Technical Paper
(max. 8-10 pages including abstract)

Session: 26-2

Session Name: Turbo compressors – research 1

Influence of Compressibility on Fan Efficiency and Fan Efficiency Scaling

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Summary

The similarity of Reynolds and Mach number for model and full scale fan cannot be retained for the dimensioning of large industrial fans. Scaling is supposed to compensate these differences in efficiency determination. The present paper indicates the influence of compressibility, e. g. the Mach number on efficiency and efficiency scaling of fans. The research is focused on experimental investigations and the use of common scaling laws proposed by Ackeret [1] and Pelz/Stonjek [2]. The investigations were done at the laboratories of the Chair of Fluid Systems at the Technische Universität Darmstadt, Germany. A centrifugal fan with the specific speed $\sigma = 0.114$ was chosen because of its high pressure rise and high operating Mach number. As expected, the results show an increase in maximum efficiency by increasing Reynolds and Mach number up to $Ma < 0.3$. If the Reynolds and Mach number are increased further, the maximum efficiency stagnates and decreases for Mach numbers above $Ma > 0.4$. Both mentioned scaling laws consider the variation of Reynolds number, but none of them consider the effects of a change in Mach number. But a change in Mach number affects the friction losses, Carnot losses and incidence losses, which are discussed for the present fan as well.

1. Nomenclature

Latin Symbols

c	velocity
c_f	friction coefficient
D	diameter
f	correction function
h	enthalpy
k	roughness height
l	length
M	Mach number
Ma	circumferential Mach number
n	rotational speed
P	power
R	specific gas constant
Re	circumferential Mach number
s	gap width
T	temperature
u	circumferential speed
Y	specific work

Greek Symbols

β	incidence angle
γ	isentropic exponent
Δ	difference
ε	inefficiency
ζ	loss coefficient
η	efficiency
κ	scaling factor
λ	power coefficient
μ	dynamic viscosity
ν	kinematic viscosity
ϱ	density
σ	specific speed
φ	flow coefficient
ψ	pressure coefficient
Ω	angular velocity

Subscripts

+	dimensionless
1	inner
2	outer
c	compressible / critical
f	friction
i	incidence
l	loss
m	model
max	maximal
opt	optimum
s	shaft
stat	statistic
sys	systematic
t	total
turb	turbulent

2. Introduction

The need of efficiency scaling and the history of scaling, as well as the scaling method developed at the Chair of Fluid Systems, are presented in the first place. The article continues by describing the experimental setup, the investigated fans and the results indicating a down-scaling effect of the efficiency. The reason for this efficiency decrease is discussed with different loss models.

Why is efficiency scaling important?

The most important quantity for the characterization of turbomachinery are the pressure rise $\Delta p(\dot{V})$ or the pressure coefficient $\psi(\varphi)$. The efficiency η of a fan is another important parameter which is equivalent to its quality. The efficiency η is measured in test rigs with standardized procedures. But building up huge test rigs for large axial or centrifugal fans (rotor diameter $D_2 > 2\ \mathrm{m}$) is costly, time-consuming and sometimes impossible due to power limitations. A second application is the efficiency scaling of large fan series with different outer diameters D_2 and different rotational speeds n. Therefore scaling methods are used to generate evaluation data for immeasurable prototypes or to reduce experimental and numerical investigations.

Figure 1 shows the efficiency η plotted versus the flow coefficient φ indicating an up-scaling effect which was measured in the test facilities of the Chair of Fluid Systems at the Technische Universität Darmstadt [3]. The fan has a specific speed of $\sigma = 0.306$. Rising the rotational speed and thus increasing the circumferential Reynolds number Re leads to higher efficiencies. This is due to a decrease of friction coefficient with increasing Reynolds number known as up-scaling. The range of circumferential Mach number is $Ma = 0.08, \dots, 0.33$. However the flow Mach number is lower than the circumferential Mach number which allows the assumption of negligible low compressibility effects. For fans or compressors with higher pressure rise, such as compressors of turbochargers, an opposite effect is observed. The efficiency versus the flow coefficient is shown in Figure 2, [4]. The peak efficiency decreases with increasing the rotational speed, which corresponds to increasing the Reynolds and Mach numbers. The common scaling laws fail for the fans with high pressure rise as discussed subsequently. The change in Reynolds number and thus the change of friction losses is taken into account. Due to this effect, the predicted efficiencies could be too high.

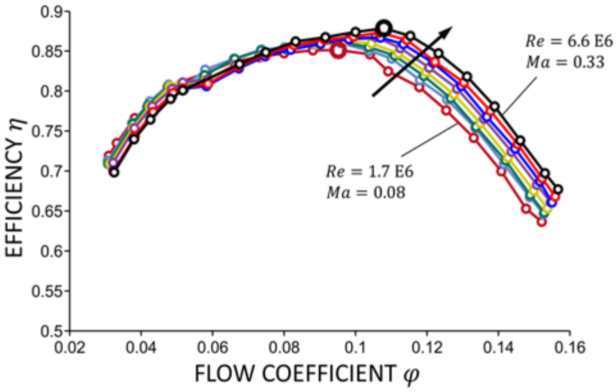


Figure 1: Efficiency of a medium pressure rise radial fan ($\sigma \approx 0.3$) with an up-scaling effect [3].

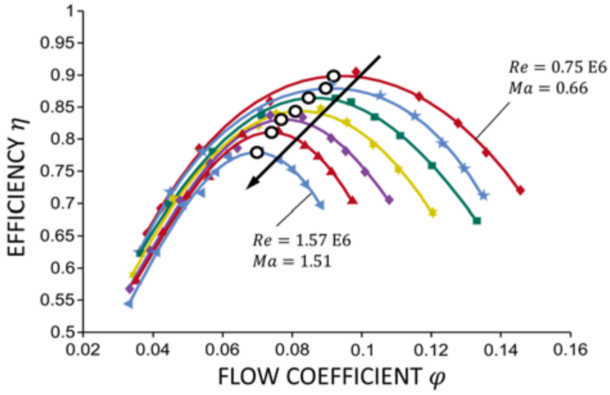


Figure 2: Efficiency of a centrifugal turbo charger compressor with a down-scaling effect [4].

The history and drawbacks of state of the art efficiency scaling methods

Efficiency scaling for turbomachines has a long history which goes back to 1925. The first scaling methods were based on empirical observations of turbines and only one geometrical parameter, the diameter of the impeller D_2, was considered (Moody [5], Staufer [6]). Later in 1947 Pfleiderer [7] introduced a physically based scaling law which includes the Reynolds number and thus the scalable friction losses. The problem with Pfleiderer's scaling method is the unrealistic asymptotic limit. For $Re \to \infty$ the efficiency $\eta \to 1$ will reach finally the value one. In 1948 Ackeret [1] improved this method by considering that half of the losses are scalable and the others are independent of the Reynolds number (equation (1)). This formula from 1948 is, until now, a common scaling method for fan efficiency. The necessary variables are the efficiency of the model η_m, the Reynolds numbers for the model Re_m and the full-scale fan Re,

$$\frac{1-\eta}{1-\eta_m} = 0.5 + 0.5\left(\frac{Re}{Re_m}\right)^{-0.2}. \qquad (1)$$

Further improvements of this ansatz were done e. g. by Wiesner [8] for radial compressors. But these improvements includes only the weighting factor for the scalable losses and the exponent of the Reynolds number ratio. This formula is used in the ISO 13348 for the conversion of measuring results of scaled test fans. But the Reynolds number shall not differ by more than $40\,\%$, [9]. A change of the Mach number is not discussed. The newest scaling method proposed by Pelz/Stonjek [2] is a physical based method, which includes the Reynolds number and has been validated for axial and centrifugal fans. It includes the effects of the surface roughness, the gap between the stationary inlet and the rotating impeller for the centrifugal fans and the tip clearance for the axial fans, as well as the operational point. But they have not considered the influence of the compressibility. Due to the physically based character, the effects of the compressibility can be included. The scaling method by Stonjek/Pelz [2] is based on the total derivative of the inefficiency $\varepsilon \coloneqq 1-\eta \coloneqq \zeta/\lambda$ which is

$$\frac{\mathrm{d}\varepsilon}{\varepsilon}=\frac{\mathrm{d}\zeta}{\zeta}-\frac{\mathrm{d}\lambda}{\lambda} \tag{2}$$

with the loss coefficient ζ and the power coefficient $\lambda \coloneqq 2\Delta h_t/u^2$ with the total enthalpy h_t. For low inefficiencies $(\mathcal{O}(\varepsilon)=0.1)$ and knowing the difference in Reynolds numbers of both operational points, equation (2) results in

$$\Delta\varepsilon=\varepsilon\frac{\Delta\zeta}{\zeta}-\varepsilon\frac{\Delta\lambda}{\lambda}=\varepsilon\frac{\Delta\zeta}{\zeta}-\varepsilon^2\frac{\Delta\lambda}{\zeta}\approx\varepsilon\frac{\Delta\zeta}{\zeta}. \tag{3}$$

The use of Froude's Hypothese allows to separate the losses into independent parts [10]. The change in loss coefficient $\Delta\zeta=\Delta\zeta_f(Re)+\Delta\zeta_c(Ma)$ is a function of the Reynolds and Mach number. The Reynolds number dependent part is investigated by several authors ([2], [3] and [10] for centrifugal fans and [11], [12] and [13] for axial fans) and is responsible for a rise in efficiency, if the Reynolds number is increasing. A brief description of the efficiency scaling due to Reynolds number effects for axial and centrifugal fans can be found in [2]. The second part represents the down-scaling part, which allows to take the compressibility of the flow into account. The compressibility depends on the flow Mach number Ma and has, once again, currently not been included in scaling laws for fans.

Dimension analysis

The total efficiency $\eta=\eta(\dot V,\Omega,D_2,\nu,\varrho,a,k,s,shape)$ and the total specific work $Y_t=Y_t(\dot V,\Omega,D_2,\nu,\varrho,a,k,s,shape)$ depends on the following physical quantities:

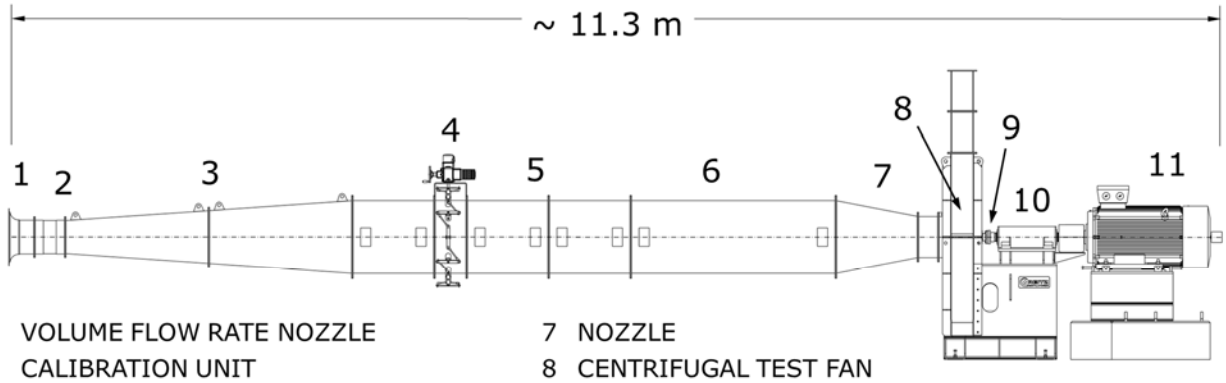
- volume flow rate $\dot V$,
- rotational speed $\Omega=2\pi n$,
- machine size represented by impeller diameter D,
- kinematic viscosity $\nu=\mu/\varrho$ (dynamic viscosity μ),
- density ϱ,
- the compressibility of the gas is measured by the speed of sound $a=\sqrt{\gamma RT}$ (the isentropic exponent γ, the ideal gas constant R and the static temperature T),
- the absolute surface roughness height k,
- the absolute gap width s between shroud and inlet for centrifugal fans and tip clearance for axial fans, and
- the shape of the fan which is described by i different numbers of geometrical parameters l_i

By using a dimensional analysis, the number of independent variables can be reduced to the following dimensionless products:

- flow coefficient $\varphi \coloneqq 4\dot V/(u_2D_2^2\pi)$ (" $\coloneqq$ " symbolizes a definition) with the circumferential speed $u_2=\Omega D_2/2$,
- circumferential Reynolds number $Re \coloneqq uD/\nu$,
- circumferential Mach number $Ma \coloneqq u/a$ (and the Mach number of the flow is $M \coloneqq c/a$),
- relative roughness $k_+ \coloneqq k/D$,
- relative gap width or tip clearance $s_+ \coloneqq s/D$,
- efficiency $\eta \coloneqq 1-\varepsilon=1-P_l/P_s$ with the inefficiency ε, the applied shaft power P_s and the dissipated power P_l, and
- pressure coefficient $\psi \coloneqq 2Y_t/u_2^2$ with the total specific work Y_t.

This yields to $\eta=\eta(\varphi,Re,Ma,k_+,s_+,shape)$, $\psi=\psi(\varphi,Re,Ma,k_+,s_+,shape)$ and the power coefficient $\lambda := \varphi\psi/\eta$. The flow coefficient φ and the pressure coefficient ψ can be replaced by the specific speed $\sigma := \varphi^{1/2}\psi^{-3/4}$ and the specific diameter $\delta := \varphi^{-1/2}\psi^{1/4}$. Now the term can be expressed as $\eta(\sigma, Re, Ma, k_+, s_+) = \eta(type, size, quality)$. The specific speed σ determines the type, the size is given by the Mach number Ma and Reynolds number Re. The Mach number is the ration of an characteristic length of the machine – in this case the outer diameter D_2 – and the length which is $2a/\Omega$. The quality is determined by the relative roughness k_+ and the relative gap s_+.

3. Test Rig



1 VOLUME FLOW RATE NOZZLE
2 CALIBRATION UNIT
3 DIFFUSOR
4 THROTTLE
5 FLOW STRAIGHTENER
6 MEASUREMENT POINT BEFORE THE FAN
7 NOZZLE
8 CENTRIFUGAL TEST FAN
9 TORQUEMETER
10 BEARING
11 ELECTRIC ENGINE

Figure 3: Centrifugal fan test rig.

The results presented in the following sections are measured on two test rigs which fulfill the DIN 24163 guidelines [14]. These test rigs (Figure 3) include a volume flow rate nozzle (1). The nozzle is calibrated to enhance the accuracy and to enlarge the measuring range. The calibration was done for a compressible flow in the flow rate nozzle due to velocities up to $100\ \mathrm{m}/s$ corresponding to the flow Mach number $M \approx 0.3$. To avoid boundary layer and compressibility effects the nozzle is calibrated with an array of total pressure probes (2) across the flow area[1]. A flow straightener (5) is placed after the throttle (4) to lower the swirl and flow inhomogeneities due to the throttle. At the position (6) the static pressure and the total temperature is measured and thus the total pressure at the fan inlet can be calculated. The torquemeter (9) is located between the investigated fan (8) with the impeller and the bearings (10). This setup allows the measurement of the aerodynamic torque which is transmitted to the fluid. The torquemeter has two different measuring ranges to ensure a satisfactory accuracy at high and low rotational speeds.

The evaluation of the DIN 24163 is extended for a compressible flow as it is explained in the VDI 2044, which is necessary because of predicted Mach numbers $Ma > 0.5$ and a pressure rise of $\Delta p_t \approx 30\ 000\ \mathrm{Pa}$.

Two scaled test rigs were used to run experiments at the same Mach number and different Reynolds numbers. The large test rig is shown in Figure 3. The small scale test rig has the same setup and the scaled size of 1:4. All measurements are performed at common ambient conditions.

[1] For the calibration the comb probe measuring points are located at the Log-Tchebycheff points which is a recommended method for flow measurements in cylindrical pipes, [20].

4. Investigated Fans

Concerning the compressibility effects, two scaled centrifugal fans (see Figure 4) with high Mach numbers and high pressure rises are considered. The fan characteristics are listed in Table 1.

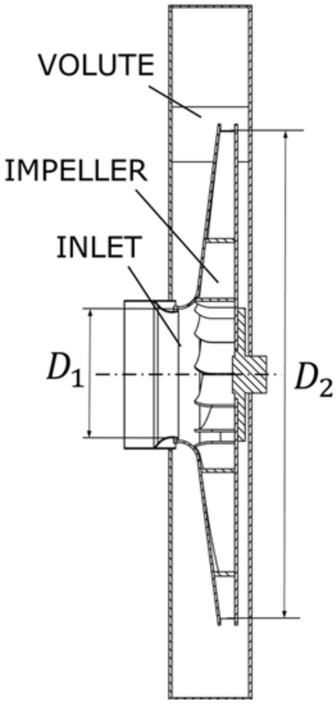


Figure 4: Investigated fan.

Table 1: Overview of the fan geometry and characteristics.

DETAIL	SYMBOL	UNIT	FANS	
			small scale	full scale
scaling factor	κ	1	1:4	1:1
specific speed	σ	1	0.114	
impeller outlet diameter	D_2	mm	331	1324
impeller inlet diameter	D_1	mm	88.75	355
rotational speed	n	1/min	2000 … 7500	500 … 3000
Reynolds number	Re	1	1.1 … 2.7 E6	2.9… 13.1 E6
Mach number	Ma	1	0.1 … 0.37	0.1 … 0.55

5. Results

The results of the full scale fan are discussed in detail and afterwards the most important results of the small scale fan are presented. The measurement errors are illustrated for selected plots.

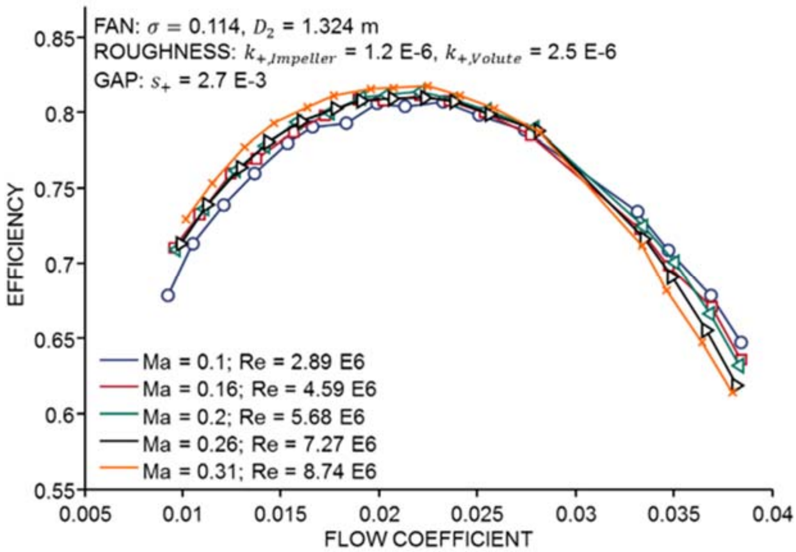


Figure 5: Efficiency characteristics of the full scale fan at low Mach numbers.

The total measurement error $\delta x = \sqrt{\left(\delta x_{sys}\right)^2 + (\delta x_{stat})^2}$ consists of the systematic error δx_{sys} and the statistic error δ_{stat}. Figures 5 - 7 present the results of the full scale fan which has a diameter of $D_2 = 1.324\ \mathrm{m}$.

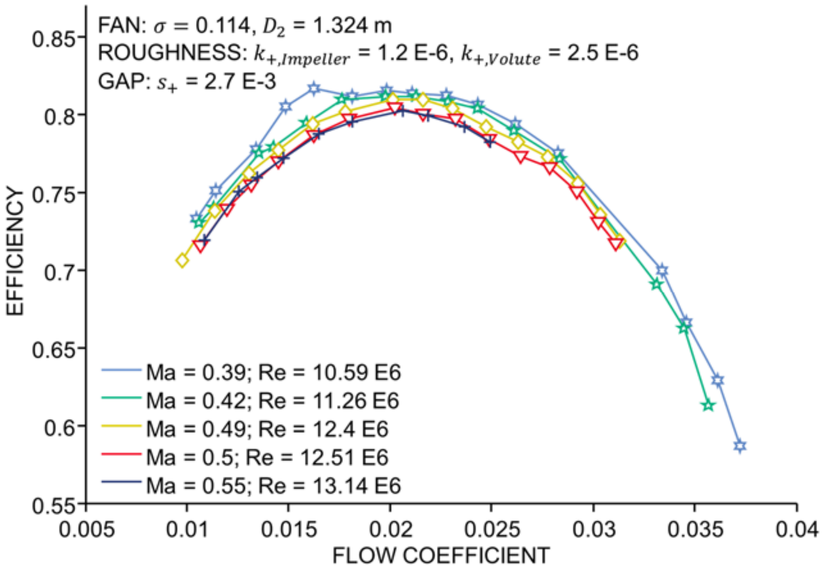


Figure 6: Efficiency characteristics of the full scale fan at high Mach numbers.

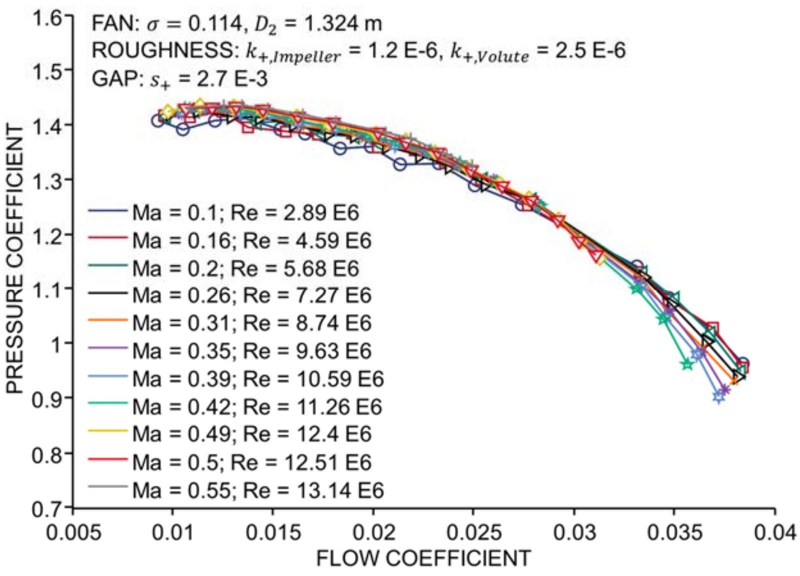


Figure 7: Pressure coefficient of the full scale fan for all rotational speeds.

Figure 5 shows the efficiency plotted versus the flow coefficient at low Mach numbers $Ma \leq 0.31$. The higher the Reynolds number the higher is the peak efficiency. In this range of Mach and Reynolds numbers the up-scaling effect due to lower friction losses is dominant. If the increase in Mach number has an influence on the efficiency, cannot be determined for this Mach number range yet. At overload conditions the opposite behavior was measured and the efficiency decreases while the rotational speed is increased. But this part is for real applications less important. Figure 6 shows decreasing best efficiency points while the Mach number increases from $Ma = 0.39$ to 0.55 and the Reynolds number from $Re = 10.27$ to 13.14 E6. This down-scaling effect shows up at Mach numbers about $Ma > 0.4$ and was not expected. At high overload conditions the test rig reaches the power and torque limit at a Mach number of $Ma = 0.42$ which is the green curve in Figure 6. All efficiency curves with higher rotational speeds have to start at a lower flow coefficient. All measurement data of the pressure coefficient are presented in Figure 7. The pressure coefficient rises continuously with increasing Mach number near peak efficiency. At overload conditions ($\varphi > 0.29$) a decrease in pressure coefficient is measurable due to high flow Mach numbers.

The use of the most common scaling law of Ackeret and the scaling law of Pelz/Stonjek (see Figure 8 and Figure 9) shows the need of an extension of the scaling laws. In both figures the peak efficiency is plotted against the Mach and Reynolds number. Due to the temperature variation the Mach and Reynolds number rise independently. Figure 8 shows the results for the full scale fan. An up-scaling effect is detected for low Mach numbers, which changes to a down-scaling effect for Mach numbers about $Ma > 0.4$. The errorbars clarify that under no circumstances a continuous efficiency up-scaling is possible. Ackeret's law predicts an efficiency rise of $\Delta\eta \approx 3\,\%$ and the scaling method from Pelz/Stonjek predicts almost $\Delta\eta \approx 5\,\%$. Both methods overpredict the efficiency. The results of the small scale fan in Figure 9 confirm this tendency. A decrease in efficiency cannot be detected for this fan due to power restrictions which allows measurements only up to $Ma \approx 0.37$.

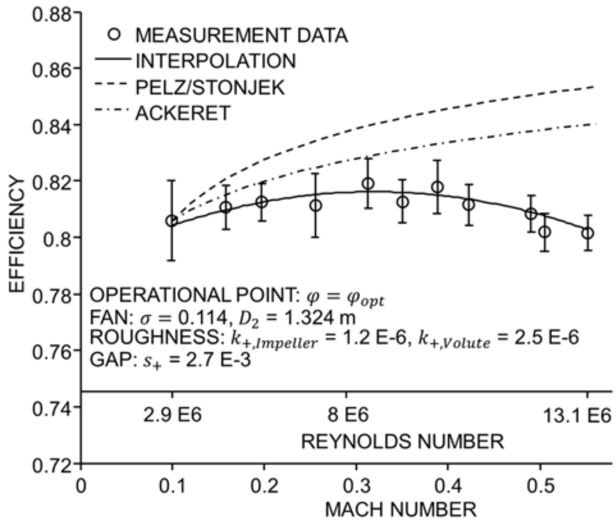


Figure 8: Total efficiency at the best efficiency point for the <u>full scale fan</u> with scaling laws based on the measurement point at the lowest Mach and Reynolds numbers.

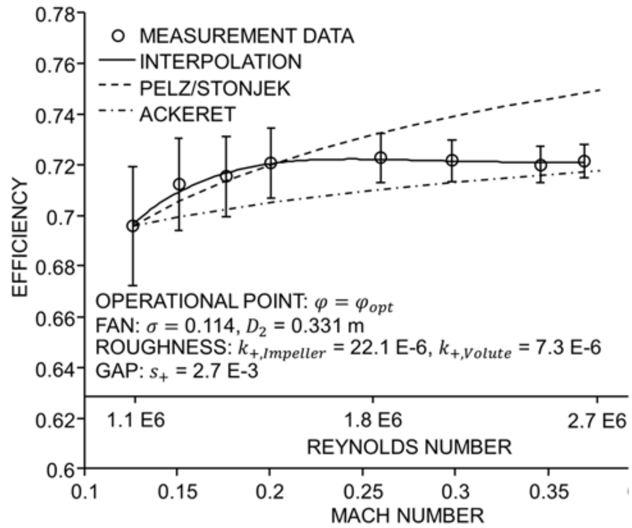


Figure 9: Total efficiency at the best efficiency point for the <u>small scale fan</u> with scaling laws based on the measurement point at the lowest Mach and Reynolds numbers.

6. Discussion

Experimental investigations are performed for one type of centrifugal fans ($\sigma = 0.114$) with two different scaled models at different rotational speeds and thus at different Mach and Reynolds numbers. For these kinds of fans, with low specific speeds and high Mach numbers, the scaling laws predict a continuous increase in efficiency. But according to the experimental measurements, the maximum efficiency rises for $Ma < 0.3$, then it stagnates and finally the peak efficiency decreases for $Ma > 0.4$. Ackeret's method seems to work better but in many cases (e. g. [2]) it under-estimates the scaled efficiency. Consequently the under-estimation matches better to the efficiency decrease. Another important point is the critical Reynolds number $Re_c = 137.5\,l/k$ which depends on a characteristic length l, which is the length of the flow channel, and the absolute surface roughness k [13]. For the full scale fan $Re_c \approx 37\,\mathrm{E6}$ which is about 3 times higher than the highest Reynolds number ($Re_{max} \approx 13\,\mathrm{E6}$). The critical Reynolds number is not reached and hence an up-scaling effect due to an increase in Reynolds number is still possible. That means that another effect becomes dominant which we may call *Mach number effect* because it depends on the compressibility of the flow.

What is the effect of the Mach number on the efficiency?

If the efficiency changes the losses have to change as well. An overview of loss models for centrifugal compressors was published by Denton [15] and Oh [16]. The most important losses for both investigated fans are:

- friction losses over the surfaces,
- the leakage flow through the gap between the impeller and the nozzle,
- incidence losses at the inlet of the rotor or impeller and
- Carnot-losses.

The loss due to the gap between nozzle and impeller is neglected because in this measurement campaign the relative gap was constant and the rotational speed and volume flow rate is changed. That is the reason why we will focus on the other three loss sources. But for completeness, a model for the gap between the impeller and the nozzle for centrifugal fans and the tip clearance for axial fans for small Mach numbers was published by Pelz and Stonjek, [10]. Friction losses are the scalable losses which depend on the Reynolds number and the surface roughness. This loss was investigated for several machine types and operational conditions, [2] and [13]. A Mach number dependency exists but it is rather small in comparison to friction losses and their changes. Krasnov [17] listed many different equation for all kinds of flow conditions on friction losses. For example the correction function for the friction coefficient $f_{f,turb}$ for a turbulent boundary layer is described by

$$f_{f,turb} = \frac{c_{f,turb}(M)}{c_{f,turb}(0)} = (1 + 0.12\, M^2)^{-\frac{1}{2}}. \tag{4}$$

At a flow Mach number $M = 0.4$ the ratio of friction coefficients $f_{f,turb} \approx 1.01$ which means that the difference is $\sim 1\%$. The incidence loss is a function of the incidence angle β and the flow Mach number M. The derivation and the application for turbomachinery for subsonic operation was published by Saul and Pelz, [18]. The interpolated correction function with the incidence β in ° is

$$f_i = \frac{\zeta_i(M,\beta)}{\zeta_i(M=0,\beta)} \approx 1 + M^2 + (4.5 - 2.5\,\beta + 23\,\beta^2)\, M^4. \tag{5}$$

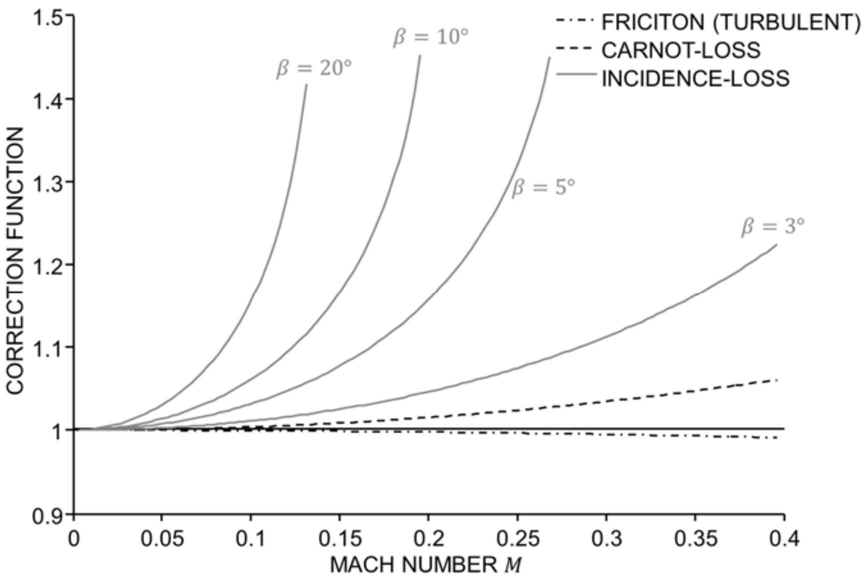


Figure 10: Correction functions for the Carnot, incidence and friction loss as a function of the flow Mach number M

The Carnot-loss for compressible flows can be found in Rist's book "Dynamik realer Gase", [19]. Both models are too complex to explain them in detail. Figure 10 shows all correction functions for the three loss models plotted versus the Mach number M. For the incidence loss the incidence angle β is an additional parameter. The area ratio for the Carnot loss is given by the geometry of the fan. It seems that the incidence angle is dominant but this loss occurs at the inlet of the impeller and the Mach number of the flow $M_1 < 0.2$. The Mach number at the outlet of the impeller is $M_2 \approx 2.5\, M_1$. At the best efficiency point the incidence angle is negligible. The friction loss decreases with rising Mach numbers but this effect is rather low in comparison to the Carnot loss which occurs at higher Mach numbers. This comparison clarifies the dependency of the Mach number and the type of loss. With a combination of all losses the down-scaling effect should be describable.

7. Conclusion

Experimental investigations of a high pressure centrifugal fan with low specific speed ($\sigma = 0.114$) show an unexpected behavior. The efficiency increase for increasing rotational speed is lower than what the common scaling laws predict and at circumferential Mach numbers $Ma > 0.4$ the efficiency decreases. This type of fan shows a very similar behavior like turbo charger compressors. The reason is the high Mach number i. e. the compressibility of the flow inside the fan, which produces additional losses. For Mach numbers $Ma > 0.4$ the Mach number effect is getting dominant and the up-scaling effect due to higher Reynolds numbers and hence lower friction losses is of minor importance. This work demonstrates the limits of Reynolds number based scaling laws (Ackeret's and Pelz/Stonjek's scaling law) which do not consider compressible effects. In standards, e. g. ISO 13348, Ackeret's law is recommended but it is only valid in a small range of Reynolds numbers. A change in Mach number is not included or discussed. The ISO 5801 gives conversion rules but only for constant Reynolds numbers.

Acknowledgement

The authors would like to thank the Arbeitsgemeinschaft industrieller Forschungsvereinigungen 'Otto von Guericke' e. V. (AiF), the Bundesministerium für Wirtschaft und Technologie (BMWi) and the Forschungsvereinigung für Luft- und Trocknungstechnik (FLT) e. V. whose support made this work possible.

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