



FAN MODEL TEST AT VARYING AMBIENT PRESSURE: EFFICIENT PRODUCT VALIDATION AT FULL SCALE REYNOLDS AND MACH NUMBER

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SUMMARY

The paper discusses the advantages of fan performance and efficiency measurements on standardized test rigs in a pressure vessel. The resulting ability to control the ambient pressure allows a decoupling of the Mach and Reynolds number and extends the achievable Reynolds number range. Thus, Mach number effects (the compressibility of the flow) and Reynolds number effects (the change of friction losses) are investigated independent of each other. The design of the pressurized test rig and the results are presented as well as the validation of scaling methods recently developed at Technische Universität Darmstadt. The here introduced concept of a pressurized test rig may serve an elegant approach to product validation in testing small-scaled models at full scale Reynolds number.

INTRODUCTION

Acceptance tests for large turbomachines, like fans deployed in wind tunnels, mines or power plants with typical impeller diameters $D_o > 2$ m, are carried out on standardized model test rigs due to better accessibility and lower measuring uncertainty. Fan models are downscaled to a good handling size to lower efforts and manufacturing and operating costs [1]. Today, rapid prototyped models with impeller diameters up to $D_o \approx 0.4$ m become more and more attractive for testing purposes due to low manufacturing times and high quality models.

The efficiency η depends on the type, dimensionless size, quality and operating point

$$\eta = \eta(\text{TYPE}, \text{DIMENSIONLESS SIZE}, \text{QUALITY}, \text{OPERATING POINT}) \quad (1)$$

and can be described by independent dimensionless products: The type of the fan is determined by the specific speed σ . The Reynolds number Re and the Mach number Ma characterize the dimensionless size. The quality of the fan is measured by the relative roughness k_+ and relative gap s_+ . The flow coefficient φ describes the operating point.

For non-similarity, the efficiency of the model, indicating by a prime (') and prototype differs $\eta' \neq \eta$ and the efficiency change $\Delta\eta = \eta - \eta'$ has to be determined. As the following section explains in detail, in most cases, full similarity cannot be achieved and so-called *scaling methods* determine the efficiency change $\Delta\eta$ [2, 3]. To be considered reliable, scaling methods must be physically based. It is deemed valuable, if the uncertainty of the scaling $\delta(\Delta\eta)$ plus the model measurement uncertainty $\delta\eta'$ is smaller than the uncertainty of an in situ measurement of the prototype $\delta\eta$

$$\delta\eta > \delta\eta' + \delta(\Delta\eta). \quad (2)$$

To be universally applicable, an efficiency scaling method for fans must be valid in a wide range of specific speeds σ , which includes axial and centrifugal fans.

A decrease in scaled efficiency uncertainty $\delta(\Delta\eta)$ can be reached (i) by increasing similarity between model and prototype or (ii) by more reliable scaling methods.

- (i) The scaling uncertainty $\delta(\Delta\eta)$ decreases for increasing degree of similarity, i.e. the ratio of Reynolds numbers Re/Re' or other ratios of independent dimensionless products decreases.
- (ii) Reliable scaling methods shall focus on the major loss sources: friction, incidence and Carnot losses [4]. It is well known that friction losses depend on Reynolds number [5, 6, 7], while incidence and Carnot losses depend on Mach number [8, 9]. For generic studies, models based on analytical and empirical results can determine these losses. However, the applicability of generic models to real world turbomachines is problematic. Mach and Reynolds number are proportional to the rotational speed n . Whilst an increasing Reynolds number Re causes lower friction losses [5], an increasing Mach number Ma causes higher incidence [8] and Carnot losses [9]. Both effects work against each other. To validate and improve the Darmstadt scaling method [4, 10], it is mandatory to investigate the effects of Mach and Reynolds number independent of each other.

This paper describes how both approaches are realized by manipulating the ambient pressure, which is achieved by a fan test rig setup inside a pressurized vessel. The independent variation of Mach and Reynolds number and the enlarged Reynolds number range is used to validated common scaling methods and improve Darmstadt scaling method.

The paper is organized as follows: in the following section, similarity is recaptured, followed by the discussion of decoupled Mach and Reynolds number. The subsequent sections present and discuss the experimental setup and results of the investigations in a pressurized vessel. The paper closes with a summary and a conclusion.

SIMILARITY THEORY

The small-scale fan model and the full-scale fan prototype are characterized by two dependent dimensionless products, i.e. pressure coefficient $\psi := 2Y/u_o^2 = \psi(\sigma, Re, Ma, k_+, s_+, \varphi)$ and efficiency $\eta := Y\dot{m}/P_S = \eta(\sigma, Re, Ma, k_+, s_+, \varphi)$, where Y denotes the specific work, u_o the circumferential velocity, \dot{m} the mass flow rate and P_S the shaft power. The dependent dimensionless products corresponding to model and prototype remain the same, only if all independent dimensionless products remain unchanged. In this case, model and prototype are similar. The independent dimensionless products are summarized below, subscript 1 denoting the fan inlet and subscript o denotes the outer part at the casing.

- specific speed $\sigma := \varphi^{1/2} \psi^{-3/4}$,
- Reynolds number $Re := D_o u_o \rho_1 / \mu_1$ (dynamic viscosity μ_1 , density ρ_1),
- Mach number $Ma := u_o / a_1$ (speed of sound $a_1 = \sqrt{\gamma R T_1}$, isentropic exponent γ , gas constant R , absolute temperature T_1),
- relative roughness $k_+ := k / D_o$ (absolute roughness k),

- relative gap $s_+ := s/D_o$ (absolute gap s) and
- flow coefficient $\varphi := 4\dot{V}_1/(\pi D_o^2 u_o)$ (volume flow rate \dot{V}_1).

The geometrical similarity is preserved, if all geometric measures of the machine are scaled with the same scaling factor $\kappa := D'_o/D_o$, being the ratio of the impeller diameter of the model to the prototype. This includes the roughness $k' = \kappa k$ and the gap $s' = \kappa s$. Besides the geometrical similarity, a full physical similarity (hence $\eta = \eta'$, $\psi = \psi'$) is only reached, if $Re = Re'$, $Ma = Ma'$ and $\varphi = \varphi'$ is also ensured. For practical reasons, the geometrical similarity is never fully achieved. The absolute surface roughness k and gap s are often not scalable [1], due to the manufacturing process. Furthermore, the Reynolds number Re cannot be preserved because the power consumption of model measurements would increase unphysically high [11]. The mentioned non-similarity results in different efficiencies of model and prototype $\eta' \neq \eta$. Therefore, there is a need to describe the efficiency change $\Delta\eta = \eta - \eta'$.

DECOUPLING OF MACH AND REYNOLDS NUMBER

The Mach number is defined as

$$Ma := \frac{u_o}{a_1} = \frac{u_o}{\sqrt{\gamma RT_1}} \quad (3)$$

and the Reynolds number as

$$Re := \frac{D_o u_o}{\nu_1} = \frac{D_o u_o \rho_1}{\mu_1}, \quad (4)$$

with the kinematic viscosity $\nu_1 = \mu_1/\rho_1$. The combination of eq. (3) and eq. (4) yields

$$Re = Ma \frac{D_o}{\mu_1} p_1 \sqrt{\frac{\gamma}{RT_1}}. \quad (5)$$

From eq. (5) we learn that Re and Ma are linear dependent. I.e. changing the Reynolds number is changing the Mach number as well. Re and Ma can only be changed independently, if (i) the temperature T_1 is changed or the pressure p_1 . Another gas (ii) has other properties, like the dynamic viscosity μ_1 , the isentropic exponent γ or the ideal gas constant R . A gas change requires an enclosed volume, like a vessel.

The change of ambient pressure, combined with another gas, is an elegant concept to investigate fan characteristics with decoupled Mach and Reynolds number. For an increasing ratio of Ma/Re , the Knudsen number increases. The Knudsen number [12] is a measure of the free path and is defined as

$$Kn := \frac{\bar{l}_1}{D_o}, \quad (6)$$

with the characteristic length of the fan D_o and the molecular mean free path length \bar{l}_1 , which is

$$\bar{l}_1 = l_1 \sqrt{\frac{\pi}{4}}. \quad (7)$$

According to Jousten [12], the equivalent free path length l_1 is

$$l_1 = \frac{\mu_1 \sqrt{2RT_1}}{p_1}. \quad (8)$$

The combination of equations (5) to (8) yields

$$Re = \frac{Ma}{Kn} \sqrt{\gamma \frac{\pi}{2}}. \quad (9)$$

As long as the Knudsen number remains constant $Kn = \text{const.}$, Reynolds and Mach number are proportional $Re \propto Ma$ (eq. (9)).

For the usage of a prototype with different rotational speeds, the Knudsen number is constant but the more common scaling from a model to prototype decreases the Knudsen number because the impeller diameter increases $D_o > D'_o$. This change is simulated with an increase of ambient pressure of a fan model test rig (eq. (8)). An increase in pressure p decreases the equivalent free path length l_1 and the Knudsen number $Kn \propto p_1^{-1}$.

These investigations are common for airplane engines in altitude test rigs, where flight conditions in height are simulated by a decrease of ambient pressure but for fan investigations, an increase in pressure is advantageous to reach higher Reynolds numbers. For a rotary pump, Rotzoll [13] publishes measurement data in a large range of Reynolds numbers. He achieves Reynolds numbers from $Re = 5.5 \text{ E}4$ to $1.74 \text{ E}7$ due to a variation of ambient temperature, change of the fluid and rotational speed. Therefore, this paper focus on experimental investigations on a standardized test rig in an atmosphere of increasing ambient pressures to change systematically the Knudsen number.

EXPERIMENTEL SETUP

Figure 1 shows the experimental setup for the measurements in a pressurized atmosphere. The main component is the pressure vessel, included in a high-pressure loop. The loop consists a heat exchanger and an auxiliary fan, which generates a cooling flow to keep the temperature constant. The cooling flow is controlled by a volume flow rate nozzle. The gas is nitrogen (N_2), which is stored in a separate high-pressure gas reservoir. Temperature and pressure are measured continuously to ensure constant conditions during the measurements.

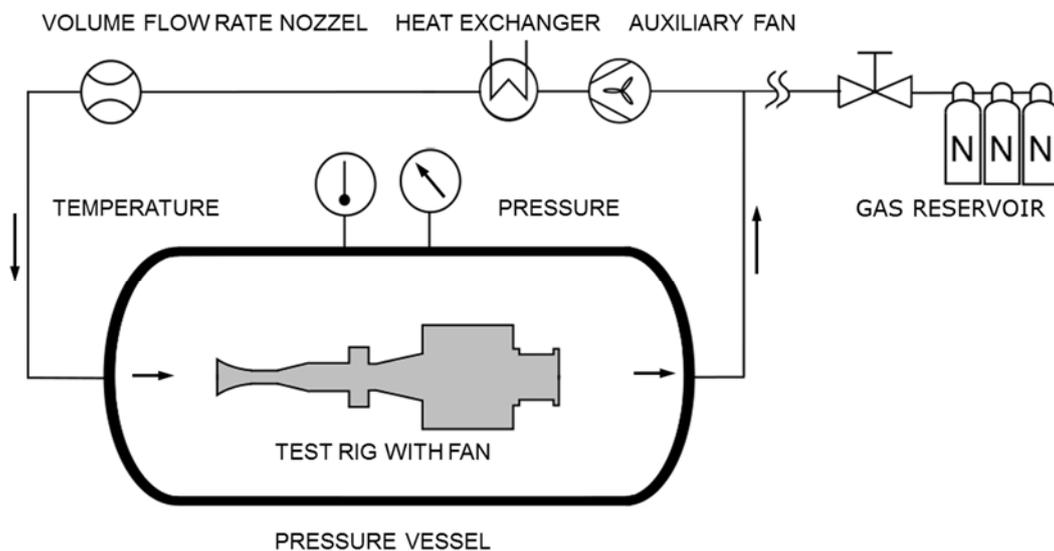


Figure 1: Experimental setup with high-pressure loop and fan test rig designed by TU Darmstadt.

The pressure vessel provides enough room for small-scaled fan test rigs (Figure 2). The vessel is more than 6 m long and the diameter of 1.8 m is necessary due to the radial outflow of centrifugal fans. For the installation, a closure lid is opened, which is not displayed in Figure 2. The electric power input as well as the measurement and control signal wires are connected to one of the three cable passages at the backside.

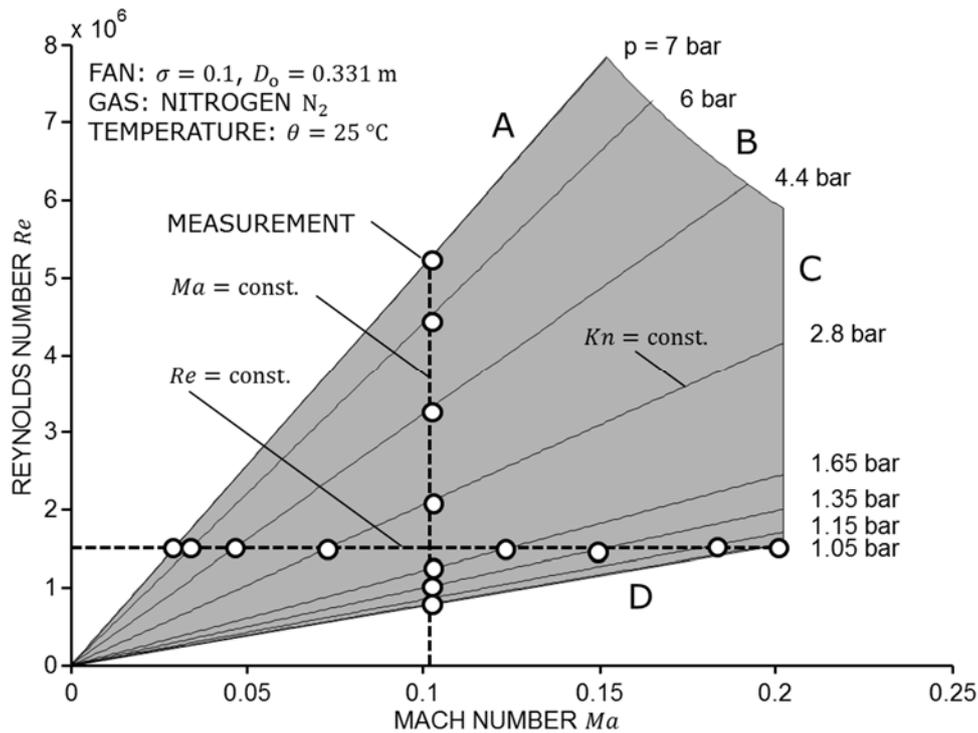


Figure 4: Mach and Reynolds number area.

RESULTS

The first part contains the measuring results, following by the validation of common scaling methods with the presented investigations.

Figure 5 shows the efficiency η versus the flow coefficient φ at constant Mach number $Ma = 0.1$. For increasing Reynolds number Re , the pressure p_1 is increased, while the rotational speed $n = 2123$ rpm and the temperature $\theta = 25$ °C are constant. A filled marker indicates the best efficiency point (BEP). The scaling effect from $Re = 0.77$ E6 to $Re = 5.21$ E6 is about $\Delta\eta = 0.07$.

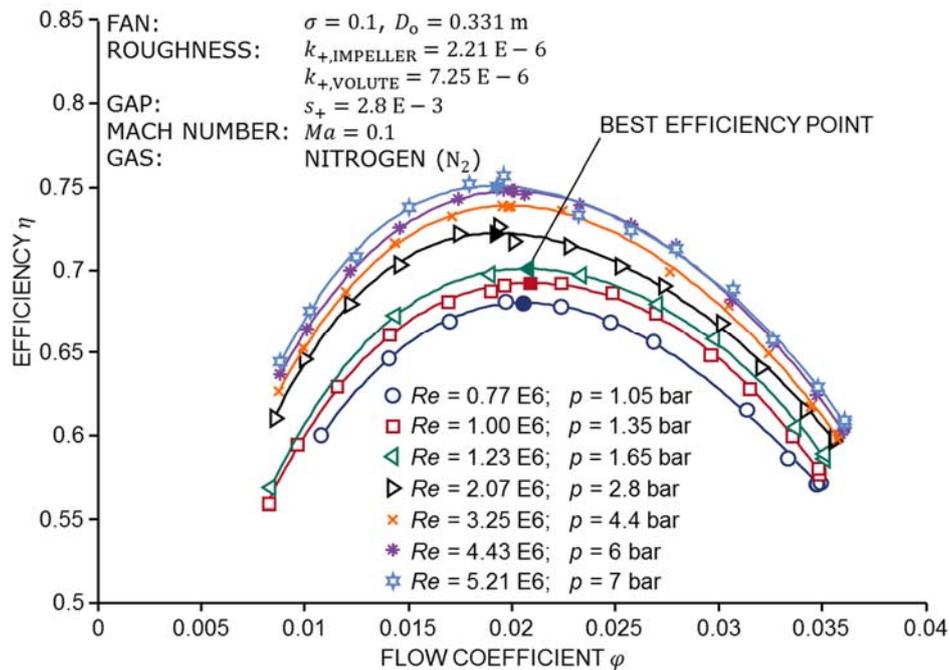


Figure 5: Efficiency characteristics at constant Mach number $Ma = 0.1$ and different Reynolds numbers.

Figure 6 summarizes the results of Figure 5 and the efficiency of the best efficiency point η_{BEP} is plotted against the Reynolds number Re . The uncertainty in Reynolds number is low in comparison to the efficiency uncertainty and therefore, only the efficiency uncertainty is plotted. Two efficiency scaling methods are proved. The first scaling method is from Saul & Pelz [4], which considers the friction losses in the impeller and the volute as well as the Carnot loss at impeller outlet. It is physically based and universally applicable. The second method is the most common scaling method from Ackeret [2], which only considers the change in Reynolds number and assumes a hydraulically smooth surface. Many standards recommend only 50 % of Ackeret’s predicted efficiency rise due to the before mentioned assumptions. Therefore, 50 % of Ackeret’s predicted efficiency rise are implemented as the reference method.

Table 1 summarizes the predicted efficiency changes from Saul & Pelz and Ackeret as well as the measured efficiency change from a reference Reynolds number $Re = 0.77 E6$ to the highest Reynolds number $Re = 5.21 E6$.

Table 1: Measured and predicted efficiency rise from $Re = 0.77 E6$ to $5.21 E6$.

	measurement	Saul & Pelz [4]	Ackeret 50 % [15]
efficiency change $\Delta\eta$	0.070	0.077	0.025

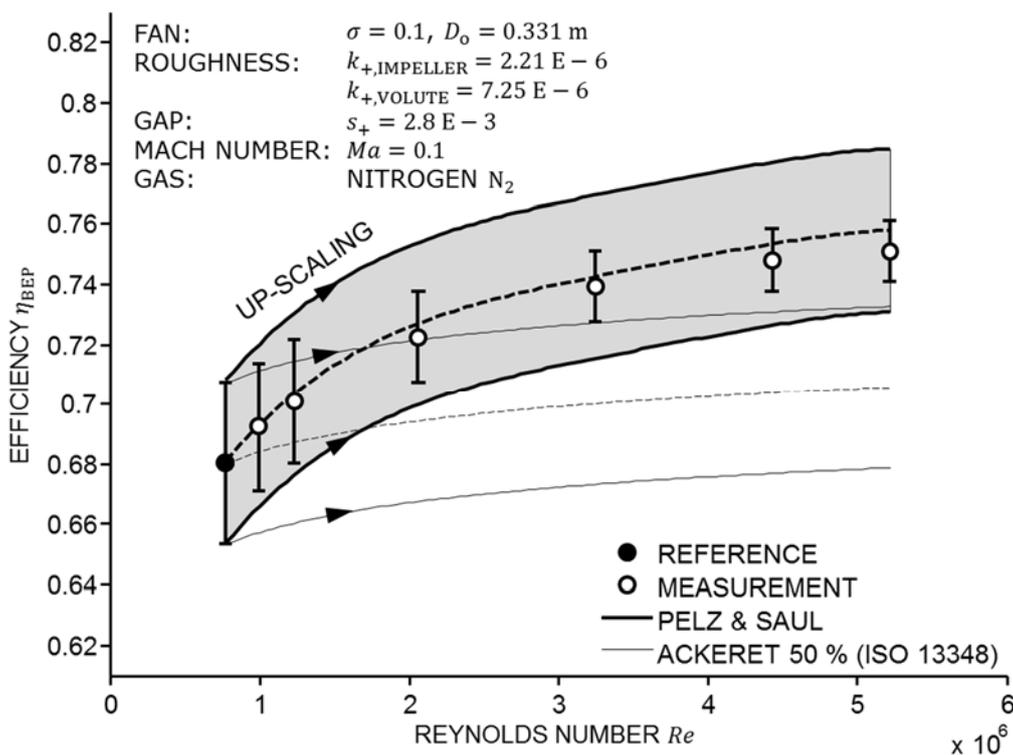


Figure 6: Measurement and prediction of the efficiency at best efficiency point versus the Reynolds number.

The adjustment of rotational speed n and ambient pressure $p_1 \propto n^{-1}$ results in a constant Reynolds number $Re = 1.5 E6$. Figure 7 shows the efficiency η_{BEP} versus the Mach number. For constant Reynolds and increasing Mach numbers, the efficiency stays constant. The scaling method from Ackeret cannot detect a Mach number effect but the method from Pelz & Saul detects a negligible decrease in efficiency due to low Mach numbers $Ma = 0.03 \dots 0.21$.

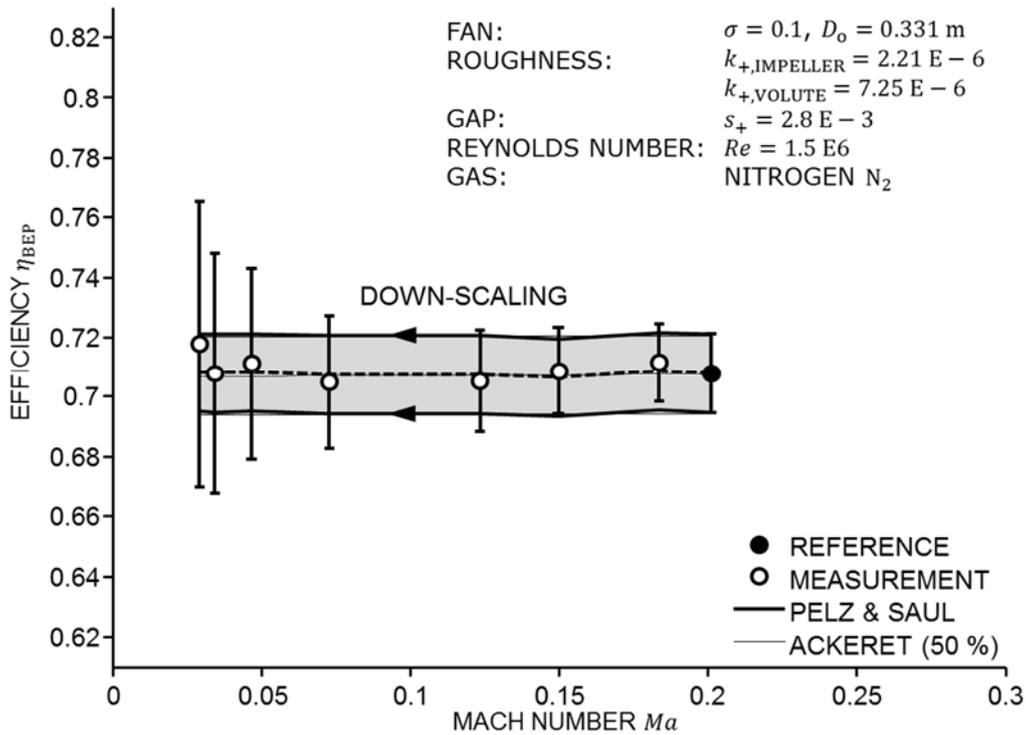


Figure 7: Measurement and prediction of the efficiency at best efficiency point versus the Mach number.

Darmstadt scaling method predicts the pressure coefficient as well, which is based on Hess' observations [11]

$$\psi_{BEP} = \psi'_{BEP} \frac{\eta_{BEP}}{\eta'_{BEP}} \quad (10)$$

Figure 8 shows the prediction of pressure coefficient. The change of pressure coefficient is $\Delta\psi = \psi - \psi' = 0.081$ in the Reynolds number range $Re = 1.2E6 \dots 5.2E6$ and the Darmstadt scaling method predicts $\Delta\psi = 0.1$. The overestimation is inside the measuring uncertainty.

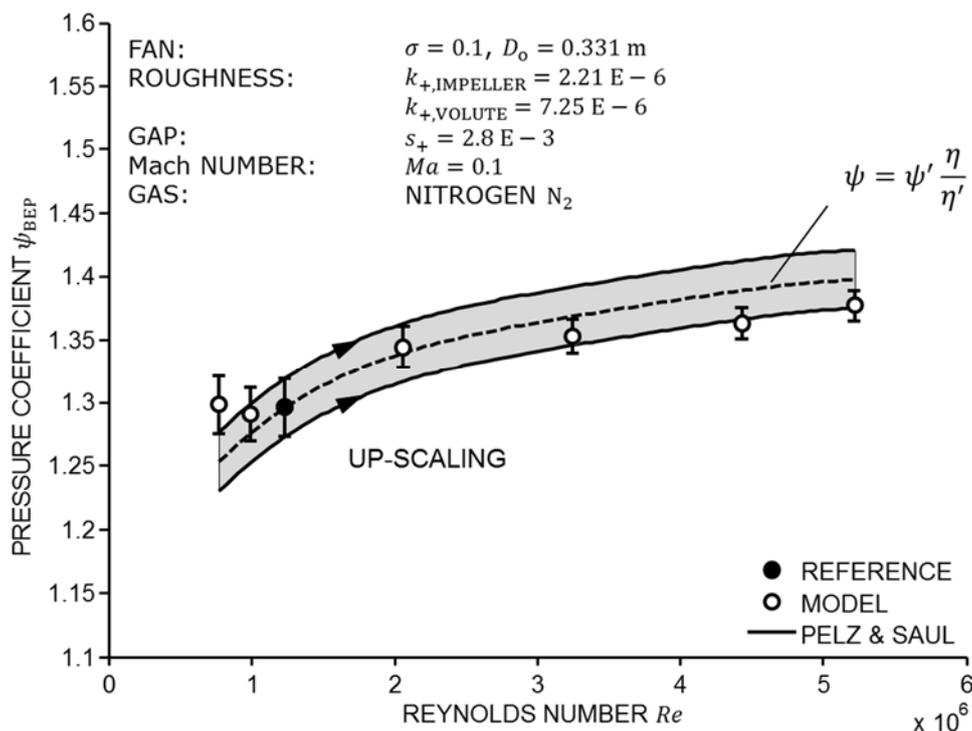


Figure 8: Measurement and predicted pressure coefficients at best efficiency point.

DISCUSSION

The experimental investigations show reasonable results. All characteristics are plausible and the efficiency at best efficiency point η_{BEP} increases for increasing Reynolds number, which was expected. The efficiency rise (Figure 6) is predictable with the scaling method from Pelz & Saul [4]. Ackeret's scaling method shows worse results than Pelz & Saul's method, due to the assumption that 50 % of all losses are scalable. In this case, only the Reynolds number is increased and all other parameters, which influence the efficiency, like the relative roughness k_+ and the relative gap s_+ , are unchanged. The pressure coefficient scaling (Figure 8) shows good results and in comparison to Ackeret's method, the Darmstadt scaling method predicts the pressure coefficient, too. In conventional test rigs a Reynolds number rise of 600 % is only possible, if the measurement techniques are changed and very often another scaled-model is necessary. This leads to higher manufacturing and measurement uncertainties and the quality of the fan is often changed because the relative roughness or relative gaps are varied. Thus, comparisons at constant Reynolds numbers and different Mach numbers or constant Mach numbers and different Reynolds numbers are possible with conventional test rigs but they are more complex and have higher uncertainties, than the described setup with a pressure vessel.

The measurement uncertainty of each measured efficiency point in the pressurized atmosphere changes because the line pressure effect increases the uncertainty of the pressure scanners. This effect can be reduced with a re-zero of the pressure scanners [16], which has to be done for every pressure level. The other measurement techniques, like torquemeter, rotary encoder and thermometers are not affected by a changing ambient pressure.

The power consumption of the electric engine and the stiffness of the rotor or impeller has to be taken into account. For increasing pressure, the density $\rho_1 \propto p_1$ increases. Therefore, the torque $M \propto p_1$ and the pressure rise $\Delta p \propto p_1$ increase as well. In order to avoid high torques and pressure rises, the rotational speed shall be matched.

CONCLUSION

Conventional test rigs for fans, which run at common ambient conditions, have the drawback that Mach and Reynolds number cannot be changed independently of each other. The easiest way to decouple Mach and Reynolds number is a change of ambient pressure. All other changes, like scaling factor or temperature, are possibilities, needing additional test rigs and measurement techniques, which increases the uncertainty of the measurement. Hence, the theory of decoupled Mach and Reynolds number is presented and a pressurized test rig is designed. First measurements show good results for constant Mach and increasing Reynolds number as well as for constant Reynolds and increasing Mach number. Additionally, the range of Reynolds number is enlarged due to the increasing pressure, being able to run the model at full scale Reynolds number.

The investigation of compressible effects is possible with this test rig but the rotational speed has to be increased to reach Mach numbers $Ma > 0.4$. Furthermore, additional fan types shall be investigated, regarding the influence of Mach and Reynolds number on the efficiency as well as on the flow, pressure and power coefficient.

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