3% in average. The modification of the struts also decreased the sound power level.

2. Stage Geometry Impact

The highest interest is obviously focused on the blading itself. Regarding the fans with axial-flow acceleration, there are of course various approaches to the rotor blade design such as controlled vortex design (Vad et al. 2008 [5]) or controlled diffusion air-foils (Savic et a. 2005 [6]). However, the common free vortex design of rotor and stator air-foils based on the NACA 65 series with reinforced trailing edge is commonly used in the process (e.g. Cyrus et al. 2012

[4]). Most, if not all, of the stages designed by Prof. Cyrus reach the efficiencies higher than 90% enabling us to cover the most important part of the performance map with reasonable combinations of fan size and rotational speed as shown on Fig. 4 for the example of axial fans for power plants for efficiencies higher than 80% and for density of 1.2kg/m^3.

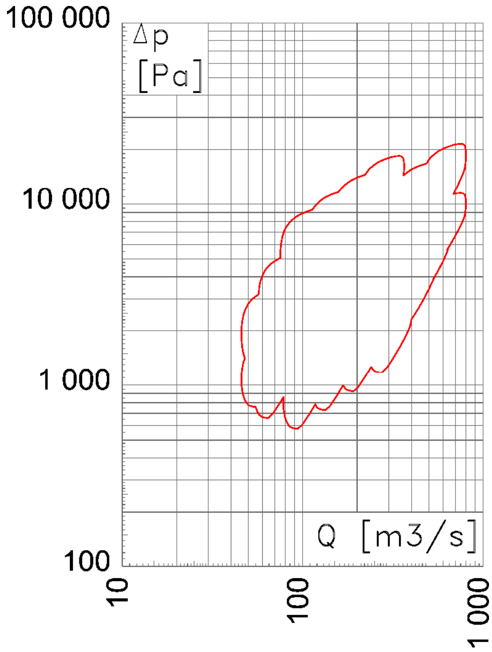
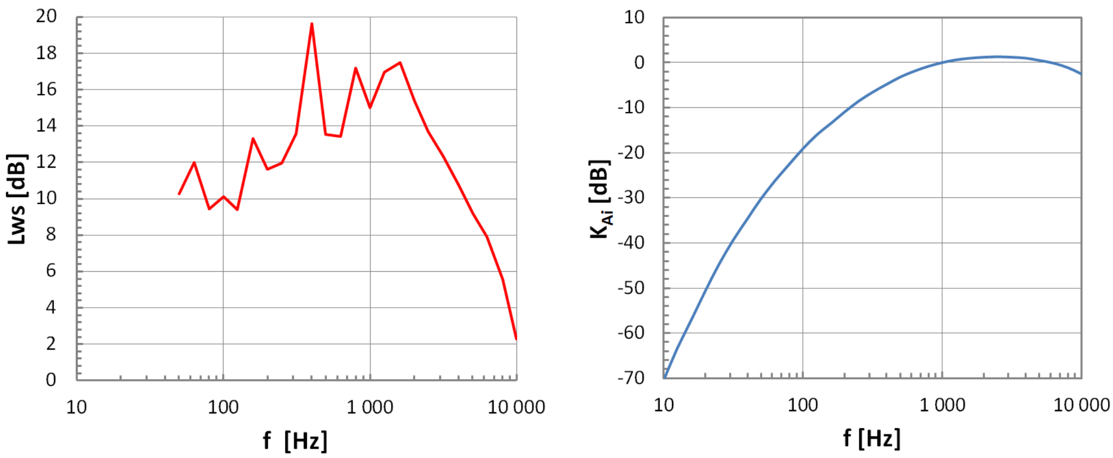


Fig. 4: Performance map of some axial-flow fans for power plants.

On the other hand, one has to deal with acoustic demands which are more and more challenging in every field of industry. The highest standards nowadays have the so called aero-acoustic wind tunnels, which are currently becoming common for every car factory.

An example of the specific sound power level distribution is shown on Fig. 5 – a). The maximum of the sound power occurs on the blade frequency, which in this case was 400Hz. Since the requirements are usually defined as A-sound power it is convenient to make the blade frequency as low as possible because the corrections of the acoustic filter A grow getting further from the frequency of 1000Hz (Fig. 5 – b), see Nový 2009 [7]).



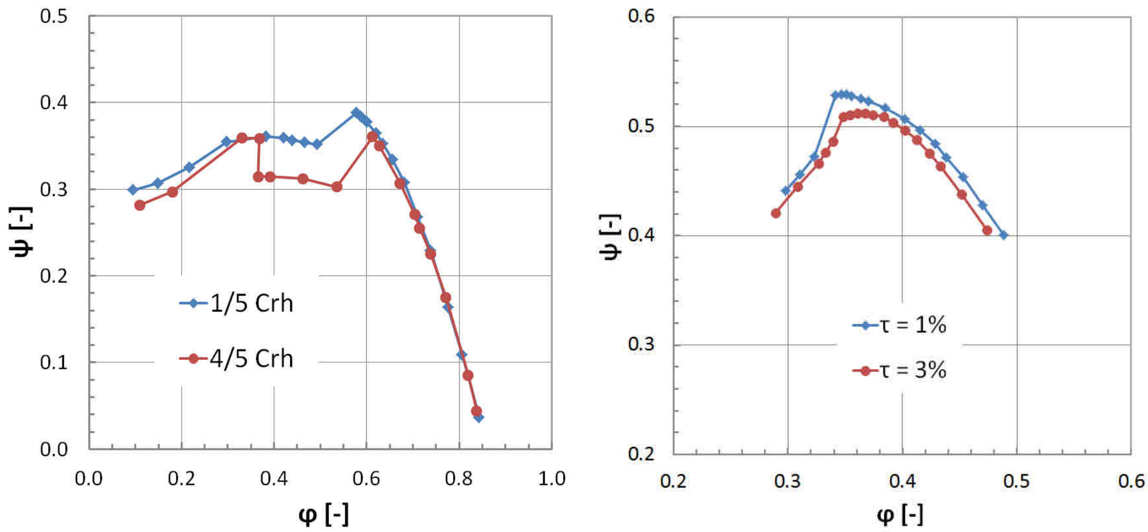
a) specific sound power level distribu- tion example

b) acoustic filter A corrections

Fig. 5: Acoustic parameters.

The blade frequency is purely a function of the rotational speed and of the rotor blades amount. That leads to machines, which can manage the desired aerodynamic parameters with low revolutions per minute in combination with smaller amount of blades. There is a certain way of manipulating the rotor blades amount. Yet, one must bear in mind that changing the amount of the rotor blades maintaining the blade row geometric similarity breaks the similarity of the wheel (Ušakov et al. 1962 [2], Eck 1952 [8]).

Direct and simple decrease of the sound power level lies in increasing the axial gap between rotor and stator blade rows, which in contrary has unfavourable effect on the fan stability as shown on Fig. 6 – a) for two different axial distances of one fifth and four fifths of the rotor blade chord measured at the hub (C_{rh}). The principle of the process from both aerodynamic as well as acoustic point of view lies obviously in the development of the profile losses, namely the wake and its interaction with the following stator blade row.



a) rotor – stator axial gap (courtesy of ISABE – see Li et al. 2005 [9])

b) tip gap (courtesy of ASME – see Inoue et al. 2004 [13])

Fig. 6: Gap influence on stall margin.

A much more sophisticated approach focused on the above mentioned principle offers the consideration of the swept blades. A nice review on sweep in axial compressor design has been given by Ji et al. 2005 [10]. However, the paper does not give much of an inspiration. Far more inspiring is the paper of Roy et al. 2005 [11], which gives a simple and efficient suggestion. The studied fan had a hub to tip ratio ν=0.5 and a decent diffusion factor distribution along the blade height that is 0.42 at the hub and 0.38 at the tip. Out of the four combinations of straight and swept rotor and stator blades, the configuration of the straight rotor and swept stator blades gave the most promising results from the acoustic point of view. The difference in noise level with respect to the straight rotor and stator blades at designed stagger angle λ was more than 10dB. From the physical point of view, this effect should not lower much the efficiency and hence particularises the further direction of attention with respect to the demands on manufacture. A very favourable effect on aero-acoustics has been shown also using forward skewed swept rotor blades (see Cai et al. 2004 [12]) although the expensive manufacturing process and uneasy usage in variable pitch fans greatly limits such a solution.

In contrary to the mixed-flow fans, the fans with axial-flow acceleration suffer from a particularly undesirable tip clearance flow, which is a significant source of aerodynamic losses as well as of noise. The tip gap is a necessary constructional condition especially in the case of explosion-proof fan design. The tip leakage flow also unfavourably influences the stall evolution process as shown on Fig. 6 – b) for two different clearance sizes, which demonstrates a typical distinction between the model laboratory condition and real on-site condition.

The simplest approach of controlling the tip leakage flow for improving the stall margin is a suitable casing treatment, such as introducing grooves (see e.g. Lu et al. 2005 [14]). However, the fans with variable pitch require a bit greater scope of handling such as e.g. air-separator (Nishioka et a. 2004 [15]) or endwall recirculation (Strazisar et al. 2004 [16]).

3. Outlet Geometry Impact

Last but not the least important part of the fan is the diffuser decreasing the mechanical energy loss due to a sudden flow expansion described by Borda-Carnot equation and the connected duct geometry.

The geometry of the diffuser has been under close analysis already for some decades in many ways and branches. The numerous experiences yielded to an optimum jaw angle in terms of friction losses and possible flow separation.

Should the fan be followed by an elbow, the separation zone created behind the diffuser must have an opportunity to close itself (Fig. 7 – hatched area denotes the so called cone of Borda vortices), otherwise the risk of instabilities propagation through the elbow impends. The resulted pressure penalty may than bring the system to oppressive situation with an insufficient power of the motor or worse with the fan falling in stall. Unfortunately, the situation on the site usually does not allow a straight duct long enough at the fan outlet. Therefore, the fan designer should also count with a reserve for this possibility during the design process and demand the complete information on the project.

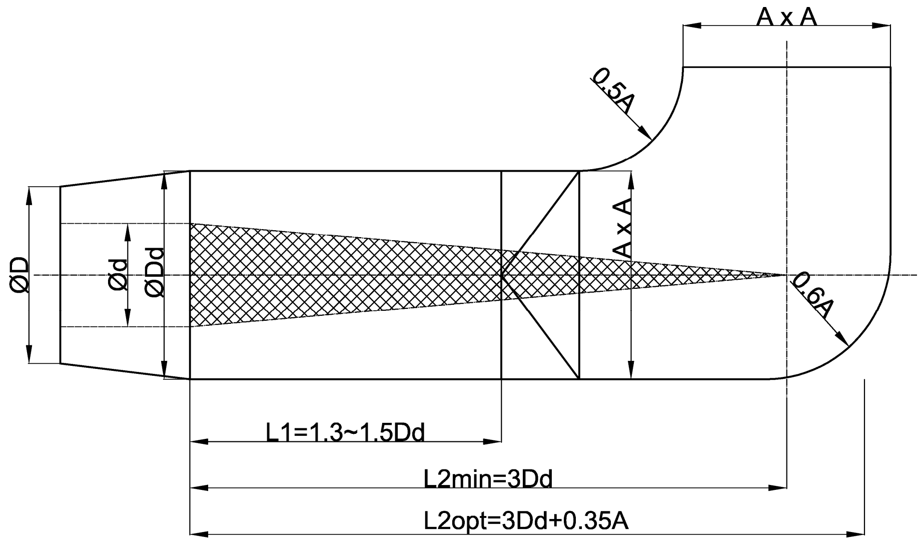


Fig. 7: Optimum outlet duct geometry.

In the case of axial fans driving a wind tunnel, the diffuser is mostly due to limited space usually substituted by a geometrically more complicated part, namely the tail-cone of the appropriate shape, such as ogival. The losses and sound power of such a solution may obviously be very advantageous.

4. Reynolds Number Impact

Since the efficiency of any turbomachine is influenced by friction, which depends on Reynolds number, a credible efficiency scaling method is needed. Yet, besides the resistance, the Reynolds number effects also the flow deviation in the blade row.

Applying the approach of Ackeret published by Mühleman 1948 [17] where appropriate has been a longstanding practice in our factory. This method has been found empirically on water turbine based on the idea of inefficiency being directly proportional to the friction factor λ, which according to the pipe flow analogy with fully turbulent flow in a duct with hydraulically smooth walls may be approximated by powered Reynolds number. On top, Ackeret considered in his method the inertia losses as well.

Generally, a similar method of Stephenson 1953 [19] was recommended for fans and gives a little more moderate results than the method of Ackeret (see Fig. 8). However, this method has based the evaluation of the Reynolds number influence on efficiency of a turbomachine on the von Karman approximation of the Darcy-Weisbach friction coefficient λ derived for a flat plate with a turbulent boundary layer.

Both above mentioned methods are empirical and quite limited. Nowadays, the friction factor can be quantified more precisely solving the Colebrook-White equation (Colebrook et al. 1937 [20]) using the approximation of Goudar-Sonnad (Goudar et al. 2008 [21]) for a full flowing circular pipe. Hence, the new physically based universal scaling method of Pelz et al. 2012 [22] meant a promising breakthrough, which takes into account also the influence of Reynolds number on the flow and pressure coefficients and shows the best validation against the experimental data so far. Nonetheless, this method tends to overestimate the effect of Reynolds number in the regions of negative gradient of the performance curve, which is obviously the most important part and makes this method in practise hazardous with respect to the allowed deviations from the agreed operating points.

Fig. 8 shows the comparison of the scaling methods considered for use in the design process. Apparently the most modest method is the Stephenson's and the most courageous the Pelz's one. For the time being, the oldest method of Ackeret presents a safe comprise.

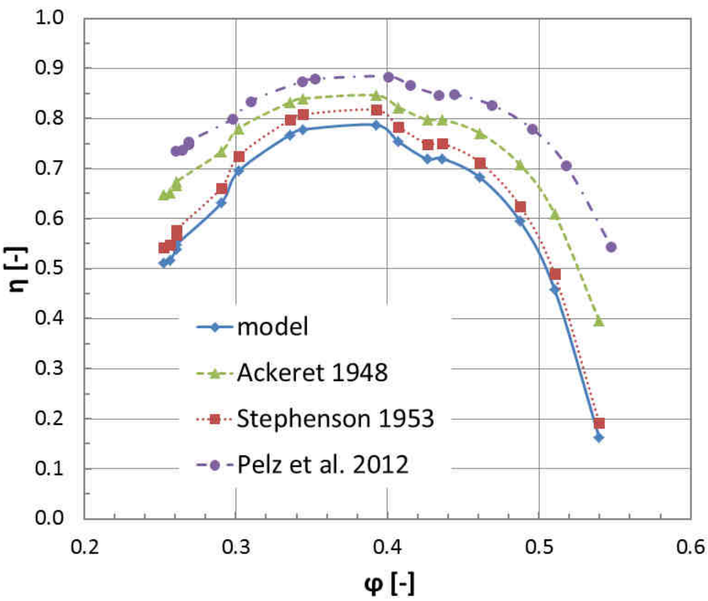


Fig. 8: Theoretical Re influence on fan aerodynamic performance.

After all, it must be pointed out that the application of the above mentioned methods is determined by the relative wall roughness and hence cannot be assumed for most of the industrial applications, since the Reynolds number of the models may already be in the autonomous zone independent on Reynolds number (see the Nikuradse also known as Moody chart on Fig. 9). Moreover, in most industrial cases the relative roughness increases with time as e.g. in the case of the induced draft fans, where the blade surfaces degrade due to the presence of the abrasive particles in the flue gas.

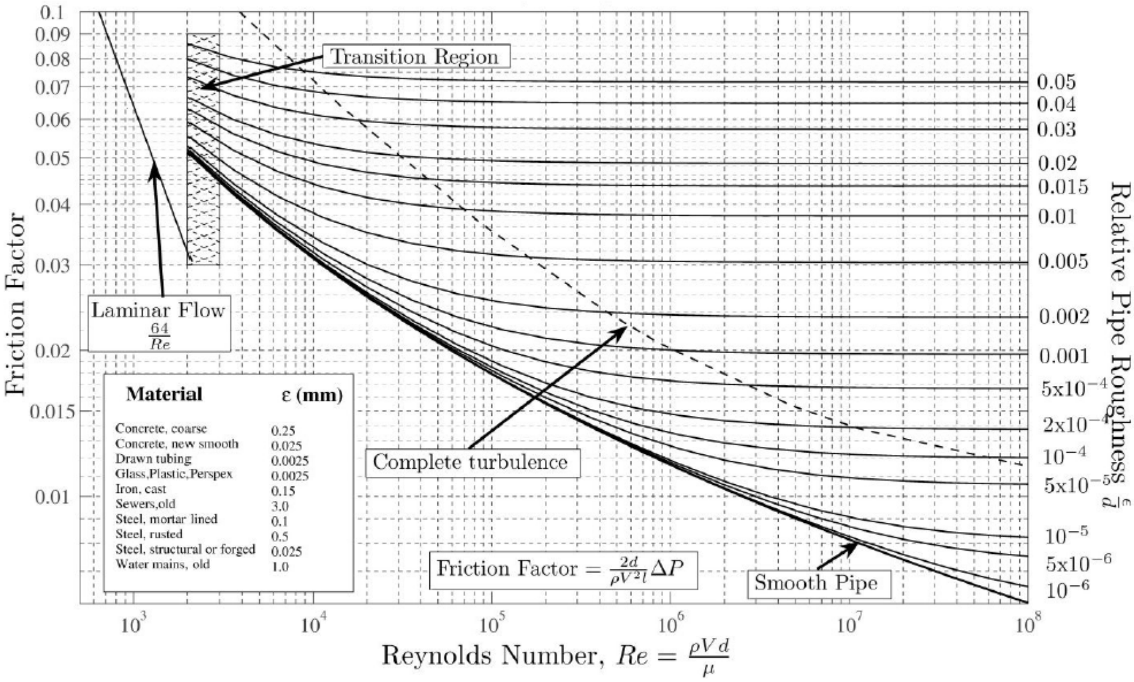


Fig. 9: Moody chart (Moody 1944 [18]).

Conclusions

The basic stationary parts of the fans produced in ZVVZ MACHINERY, a.s. have been discussed with the focus on their geometry impact on the aerodynamic and acoustic performance. The importance of the fan installation quality has been shown. The efficiency scaling methods have been discussed. The attempted feedback of the manufacturer in terms of experience with respect to research has been given.

List of Symbols

C	[m]	blade chord
D	[m]	rotor diameter/fan size
f	[Hz]	frequency
K_A	[dB]	acoustic filter A correction
L	[m]	length
L_W	[dB]	sound power level
Δp	[Pa]	fan total pressure drop
Q	[m^3/s]	fan volume flow

n	[rpm]	rotational speed
Re	[-]	Reynolds number
η	[-]	efficiency
λ	[°], [-]	stagger angle, friction coefficient
ν	[-]	hub to tip ratio
τ	[-]	tip clearance height to blade chord length at tip ratio
ϕ	[-]	flow coefficient
ψ	[-]	pressure coefficient

References

[1] GRÖNWOLDT-HESSE, H. Stall and Parallel Operation, 6th International Conference "Tunnel Safety and Ventilation" 2012, Graz, pp. 230-235, 2012.

[2] UŠAKOV, K. A., BRUSILOVSKIJ, I. V., BUŠEĽ, A. R. Aerodynamika osových ventilátorů a jejich konstrukční prvky (Aerodynamics of Axial Fans and their Constructional Features), SNTL, Prague, 1962.

[3] BOEHLE, M., CAGNA, M., ITTER, L Compressible Flow in Inlet Guide Vanes with Mechanical Flaps, Proceedings of ASME Turbo Expo 2004 – GT2004-53191, Vienna, June 14-17, 2004.

[4] CYRUS, V., CYRUS, J., PANEK, P. Design and Aerodynamic Performance of a High Pressure Axial Flow Fan, FAN 2012, Senlis (France), April 18-20, 2012.

[5] VAD, J., HORVÁTH, C. The Impact of the Vortex Design Method on the stall behavior of Axial Flow Fan and Compressor Rotors, Proceedings of ASME Turbo Expo 2008 – GT2008-50333, Berlin, June 9-13, 2008.

[6] SAVIC, S. M., MICHELI, M. A., BAUER, A. C. Redesign of a Multistage Axial Compressor for Heavy Duty Industrial Gas Turbine (GT11NMC), Proceedings of ASME Turbo Expo 2005 – GT2005-68315, Reno-Tahoe, June 6-9, 2005.

[7] NOVÝ, R. Hluk a chvění (Noise and vibrations), ČVUT v Praze, ISBN 978-80-01-04347-9, Praha, 2009.

[8] ECK, B. Ventilatoren (Fans), Springer-Verlag, ISBN 3642556507, Berlin, 1952.

[9] LI, CH., HU, J., WU, T., WANG, Y. Effect of Axial Spacing on Compressor Stability, Proceedings of the 17th International Symposium on Airbreathing Engines, ISABE-2005-1141, Munich, September 4-9, 2005.

[10] JI, L., CHEN, J. Review and Understanding on Sweep in Axial Compressor Design, Proceedings of ASME Turbo Expo 2005 – GT2005-68473, Reno-Tahoe, June 6-9, 2005.

[11] ROY, B. CHOUHAN, M. Aerodynamic Studies on Swept Blades for Axial Flow Fan/Compressor, Proceedings of the 17th International Symposium on Airbreathing Engines, ISABE-2005-1184, Munich, September 4-9, 2005.

[12] CAI, N., BENAISSA, A., ALLAN, W. D. Aerodynamic-Aeroacoustic Investigation of Rotating Stall in Conventional and Skewed Rotors, Proceedings of ASME Turbo Expo 2004 – GT2004-53958, Vienna, June 14-17, 2004.

[13] INOUE, M., KUROUMARU, M., YOSHIDA, S., MINAMI, T., YAMADA, K., FURUKAWA, M. Effect of Tip Clearance on Stall Evolution Process in a Low-Speed Axial Compressor Stage, Proceedings of ASME Turbo Expo 2004 – GT2004-53354, Vienna, June 14-17, 2004.

[14] LU, X., ZHU, J., CHU, W., WANG, R. The Effect of Stepped Tip Gap on Performance and Flowfield of a Subsonic Axial-Flow Compressor Rotor, Proceedings of ASME Turbo Expo 2005 – GT2005-69106, Reno-Tahoe, June 6-9, 2005.

[15] NISHIOKA, T., KURODA, S., KOZU, T. Improving Stall Margin by Using an Air-Separator for Variable-Pitch Axial-Flow Fan, Proceedings of ASME Turbo Expo 2004 – GT2004-53663, Vienna, June 14-17, 2004.

[16] STRAZISAR, A., BRIGHT, M. M., THORP, S., CULLEY, D. E., SUDER, K. L. Compressor Stall Control through Endwall Recirculation, Proceedings of ASME Turbo Expo 2004 – GT2004-54295, Vienna, June 14-17, 2004.

[17] MÜHLEMANN, E. Zur Aufwertung des Wirkungsgrads von Überdruck-Wasserturbinen (On Efficiency Scaling of Overpressure Water Turbines), Schweizerische Bauzeitung, Band 66, Heft 24, Switzerland, 1948.

[18] MOODY, L. F. Friction Factors for Pipe Flow, Transactions of ASME, Vol. 66m, No. 8, 671-684, 1944.

[19] STEPHENSON, J. M. Efficiency and Drag of an Axial-Flow Compressor Stage: The Connexion Between the Efficiency of a Stage, the Forces on the Blade Sections and the Profile Drag; the Relation Between Different Definitions of Efficiency, and the Effect of other Sources of Loss, Aircraft Engineering and Aerospace Technology, Vol. 25, Iss. 6, pp.158 – 160, June, 1953.

[20] COLEBROOK, C. F., WHITE, C. M. Experiments with Fluid Friction in Roughened Pipes, Proceedings of the Royal Society of London, Series A, Mathematical and Physical Sciences, Vol. 161 (906), 367-381, August 3, 1937.

[21] Goudar, C. T., Sonnad, J. R. Comparison of the Iterative Approximations of the Colebrook-White Equation, Hydrocarbon Processing, Vol. 87, No. 8: 79-83, 2008.

[22] PELZ, P. F., STONJEK, S., MATYSCHOK, B. The Influence of Reynolds Number and Roughness on the Efficiency of Axial and Centrifugal Fans – a Physically Based Scaling Method, FAN 2012, Senlis (France), April 18-20, 2012.

Acknowledgment

This paper has been written thanks to years of extensive research work of Prof. Václav Cyrus and thanks to the immense amount of experiences of the ZVVZ senior aerodynamicist Eng. Petr Polanský, who has been a kind leader in the learning time.

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