ANALYTICAL APPROACH FOR THE PERFORMANCE PREDICTION OF HIGH PRESSURE CENTRIFUGAL FANS IN SERIAL ARRANGEMENTS

Martin NUDISCHER[1], Hansgeorg BINZ[1],
Matthias BACHMANN[1], Stefan RECKER[2]

[1] *University of Stuttgart, Institute for Engineering Design and Industrial Design, Pfaffenwaldring 9,
70569 Stuttgart, Germany*

[2] *Elektror airsystems gmbh, Technology Development & Simulation.
73760 Ostfildern, Germany*

SUMMARY

This paper introduces a new method of calculating the performance of two or more fans which are serially arranged in order to increase the pressure. In the context of the paper serially arranged fans are defined as a combination of two or more single centrifugal fans. According to the literature the final performance of serial arrangements can be obtained by doubling the total pressure of the single fan at a constant volume flow rate. The measured results of such arrangements showed deviations to the predicted characteristic curve by the method of literature. The presented new calculation is based on the specific work instead of the total pressure increase and is evaluated with measurement results of different fan combinations. The proposed method calculates the characteristic curve in an iterative process and achieved a high accuracy in predicting the final performance compared to the measurement results. The introduced method covers serial arrangements of identical fans as well as serially arranged combinations of different fans.

INTRODUCTION

Centrifugal fans have a wide range of applications in industrial areas and consume a considerable amount of energy. In the year 2000 approximately 50 % of the world market of fans were centrifugal fans [1]. This could be one reason to put high efforts in research and development over decades, with the intentions of increasing efficiency, reducing noise or optimizing the flow in a single fan for example. For the integration of a fan into a system, it is designed for a defined operating point. Due to changes in the system affecting its resistance, the fan operates at a different point than originally planned. If the system resistance exceeds the capacity of the fan an intervention of the planner is needed. There are several options to increase the fan capacity: accelerating the fan impeller and operating it at higher speed, until the maximum permissible strength is achieved. Due to the feasible strength of the used materials (often steel or aluminum), the peripheral speed is limited to approximately 150 m/s for welded or riveted sheet metal impellers [2]. Thus, the available pressure and volume flow rate is still limited. If speeding up the impeller is not sufficient for the increased system demands, the fan can be replaced by a larger fan. Another solution to increase the available pressure is to combine two or more fans in a serial arrangement. Aside from the advantage of higher pressure, there is the potential to significantly save energy at part load conditions and for a larger variability in structural arrangement and electrical requirements. In the context of the paper serially arranged fans are defined as a combination of two or more single centrifugal fans and not as one fan with two or more impellers on one shaft. Despite only few scientific investigations on serially arranged centrifugal fans, many installed fan systems can be discovered in plants already. However, the coupled fans influence each other and are interdependent through the connected air paths and the mechanical connection.

Although the connection of two centrifugal fans appears to be very simple, barely any information or publication on this subject can be found. In the literature for centrifugal fans only short theoretical descriptions are given for a serial connection of two or more fans [3], [4], [5], [6]. Considerable changes in the design or behaviour when two or more fans are coupled cannot be found. Banzhaf [7] describes failures in power plants in which more than one fan is installed over the whole pipe system. However, the focus in these investigations is not on the fans and their influence on each other. Carolus [2] investigated the phenomena of the rotating stall and surge of fans in systems with two centrifugal fans. Rotating stall and surge are a kind of unsteady flow occurring in fans at low volume flow rates. The focus of this investigation is on the incidence and cause of these phenomena. Schulze-Dieckhoff [8] outlined how defined control interventions influence the behavior of serially arranged fans at non-stationary operating points.

As recent studies have shown, the method to calculate serially arranged fans which is described in literature has led to insufficient results [9]. This experimental study investigated a serial arrangement of two geometrically identical fans running at the same speed. Below the results are briefly described. For detailed information see [9]. Figure 1 shows the relative total pressure against the flow coefficient which can be equated with the volume flow rate. The relative total pressure is the actual pressure normalized by the maximum total pressure of the single fan (in this application 15 kPa).

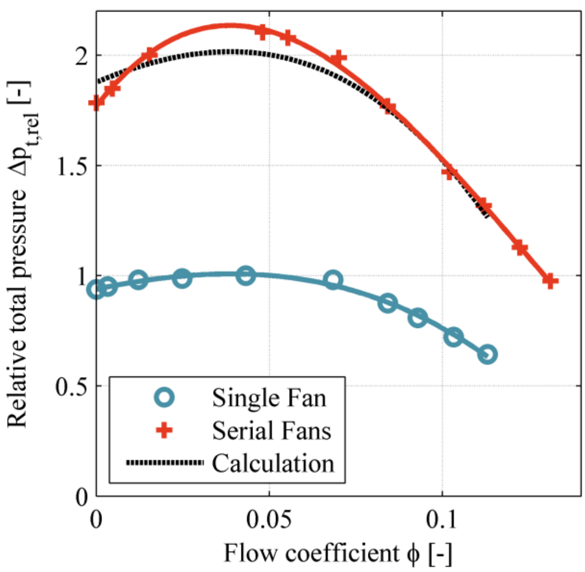

Figure 1: Relative total pressure against flow coefficient [9]

Therefore the maximum total pressure increase of the single fan is 1.0. According to the literature, the total pressure of a serial arrangement of two fans is the sum of total pressures of the individual fan [3], [4], [5], [6]. Thus, the maximum relative total pressure of two identical fans is 2.0 at best.

However, the measurements have shown that the maximum achievable pressure is higher than the predicted values of the calculation and the volume flow rate is slightly higher as well [9]. Furthermore, it is remarkable that the two fans showed deviations in their characteristic curves despite the geometrically identical structure of the fans [9].

Figure 2 shows a schematic of the serially arranged fans and their connection to the test bench. The schematic with its notation is the basis for the following calculation approach. The schematic and the calculation method concern discharged applications.

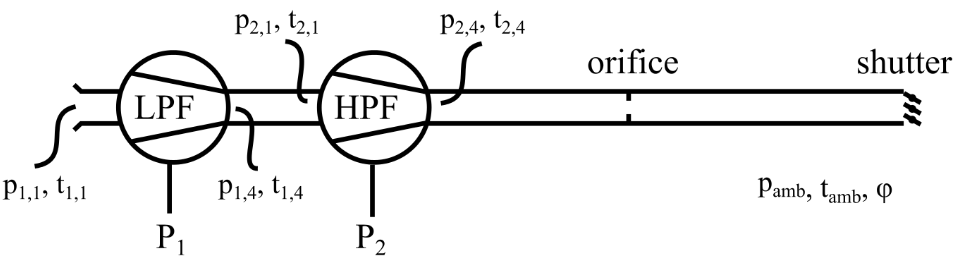

Figure 2: Schematic of serially arranged fans

The first fan of the serial arrangement, called low pressure fan (LPF), sucks the air from the atmosphere and discharges the compressed air into the second fan, called high pressure fan (HPF). Downstream of the arrangement the flow rate is determined via an orifice plate and a shutter is placed at the end of the outlet pipe to adjust the operating point. The behavior of the LPF is comparable to that of the single fan. The characteristic curve of the HPF has a different shape. At low volume flow rates, the pressure increase is lower compared to LPF. At higher volume flow rates the ratio of the pressure increase is higher in HPF. Thus, the total pressure increase of the serial arrangement is unevenly distributed except at one operating point. Therefore, a new method of calculating the performance of serially arranged fans is introduced based on the specific work and evaluated with measurement results of two different serially arranged fan combinations. It should further be noted that the pressure and temperatures are indexed by a subscripted number. The first number denotes the fan (1, 2, …) and the second number the position (1 for inlet, 4 for outlet).

SPECIFIC WORK

For fans it is common to describe the performance of a fan by the pressure increase as a function of the volume flow rate. Alternatively, the performance of a fan can be described through the specific work as a function of the volume flow rate. In general, the specific work is the amount of work done per unit mass of gas. The specific performance or specific work Y is described by the following integral [3]:

$$Y = \int_1^4 \frac{dp}{\rho} \tag{1}$$

The compression of the air in a fan is assumed to be a polytrophic process, hence:

$$pv^n = \text{const.} \tag{2}$$

Thus, the solution of the integral of formula (1) is:

$$Y = \frac{p_1}{\rho_1} * \frac{n}{n-1} * \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{n}} -1 \right] \tag{3}$$

The polytropic exponent n can directly be calculated with the inlet and outlet conditions of the fan:

$$n = \frac{\ln\left(\frac{p_4}{p_1}\right)}{\ln\left(\frac{\rho_4}{\rho_1}\right)} \tag{4}$$

Formula (4) shows the dependence of the polytropic exponent n on the operating point. Marcinowski [10] showed that the specific work can be approximated by:

$$Y_t = \frac{2 * \Delta p_t}{\rho_1 + \rho_4} \tag{5}$$

This formula is a simplification to calculate the specific work and is used for further calculations. The relative error by using formula (5) is smaller than 0.2 % up to a compression ratio $p_4/p_1 < 1.3$ instead of using formula (3) [10]. To determine the temperature after the compression, the ISO 5801 [11] proposed the following estimation:

$$T_4 = T_1 + \frac{P_{In}}{\dot{m} * c_p} \tag{6}$$

So it is possible to calculate the temperature and thereby also the density at the outlet based on the temperature at the inlet, the mass flow rate and the mechanical power input. The specific heat capacity remains constant [11].

CALCULATION APPROACH

By means of the formulas above, the calculation of serial arrangements can be conducted. Figure 3 summarizes the proposed steps in a flow chart. The first step of every calculation, whether a single fan or a serial arrangement, is to define the final performance of the fan(s). Therefore, information about the operating conditions is mandatory as well as the characteristics of the single fans. Afterwards, for a first point of reference a preselection of the fans for the arrangement is done by the standard calculation approach given in literature.

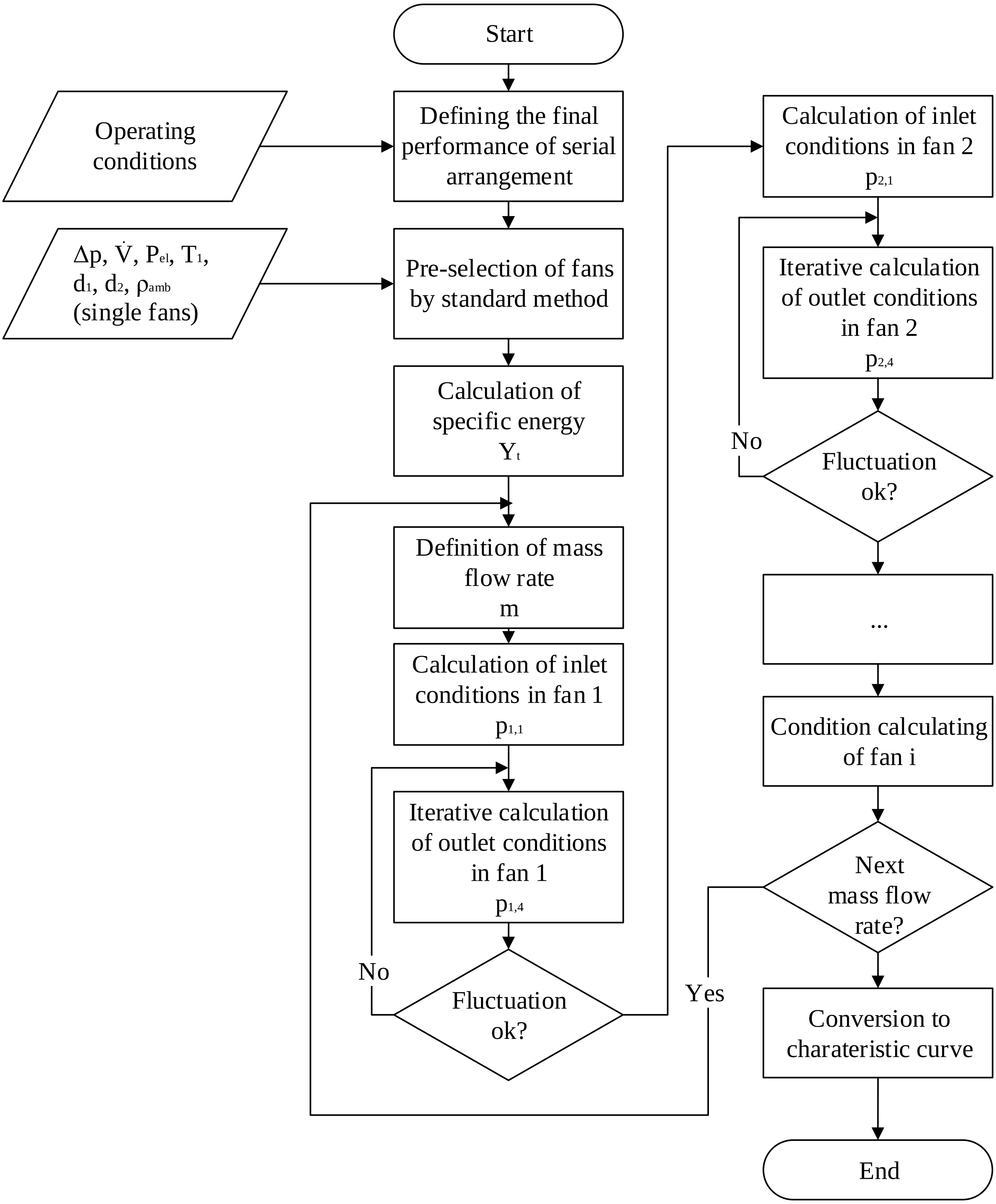

Figure 3: Flow chart to design serial arrangements

The required data of the preselected fans are the total pressure increase, the volume flow rate, the mechanical power consumption, the inlet temperature, the in- and outlet diameters and the ambient density. According to formula (5) and (6) the specific work can be calculated at different operating points. Therefore, different discrete mass flow rates have to be determined for the calculation. The pressure increase of the serial arrangement is calculated at these discrete mass flow values. The variation of the density within higher pressure fans cannot be neglected anymore. The calculation of the inlet conditions of fan 1 can easily be done with the measurement data of the single fan and the operating conditions with the standard formulas [11]. The calculation of the outlet conditions of fan 1 is a recursive process, because there are dependencies between the specific energy, the outlet temperature and the density. Depending on the starting values the calculation has a rapid convergence within just a few iterations. The proposed stop criterion for the iterative process is the fluctuation of less than 0.5 % of the outlet pressure between two iteration steps. The outlet conditions of fan 1 are used as inlet conditions for fan 2. Geometrical differences between the outlet of fan 1 and the inlet of fan 2 have to be taken into account. The next step of calculating the outlet conditions of fan 2 and each following fan is comparable to the iterative procedure for fan 1. This whole procedure has to be repeated for each of the selected discrete mass flow rates. By finishing the calculation for all selected mass flow rates, the resulting characteristic curve can be generated and the calculation is finished.
The proposed calculation approach is evaluated by existing measurements of two different serial arrangements. In addition, the calculation can be adapted to suction applications. Therefore, the calculation has to start with the last fan in the serial arrangement and other boundary conditions.

EXPERIMENTAL VALIDATION

The final performances of the arrangements are measured on a test rig according to international standard ISO 5801 [11]. A detailed description of the measurement uncertainties can be found in [11], [12], [13], [14]. The estimation of the uncertainties is stated in Table 1.

Table 1: List of uncertainties

Static pressure	1.4 %
Volume flow rate	2.0 %

Figure 4 shows the experimental setup. Both examined configurations consist of two identical fans at the same speed to avoid distortion due to design differences. Only the necessary pipe elements are installed to connect the fans in order to reduce the flow losses. The technical specifications of the different fans are stated in Table 2. All results concerning the volume flow rate are displayed as a function of the dimensionless flow coefficient [3]:

$$\phi = \frac{4*\dot{V}}{\pi*D^2*u_2} \tag{7}$$

In this instance D stands for the outer impeller diameter and u_2 for the circumferential speed of the impeller at its outlet. Differences in the flow coefficient between the LPF and the HPF are the result of the different average volume flow rates in the fans due to an increasing density within the system. All results regarding the pressure increase are displayed as the dimensionless head coefficient [3]:

$$\Psi = \frac{\Delta p_t}{\frac{\rho}{2} * u_2^2} \tag{8}$$

The head coefficient establishes the link between the pressure increase and the circumferential speed at the impeller's outlet.

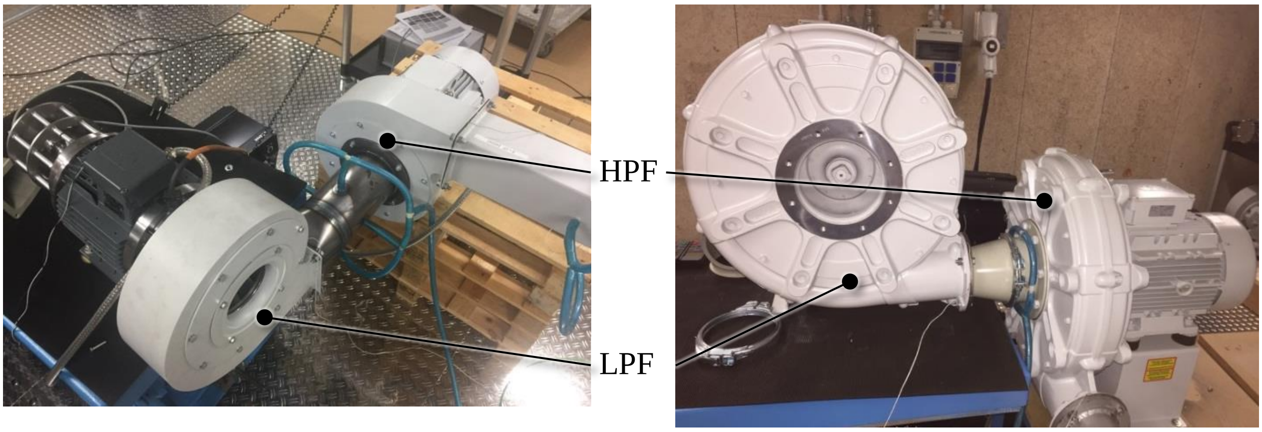

Figure 4: Two different serial arrangements (left: Config. 1, right: Config 2)

Table 2: Technical data of the fans

	Config. 1	**Config. 2**	
Max. volume flow rate	38	70	m³/min
Max. total pressure increase	14 800	14 500	Pa
Engine power	11	15	kW
Rotational speed	11 850	7480	rpm
Impeller diameter	240	378	mm

Figure 5 shows the characteristic curves of the serial arrangements with the head coefficient against the flow coefficient of Configuration 1 and Configuration 2. The red crosses are measurement results on the test rig at different operating points. The described calculation approach is visualized in the blue curve. The black curve represents the calculation of the serial arrangement according to the standard theory based on the addition of the pressure values at a constant flow rate.

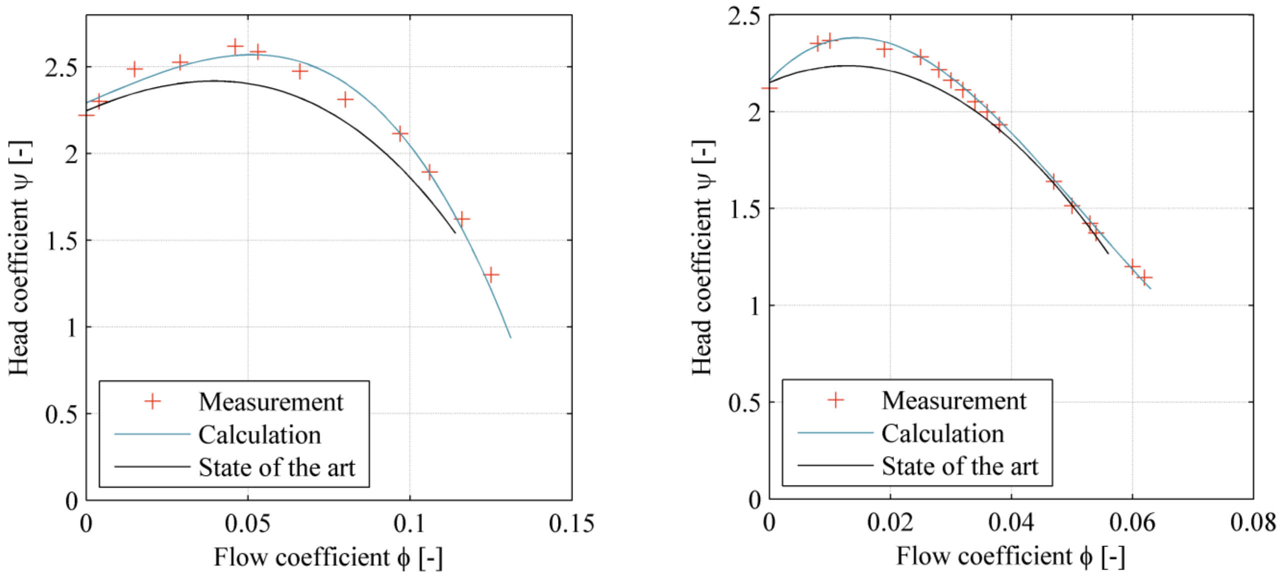

Figure 5: Characteristic curve of Configuration 1 on the left and Configuration 2 on the right

It also can be seen, that the calculation by the state of the art undervalues the head coefficient at many operating points in both configurations. Only at low flow coefficients close to zero and high flow coefficients in the area of the maximum flow coefficient the measurement and the state of the art converge. The proposed calculation approach achieves a higher accuracy compared to the standard calculation approach. Especially in Configuration 2 the calculation approach based on the specific energy predicts the characteristic curves of the serial arrangement precisely. In summary, the proposed calculation is an adequate way of calculation with a higher accuracy compared to the standard method according to the literature.

In Figure 6 the head coefficients of the individual fans in the serial arrangement at discrete operating points are examined for both configurations. The input for the calculation is a measurement of single fans. So the characteristic curve of the LPF is similar to the measurement of the single fan. The data for the HPF has been obtained from the calculation method above. The head coefficients at the same mass flow rate are labelled with the same number (within one chart). Both configurations show that the deviation between the LPF and HPF increases with an increasing flow coefficient. And the operating point of the HPF is almost always moved to the left on the characteristic curve compared to the LPF. This leads to the conclusion, that in both serial arrangements the LPF and HPF operate at different points. A fitted curve through all the points in the chart represents the characteristic curve of the single fan. Therefore, Figure 6 reveals that at the same mass flow rate (same number of operating point) the LPF has higher flow coefficients compared to the HPF due to the lower medium density of the air.

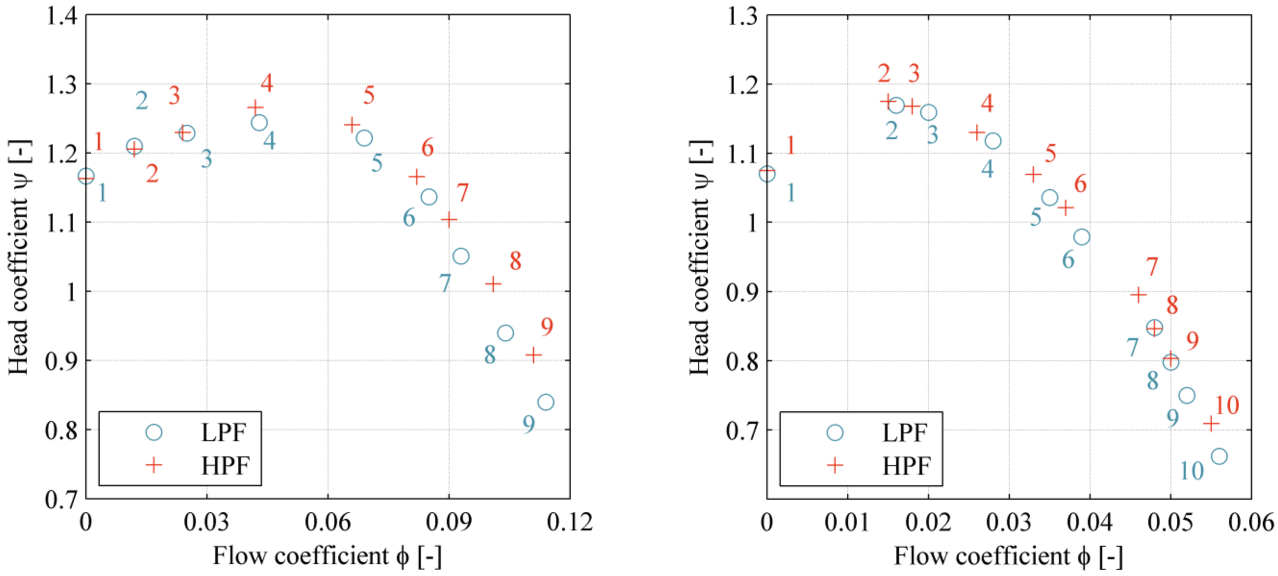

Figure 6: Operating points of LPF and HPF of Configuration 1 on the left and Configuration 2 on the right

CONCLUSION AND OUTLOOK

The current paper introduces a new analytical calculation method for the performance prediction of high pressure centrifugal fans in a serial arrangement. The demand for these investigations is based on the fact that inadequate results were created by calculating the performance of serially arranged fans with the available calculation approach in literature. The proposed method calculates the characteristic curve using the specific work at defined mass flow rates in an iterative process. For validation of the described procedure it has been applied to two different serial arrangements. The new calculation approach achieved a high accuracy in predicting the final performance and therefore the measurement results. Therefore a calculation based on the specific work yields a more precise prediction for the performance of a serial arrangement. This way, the fact that fans in serial arrangements do not operate at the same point due to a compression of the fluid from one fan to the next is taken into account. Thus, at almost every operating points of the serial arrangement, the flow coefficients of the individual fans at a lower stage are always higher than that of the individual fan of a higher stage. Therefore, the focus of further investigations will be placed on the different operating points to clarify these changes via computational fluid dynamics in combination with complementary experiments. Additionally, the new method will be extended to calculate the efficiency of the serial arrangement.

BIBLIOGRAPHY

[1] Radgen, P. – *Market study for improving energy efficiency for fans*. Stuttgart: Fraunhofer IRB Verlag, **2002**

[2] Carolus, T. – *Theoretische und experimentelle Untersuchungen des Pumpens von lufttechnischen Anlagen mit Radialventilatoren*. Dissertation, Karlsruhe, **1984**

[3] Bohl, W. - *Ventilatoren: Berechnung Konstruktion, Versuch, Betrieb*. Vogel Fachbuch, 1. Edition, **1983**

[4] Eck, B. - *Ventilatoren: Entwurf und Betrieb der Radial-, Axial- und Querstromventilatoren*. Springer, 6. Edition, **2003**

[5] Osborne, W. C. – *Fans*. Pergamon Press, 1. Edition, **1966**

[6] Bommes, L. – *Ventilatoren*. Vulkan Verlag, 2. Edition, **2002**

[7] Banzhaf, H.-U. – *Anlagenspezifische Fragen bei Ventilatoren in Reihenschaltung*. Brennstoff, Wärme, Kraft, Nr. 41, Page 82-86, **1989**

[8] Schulze-Dieckhoff, B. - *Das instationäre Betriebsverhalten ungeregelter und geregelter lufttechnischer Anlagen mit einem, zwei in Reihe oder zwei parallelgeschalteten Ventilatoren*. Dissertation, Siegen, **1999**

[9] Nudischer, M. Binz, H. Bachmann, M. Recker, S. - *Experimental Investigation of Two Centrifugal Fans in a Serial Arrangement*. Proceedings of Isromac 2017, Maui, **2017**

[10] Marcinowski, H. – *Die Auswertung von Messungen an Ventilatoren unter Berücksichtigung der Kompressibilität des Fördermittels*. in: Strömungsmechanik und Strömungsmaschinen: Mitteilungen des Instituts für Strömungslehre und Strömungsmaschinen, Karlsruhe, **1973**

[11] EN ISO 5801 - *Industrial Fans : Performance Testing Using Standardized Airways*, **2007**

[12] DIN EN ISO 5167-1 - *Durchflussmessung von Fluiden mit Drosselgeräten in voll durchströmten Leitungen mit Kreisquerschnitt - Teil 1*, **2004**

[13] DIN EN ISO 5167-2 - *Durchflussmessung von Fluiden mit Drosselgeräten in voll durchströmten Leitungen mit Kreisquerschnitt - Teil 2*, **2004**

[14] EN ISO 5168 - *Measurement of fluid flow -- Procedures for the evaluation of uncertainties.* **2005**