TESTING, MODELING AND SIMULATION OF FANS WORKING WITH ORGANIC VAPORS

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ABSTRACT

The present contribution presents a test facility for fans working with organic vapors where non-ideal gas effects are relevant. In contrast to conventional testing of atmospheric fans a thermodynamic approach based on very precise temperature measurements is employed for obtaining the performance curves. The experimental data are used for validation of three-dimensional computational fluid dynamics (CFD) simulations of the organic vapor flows within the turbomachinery including real gas effects. The combined experimental and numerical analysis approach enables the assessment of conventional design rules and correlations. It is found that the influence of the Mach number on efficiency is dominant, and the expected Reynolds number scaling is fully compensated by compressibility effects in the case of an organic vapor with its low speed of sound.

FANS AND BLOWERS, SCALING METHODS, REAL GAS FLOW, TESTING, CFD

NOMENCLATURE

a speed of sound
c_{f} friction coefficient
D impeller diameter
h specific enthalpy
K transonic similarity parameter
Ma Mach number
n running speed
p pressure
P power
Re Reynolds number
u circumferential speed
T temperature
T torque
V volume
Y specific work

Greek Symbols

α correlation exponent
γ isentropic exponent
Γ fundamental thermodynamic derivative
ϕ flow coefficient

ρ density
η efficiency
ν kinematic viscosity
ψ pressure coefficient

Subscripts
o total
1,2 inlet, outlet
av average
m model or datum
s entropy

INTRODUCTION

In several technical applications, fans for organic vapors are needed. This market might be of increasing relevance when synthetic fuel production plants would become more important in the near future. In the published literature on the design of fans and blowers, most investigations consider air or an ideal gas law for describing the thermodynamics and fluid dynamics of this kind of turbomachinery, see for instance Dixon and Hall (2010). Organic vapors or other real gas fluids deviate significantly from the ideal gas behavior, especially near their saturation lines. In the past, no specific attention was spent on non-conventional real gas effects in fans. Only in very few exceptions it is possible to re-formulate the governing flow equations of real gases in order to get mathematically equivalent ideal gas flow equations as pointed out in detail by Traupel (1952). In the case of compressors (working with substantial pressure ratios or density changes), it was earlier mentioned by Cui (2000) that "the results from the studies using air or other ideal gases need to be confirmed before applying them to the real gas situation". In the case of fans and blowers (working with much smaller pressure ratios and density changes), it is typically assumed that the flow would be fully incompressible and hence there would be no need to modify the conventional design rules of fans. This means, that a manufacturer of a fan desired for organic vapor or real gas applications could directly apply the conventional fan laws (see, for instance, Bohl (1983)) for designing the turbomachinery. Most workers in this field would certainly agree that this is the common industrial and academic practice. However, a close inspection indicates that such an approach is – at least in principle – not free of difficulties.

Firstly, organic vapors or heavy gases lead typically to high Reynolds number flows due to their high density. This means that the influence of Reynolds number and roughness on the efficiency of fans becomes an important design challenge. Even in the case of fans working with atmospheric air, the choice of a good scaling method is still far from being trivial as demonstrated by Pelz and Stonjek (2013). This issue is more important regarding organic vapor fans because even a full-scale test with air would underestimate the Reynolds number by a magnitude or more. Secondly, high-speed flows (with Mach numbers Ma significantly greater than zero) of organic vapors are characterized by a significant real gas behavior in contrast to the well known ideal gas dynamics. Due to the high molecular mass, the speed of sound a is typically much smaller for organic vapors than for air or steam. This means that locally the actual Mach number level can be comparable high in fans working with organic vapors, and hence the reliability of conventional fan design rules becomes questionable. Even in the case of air, it was found in a recent study (Stonjek (2015)) that the effect of the Mach number on the efficiency of a fan is much more complicated than frequently assumed. In addition to that, the similarity analysis does not provide a simple reformulation of common ideal gas correlations in the case of organic vapors. Regarding so-called Organic-Rankine-Cycle (ORC) turbines, these facts are well known (Colonna et al. (2015)). Missing so far is a detailed investigation of the fluid dynamics of organic vapor fans and compressors. The present contribution might be interpreted as a first step into the new field of developing appropriate scaling methods for the design of fans working with organic vapors or heavy gases.

REVIEW OF SCALING METHODS

With a look to the present application, two different classes of scaling methods can be identified in literature, namely (i) scaling methods for fans and (ii) transonic scaling laws for real gas flows past airfoils. The first class of scaling methods has been developed for comparing fans of the same shape but at different Reynolds and Mach numbers. These scaling methods include the change of the working fluid implicitly but are essentially limited to ideal gases characterized by an isentropic exponent γ. The latter assumption is not free of difficulties for organic vapors because the second class of scaling laws indicates that such fluids might require a more complex thermodynamic approach than the simple gas laws.

The total pressure rise $\Delta p_o = p_{o2} - p_{o1}$ and the (aerodynamic or hydraulic) efficiency η of a centrifugal fan depend on the machine size expressed by impeller diameter D, rotational speed $2\pi n$, representative roughness height, gap width, and fluid properties. The relevant fluid properties are given by density ρ, kinematic viscosity ν, speed of sound a, and typically by the isentropic exponent γ.

In the classical treatment (see, for instance, Bohl (1983)), the volume flow rate $\dot{V} = \dot{m} / \rho$ and the total specific work $Y_o = \Delta p_o / \rho_{av}$ have proven their usefulness for characterizing the operation point of a fan. Useful dimensionless parameters among others are: flow coefficient $\varphi = 4\dot{V} / (\pi D^2 u)$, Reynolds number Re = uD/ν, Mach number Ma = u/a, and pressure coefficient $\psi = 2Y_o/u^2$ defined by means of the blade speed $u = \pi D n$.

The total-to-total internal efficiency η of a fan is defined by (see Dixon and Hall (2010))

$$\eta = \frac{h_{o2s} - h_{o1}}{h_{o2} - h_{o1}}. \tag{1}$$

In the case of organic vapors, the values of the specific enthalpy h required for evaluating equation (1) have to be computed by means of REFPROP data or appropriate equation of states on the basis of measured temperature and pressure values. In the case of an ideal gas (air) with practically constant specific heat capacities, equation (1) reduces to the simple relation

$$\eta = \frac{\left(p_{o2} / p_{o1}\right)^{\frac{\gamma-1}{\gamma}} - 1}{T_{o2} / T_{o1} - 1}. \tag{2}$$

For fans, it is often common (Bohl (1983)) to use the efficiency definition

$$\eta = \frac{Y_o}{h_{o2} - h_{o1}} \qquad \text{(ideal gas:} \qquad \eta \cong \Delta p_o / (\rho_{av} c_{p,av} (T_{o2} - T_{o1}))) \tag{3}$$

instead of the isentropic efficiency (2), but the resulting difference is rather academic for the following purposes (eq. (3) is typically chosen in the case of incompressible fluids).

In literature, many scaling methods are available for centrifugal fans. An important task is to estimate the so-called size effect (better described by the term Reynolds number effect), i. e. the change of efficiency η for two machines of the same shape but operated at different Reynolds numbers. The first physically based scaling method can be traced back to Pfleiderer (1955) who proposed the loss relation

$$\frac{1-\eta}{1-\eta_m} = \frac{\zeta}{\zeta_m} = \left(\frac{\text{Re}}{\text{Re}_m}\right)^{\alpha}. \tag{4}$$

with an empirical correlation exponent α with values between -0.25 and -0.1. Obviously, correlation (4) does not account for Mach number or further fluid property effects.

A promising physically based scaling method has been developed at the TU Darmstadt, see Pelz and Stonjek (2013) and Stonjek (2015). This method employs the concept of a master curve, i.e. a machine-individual performance curve in the η–ϕ-plane. For a machine obeying a master curve, the relation

$$\eta(\phi, c_{\mathrm{f,\,m}}) + \Delta\eta = \eta(\phi + \Delta\phi, c_{\mathrm{f}}) \quad \text{with } \Delta\eta \propto \Delta\phi \tag{5}$$

is fulfilled. The logarithmic change of loss and friction coefficient calls

$$\frac{\Delta\zeta}{\zeta} = \frac{\Delta c_{\mathrm{f}}}{c_{\mathrm{f}}} \tag{6}$$

and the determination of the friction coefficient c_{f} remains as task. This can be done by assuming suitable friction models for the turbomachinery. This method is able to account for Reynolds number, roughness, and gap width effects, but the question about the Mach number effect is still unanswered. Based on case studies (Stonjek (2015)) it was found that a Mach number effect exist even when the Mach number level seems to be lower than the classical threshold value (Ma < 0.3).

The use of heavy or dense gases for aerodynamic research in closed wind tunnels has been discussed in literature in detail; see Wagner and Schmidt (1978), Cramer et al. (1996) or Anders et al. (1999). The outcomes of these studies is that for compressible real gas flows obeying an inviscid two-dimensional transonic small disturbance equation the dimensionless transonic similarity parameter

$$\mathrm{K} = \frac{1 - \mathrm{Ma}^2}{(\Gamma\,\mathrm{Ma}^2)^{2/3}} \tag{7}$$

with the fundamental thermodynamic derivative (Thompson (1971))

$$\Gamma = 1 + \frac{\rho}{a}\left.\frac{\partial a}{\partial \rho}\right|_s . \tag{8}$$

is of importance. In the perfect gas limit, equation (8) leads $\Gamma \to (\gamma + 1)/2$. For real gas flow applications the well-known γ-similarity law for turbomachines of the same shape but working with different fluids should be re-considered because K and Γ are the relevant quantities instead of Ma and γ as in ideal gasdynamics. The friction coefficient c_{f} is also affected by real gas effects in a sophisticated manner. With regard to this, the available literature is not able to provide a definite answer yet.

TEST FACILITY AND EXPERIMENTAL PROCEDURE

The main part of the test facility which was recently erected is given by a closed loop organic vapor wind tunnel (called CLOWT) which has been described in more detail by Reinker et al. (2015, 2018). The complete test facility without a thermal insulation casing is shown in Figure 1. Due to an inventory control approach and an independent temperature level control, different Mach and Reynolds numbers can be investigated. The fans and compressors under investigation are part of this closed wind tunnel working with air or vapors at variable pressure and temperature levels.

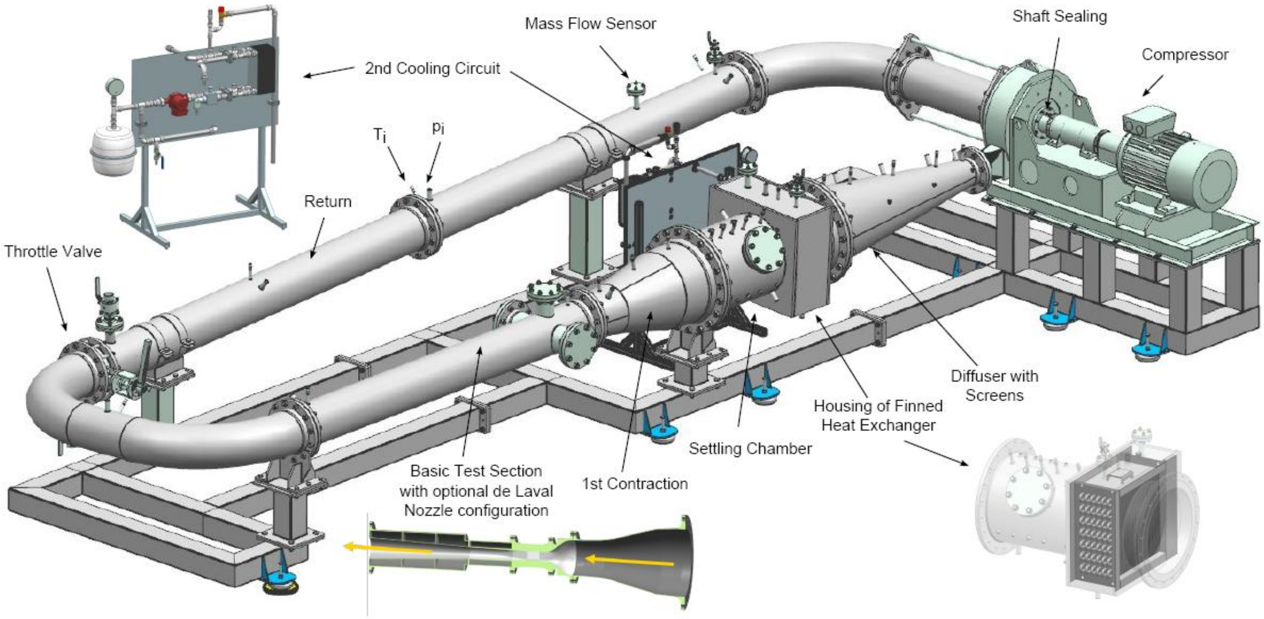

Figure 1: Test facility CLOWT and main components (after Reinker et al. (2018))

In the case of air, the test facility can be also operated in an open mode without 180°-return and without further internal flow obstacles in order to minimize pressure drops. However, due to the working fluid restrictions, it was not possible to operate the test fan fully in accordance to the standard test conditions recommended for fans working with atmospheric air.

The considered fan of this case study with impeller diameter $D = 0.35$ m is shown in Figure 2. It is important to note that the fan was designed for an operation point for the organic vapor Novec 649 by 3M as working fluid. The dynamic viscosity of Novec 649 is comparable with the value for air and it does not substantially depend on density. In many technical applications the density level of the organic vapors is rather high (of order 50 kg/m^3). This means that their Reynolds number level is also rather high due to the low kinematic viscosity of organic vapors. The speed of sound of Novec 649 is much lower than for air; typical values are $a = 85$ up to 110 m/s depending on temperature and density. In conventional performance test codes it is recommended to measure the fan input power $P = 2\pi n\mathrm{T}$ by means of a torque measurement T. In the case of organic vapor fans this standard approach is sometimes challenging due to sealing issues.

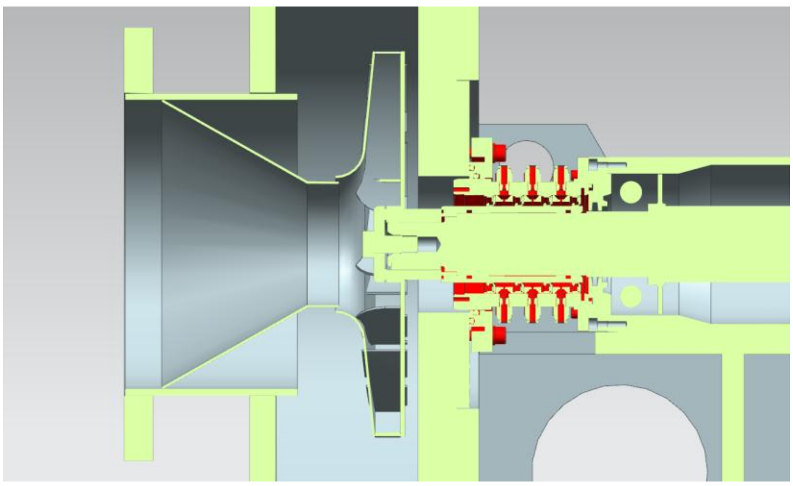

Figure 2: Considered fan with multiple dynamic lip seal system (indicated by red color)

This was also the case for the considered fan with a multiple dynamic lip seal system shown in Figure 2. The fan was operated with different running speeds n; the nominal design point corresponded to $n = 3000\ \mathrm{min}^{-1}$. Different flow rates were adjusted by means of a throttle valve located at the entrance of the return pipe. The mass flow rate was measured by a mass flow sensor located at the return. The static wall pressure p_1 and p_2 and the (total) temperatures T_{o1} and T_{o2} were measured at the beginning of the 90°-elbow and at the beginning of the diffuser, respectively. For the temperature measurement a recovery factor of unity was assumed, but due to the low velocity level in the pipes, this assumption was not crucial. During operation, running speed, electric power consumption, and bearing temperatures of the fan were recorded. The volume flow rate was independently checked by velocity measurements by means of hot-wire anemometry and by Prandtl probes. This independent check enabled a good assessment of the experimental uncertainty level.

The static pressure rise $p_2 - p_1$ was measured with a total uncertainty level of $\Delta p = 5.3$ Pa under best conditions. The uncertainty level of the mass flow rate was of order 1 up to 2 %. The major uncertainty contribution was given by the temperature measurements. As indicated by equation (3) accurate measurements of inlet and outlet temperatures were of major importance for the determination of fan efficiency. In the case of air at atmospheric pressure, the temperature difference $\Delta T_o = T_{o2} - T_{o1}$ was of order 1 up to 2 K. This means that the experimental uncertainty level of the temperatures had to be well below 0.1 K. To achieve such a level, the following items were considered:

(i) The involved high-resolution thermocouples were carefully calibrated using a laboratory normal and a calibration device (thermal bath) at the beginning of the measurements and the high resolution of the thermocouples was checked. It was possible to achieve a level of order 0.05 K.

(ii) The entire test facility CLOWT was placed in a closed laboratory room but due to the different sunshine conditions and the position of the room in the building, small initial temperature differences of order $\Delta T = 0.1$ up to 0.4 K between the measurement locations 1 and 2 were observed. In order to avoid systematic errors caused by the different ambient temperature zones in the room, the test facility was set to thermal equilibrium prior to the experiments.

(iii) In a first set of experiments, the temperature and pressure data were continuously recorded, and their transient behavior were controlled. Final data were accepted only when statistically steady states were achieved. Due to the significant thermal inertia of the involved test facility (pipe walls, casing), this requirement led to rather long periods of operation for each final data point. During long-time operation, a systematic temperature drift of order 0.1 K up to 1 K was observed due to heating of the laboratory hall. That drift was identified as major source of uncertainty. Due to heat conduction within the thermocouples and the casings, parasitic heat flows to the surroundings occurred. The existence of a parasitic heat flow from the temperature sensor head to the outer casing wall led to systematic errors in final temperature results. In order to avoid that, the metal temperatures at two different radial positions were recorded at the sensor locations. A final data point was accepted only when that parasitic temperature difference was below 0.1 K. However, this procedure led to a total uncertainty level of order $\Delta\eta = \pm$ 10 % which was assumed to be too high.

(iv) In order to reduce the total uncertainty level $\Delta\eta$, a novel thermal measurement approach based on system identification was developed. Details about this novel method can be found in aus der Wiesche at al. (2019). The main idea of this method is to record the entire step response of the temperature measurement system and to identify the system parameters by means of a fitting procedure. Since the step response is governed by the temperature difference $\Delta T_o = T_{o2} - T_{o1}$ created by the fan, the time signal of the response can be employed for identifying this desired quantity. The identification process requires knowledge of the involved transfer functions of the dynamic system, but it was found that this more sophisticated approach led to a much better uncertainty level as the initially suggested direct temperature difference measurement approach.

Finally, it is emphasized that the above methodology did not refer to ISO standards, but the special design and the elevated pressure and temperature levels of organic vapor flows required an alternative approach.

NUMERICAL METHOD

The simulation of fan performance by means of computational fluid dynamics (CFD) methods is still challenging. Based on the wind tunnel CAD model, the fluid domain was derived for the following numerical analysis. The geometry consisted of an elbow at the system inlet, a volute with a rotating impeller and a diffuser at the system outlet, fully in accordance to the experimental setup and its measurement positions. The discretization of the fan was done with unstructured tetrahedral elements used for the meshing of all regions, as can be seen in Figure 3. The numerical simulations were performed utilizing the commercial CFD software ANSYS CFX 18.1 based on the finite volume method.

For solving the three-dimensional, steady, compressible, viscous Reynolds averaged Navier-Stokes equations (RANS) were applied (see, for instance, Ferziger and Peric (2002)). For turbulence modeling the Shear Stress Transport (SST) turbulence model by Menter (1994) and automatic wall functions were used. The rotational domain was modeled by the Multiple Frames of Reference (MFR) approach and applying the Mixing Plane (Stage) model. As boundary conditions, a mass flow inlet at the elbow entry, a pressure outlet (opening) at the diffuser exit and a rotation of the rotating domain were applied, see Figure 3. All other remaining faces have been set to adiabatic walls with no slip condition.

For simulating air flows, the well-known equation of state (EOS) $p\mathrm{v} = RT$ for the ideal gas with specific gas constant R can be selected. In the case of organic vapors, the assumption of an ideal gas flow leads typically to significant errors, and a more appropriate EOS is required. After some preliminary tests, the Peng-Robinson model (Peng and Robinson (1976)) has been selected for performing the compressible CFD analysis. In the case of air, the deviations between the Peng-Robinson model

$$p = \frac{RT}{\mathrm{v}-\mathrm{b}} - \frac{\mathrm{a}\,\alpha}{\mathrm{v}(\mathrm{v}+\mathrm{b})+\mathrm{b}(\mathrm{v}-\mathrm{b})} \quad \text{with} \quad \mathrm{a} = 0.4572\frac{(RT_{cr})^2}{p_{cr}},\ \mathrm{b} = 0.0778\frac{RT_{cr}}{p_{cr}} \tag{9a}$$

$$\sqrt{\alpha} = 1 + (0.3746 + 1.5423\omega - 0.2699\omega^2)(1-\sqrt{T/T_{cr}}) \tag{9b}$$

and the ideal gas EOS remained negligible, but in the case of NOVEC, the deviations were significant. In equation (9), ω is the acentric factor, and the subscript cr denotes the critical point. These data can be obtained from databases such as REFPROP.

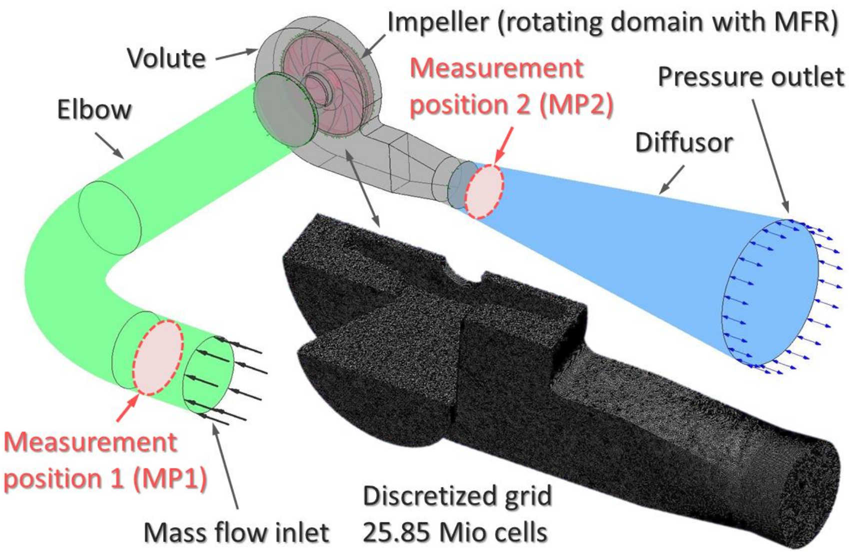

Figure 3: Meshed fluid domain with boundary conditions

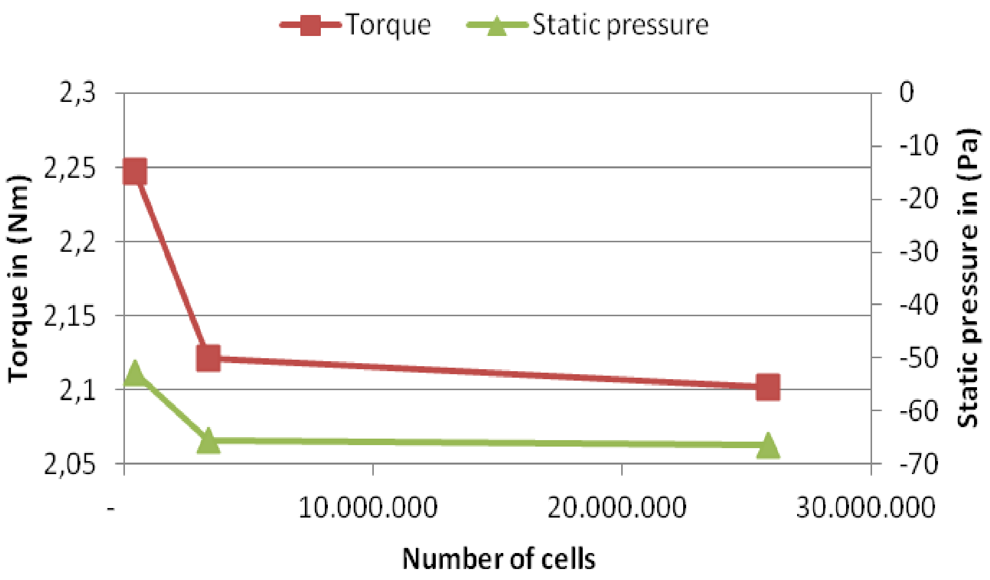

Figure 4: Grid study for two typical fan performance quantities

The convergence criteria was based on monitor points of the total pressure rise, the comparison of the mass flow rate at the inlet and the outlet and combined with residuals less than 10^{-4}. In order to estimate the discretization error, a detailed grid study was performed with three specific refined grids, i.e. the average element size was halved in each case (8.54 mm, 4.32 mm and 2.10 mm). Torque and static pressure were evaluated for each grid and plotted in Figure 4. The GCI method (Grid Convergence Index), which is based on the Richardson extrapolation, was applied by evaluating the torque on the impeller and the static pressure at diffuser entry of each grid (Roache (1997)). The fine-grid convergence index was calculated considered a safety factor of 1.25. According to that method, the numerical uncertainty in the fine grid solution for the torque was about 0.229 % and for the pressure about 0.128 %.

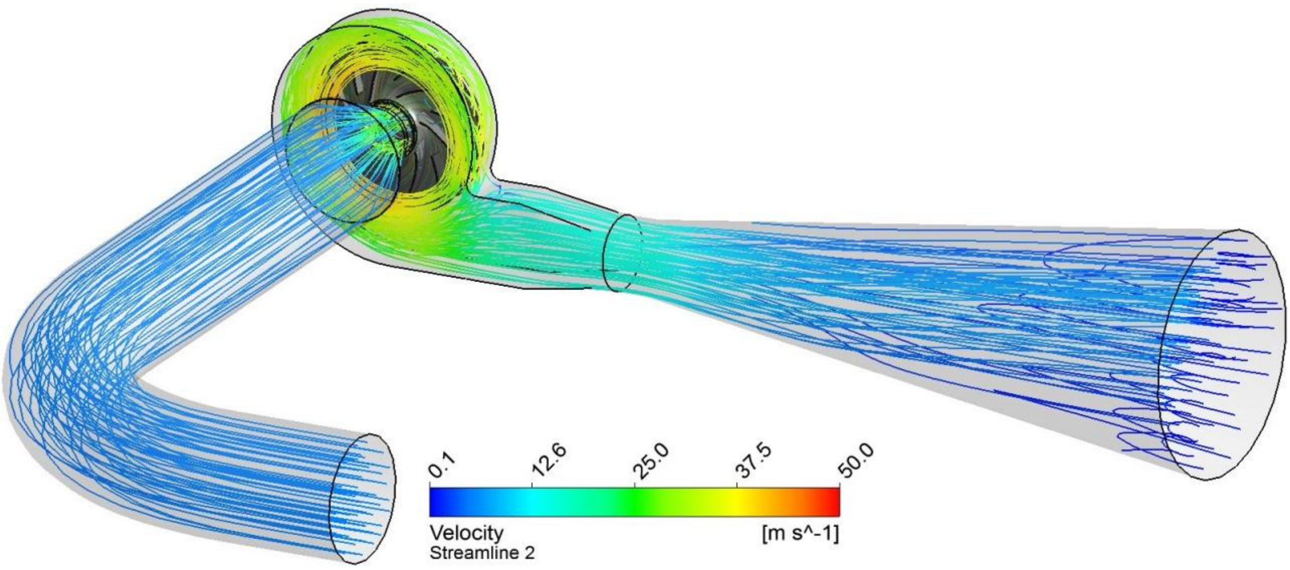

Figure 5: Computed pressure (top) and velocity fields (bottom) for a fan working with air

In Figure 5 computed stream lines (air at design point operation) are illustrated. The minor effect of the actual inflow configuration (elbow) and the diffuser on the inlet and outlet flow conditions of the fan can be observed by means of the streamlines in Figure 5. During post processing of the CFD results, the total pressure rise Δp_o was evaluated at measurement positions 1 and 2 (mass flow averaged), see Figure 3, corresponding to the actual experimental approach. The numerically computed torque T was evaluated on the impeller surface. The fan efficiency η was then numerically calculated by means of the common relation $\eta = \Delta p_0 \dot{V} / (2\pi n \mathrm{T})$.

RESULTS AND DISCUSSION

The considered fan of the present case study was designed in accordance to the well-known fan laws and similarity methods which were to certain extend reliable for air flow applications. The geometry of the impeller and the casing were adapted from a successful class of atmospheric air fans. The volume flow at design point was set to be of order 0.2 up to 0.3m^3/s.

The density of the organic vapor was of order 15 up to 55 kg/m^3 that is much higher than the density of atmospheric air. It should be kept in mind that in technical applications organic vapor flows are typically realized at elevated temperature and pressure levels leading to a complete different Reynolds number level in comparison to atmospheric air flows. Furthermore, due to the comparable low speed of sound *a*, the Mach number level of the organic vapor fans are typically much higher than in the case of air. In the present case study, a level of $\mathrm{Ma} = u/a = 0.65$ was achieved for NOVEC.

In a first validation test run with atmospheric air at the fan operation point, the accuracy of the novel thermal measurement approach for determining the fan efficiency was checked. For this flow condition, the efficiency η of the fan was known from prior ISO tests. The result of the ISO standard test was an optimum efficiency of $\eta = 81 \pm 3$ % at a volume flow rate of 0.25 m^3/s (in accordance with Kaufman and Falk (2007), at least a tolerance of 3 % has to be considered in the case when specified and site conditions do not exactly match). The initially conducted direct temperature measurements led to an experimental result 60 % < η < 86 % with an average value of all test runs of order 70 %. Obviously, the experimental uncertainty level of the direct temperature rise method was inacceptable high. The novel thermal measurement approach based on system identification led to a result $\eta = 80 \pm 4$ % which was in excellent agreement with the ISO standard testing. The CFD predicted a best efficiency of $\eta = 70$ %. This value was somewhat lower than the experimentally obtained efficiencies.

In a second test run, the dependency on the working fluid of the fan performance was investigated. As model of the organic vapor fan, the same machine but working with atmospheric air was considered. This means that both Reynolds and Mach number levels of the model remained much smaller than for the final organic vapor flow application for which the fan was considered. The non-matching of Reynolds and Mach number levels is a typical situation for fan testing and designing. In a first set of tests, fan performance and efficiency were assessed under model conditions (i. e. atmospheric air as working fluid). The experimental data were then used for validation of three-dimensional computational fluid dynamics (CFD) simulations. The results are shown in Figure 6 (using the high-resolution computational mesh for the CFD results). The computed performance (pressure coefficient) agreed very well with the measured performance at design and part load conditions. Only in the case of very high volume flow rates, noticeable deviations between the CFD and the experimental data were recorded. The experimental uncertainty level of the pressure rise measurements was rather small (of order 10 Pa) that was smaller than the symbol dimensions in the top diagram of Figure 6. Between different test runs, no significant data scattering regarding the total pressure rise as function of the volume flow rate were observed. Due to the actual temperature measurement approach for obtaining the efficiency, the situation was more difficult. The qualitative behavior of the measured efficiency and the computed curve agreed, see the bottom diagram of Figure 6.

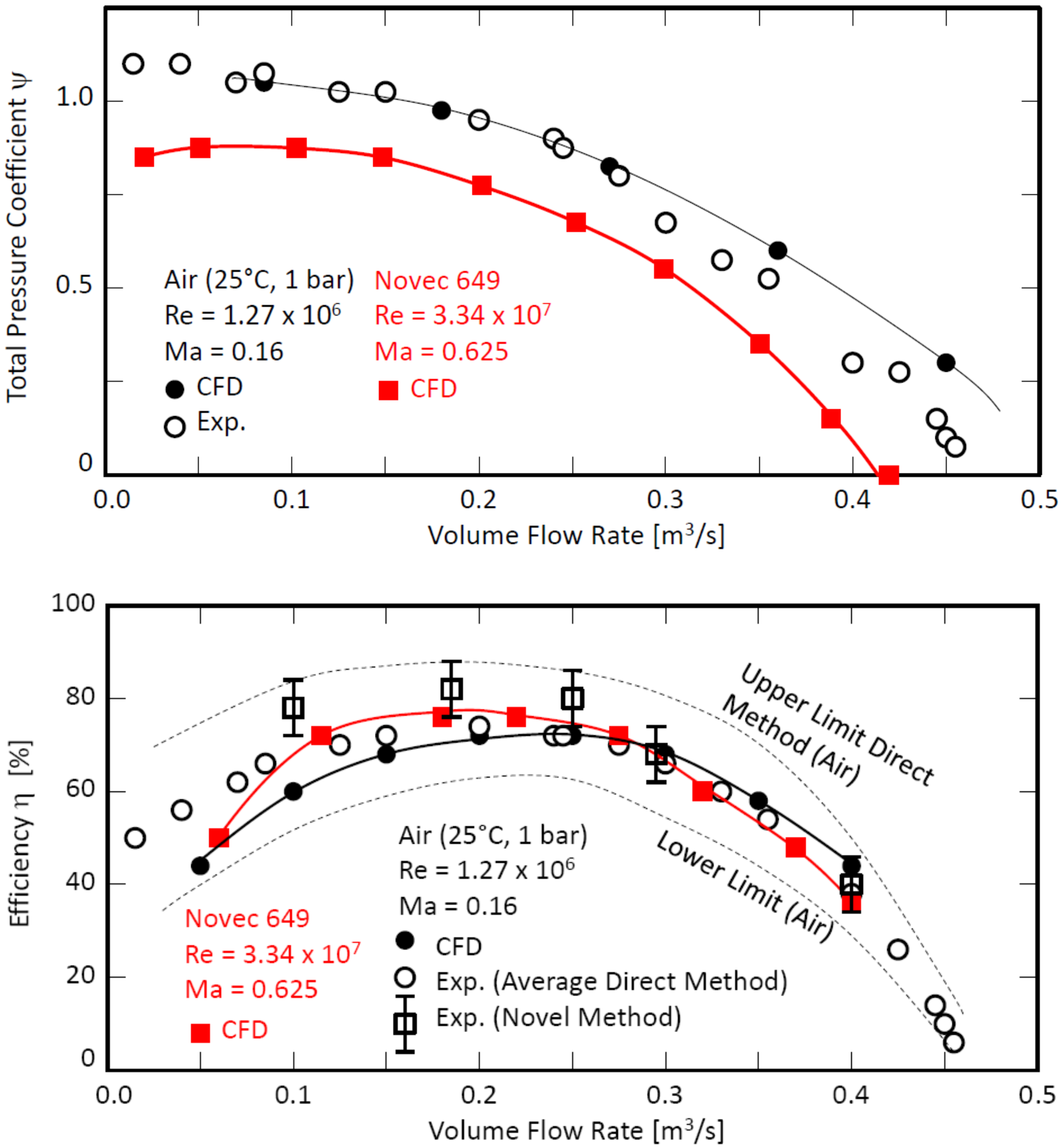

Figure 6: Performance curve (top) and efficiency (bottom) against volume flow rate for the considered fan working with atmospheric air and Novec 649 at nominal speed (3000 rpm)

However, due to the small temperature differences and drift effects during the testing operation, a large scattering of experimental efficiency data occurred in the case of the direct method (i.e. measuring the resulting temperature difference at inlet and outlet). The significant uncertainty level of the direct method is indicated by the both additional black boundary curves in the corresponding diagram showing the upper and the lower limits of the individual data points. The *average values* of the experimentally determined efficiency agreed very well with the computed value at design point and part load operation, but the actual experimentally uncertainty level opened the validation study to criticism. Applying the more sophisticated novel thermal measurement methid based on system identification led to a smaller uncertainty level but also to a slightly different picture: The efficiencies for air at part load condition was much higher and these values were very close to the available ISO standard test results. In the case of higher volume flow rates, the CFD predictions and all experimental methods led to rather similar results in the case of air. Based on the good (overall) agreement it was concluded that the present CFD method was capable for simulating the fan performance even under the conditions of real gas flows.

In Figure 6, numerical results (CFD) are plotted in red color for the fan working with the organic vapor Novec 649 at design running speed at a pressure and temperature level leading to rather high Reynolds and Mach number values. Obviously, the conventional fan laws are not appropriate for describing such a real gas flow, because significant deviations regarding the total pressure coefficient ψ were observed. Probably the higher Mach number level led to a systematic decrease of efficiency although the Reynolds number was much higher than for air. Such a qualitative observation was reported earlier by Stonjek (2015) in case of centrifugal fans, too. The present experimental data are not sufficient to answer the important question about the influence of Mach number and Reynolds number levels on fan performance and efficiencies. Subject of futire work is the experimental investigation of air and organic vapor flows for a large range of rynolds and Mach numbers.

CONCLUSIONS

The present contribution presents a test facility based on a closed wind tunnel for fans working with organic vapors employing temperature measurements for obtaining the efficiency. The experimental data were used for validation of three-dimensional computational fluid dynamics (CFD) simulations of the organic vapor flows within the turbomachinery including real gas effects. It is found that the influence of the Mach number on efficiency is dominant, and the expected Reynolds number scaling is compensated by compressibility effects in the case of an organic vapor with its low speed of sound. The next step is to experimentally investigate the real gas flow through the fan and to assess the accuracy of available scaling laws.

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