A STATISTICAL SURVEY ON THE ACTUAL STATE-OF-THE-ART PERFORMANCE OF RADIAL-FLOW FANS BASED ON MARKET DATA

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SUMMARY

In this paper, a statistical survey on performance of radial-flow fans based on market data is presented. A large number of existing centrifugal fans is analyzed and their overall performance at the best efficiency point, taken from manufacturers' product catalogues, are expressed in non-dimensional form by means of characteristic non-dimensional parameters. The analysis is performed considering definitions for non-dimensional parameters from both English and German literature. Either single- or double-inlet centrifugal fans with backward- or forward-curved blades are considered for the study. The trends outlined for non-dimensional parameters can help designers in the preliminary design of high-efficiency radial-flow fans.

INTRODUCTION

In the current energy and environmental context, sustainability is emerging as the new paradigm of reference. In this global context, each machine that performs an energy conversion must be characterized by the highest possible efficiency. In the field of turbomachinery, little attention has been given by researchers to the challenge of improving the energy efficiency of fans until the early 2000s. This was because their research efforts were primarily devoted to the design and optimization of turbomachinery for energy production and propulsion and also because there was moderate interest from the manufacturers. As pointed out by Radgen [1] in fact, the manufacturers' efforts were directed towards other aspects of fan design, such as the reduction of noise, vibrations, dimensions and costs, and the developed know-how was such that basic designs were generally regarded as being "mature". This adjective does not mean that fan efficiency could not be further improved, but rather underlines the fact that a standard practice existed for their design shape, production technology and use. Therefore, developments of the products and the market were inhibited by this common approach to product design.

At the beginning of the 2000s, Radgen [1] performed a market study in order to highlight the importance of fan systems in different fields such as energy consumption and saving potential. The electricity consumption for fan systems in the European Union was calculated to be in the order of 197 TWh per year [1]. The energy saving potential of fans was estimated to be in the range of (3.5 – 8.3) % [1] depending on fan type and sector. Even if savings would be realized, it was estimated that the electricity consumption would increase to about 220 TWh in 2020 [1]. This is mainly due to the fact that the total usage of fans in all sectors will increase during the period considered. If no new measures were implemented, the consumption would increase to about 290 TWh in 2020 [1].

The Energy-Using Products Directive 2005/32/EC identified industrial fans as a priority product group to be considered for implementing measures to reduce the eco-impact of energy-using products within the European Union. A state-of-the-art analysis of fan technology was made by Radgen et al. [2] in the framework of the aforementioned European Directive. A life-cycle assessment of ventilation fans was carried out, which aimed to assess the environmental as well as the monetary impacts of these products. Furthermore, this study showed improvement potential and suggested suitable political instruments. Eight fan categories were considered in order to reflect the different applications in the market. Based on market data, the authors collected efficiency data for the best efficiency point of a large number of existing fans in the product categories defined.

The Eco-design Directive 2009/125/EC established a framework for the setting of eco-design requirements for energy-related products. It is an important instrument of the Union policy for achieving the energy saving objectives for 2020 and for improving other environmental aspects of products placed on the market or put into service in the European Economic Area. In 2011, with the aim of improving the penetration of high-efficiency industrial fans in the European market, Regulation (EU) No. 327/2011 on eco-design requirements for fans was published. This Regulation sets minimum energy efficiency requirements for fans driven by electric motors with an input power between 125 W and 500 kW.

In order for fan efficiency to meet the eco-design demands set out in the European Regulation 327/2011, advanced design methodologies based on the use of CFD simulations have been applied over recent years to the design of industrial fans [3]. Three-dimensional numerical analyses are primarily employed to optimize the impeller/blade geometry in order to reduce flow separation losses. The preliminary design of fans, however, is generally based on the experience of the designer and makes use of consolidated empirical correlations. At the outset of the design process, in fact, the designer is faced with the basic problem of deciding what type of machine will be the best choice for a given duty. The specific speed Ω is often used to decide upon the choice of the most appropriate machine [4]. The value of Ω gives the designer a guide to the type of fan that will provide the requirement of high efficiency at the design conditions. Moreover, the designer must select the main dimensions of the impeller as a function of the overall required performance, i.e. the volume flow rate Q and the total pressure rise Δp_t. A fundamental guide to the selection of the most appropriate size of fan for a given duty and optimum efficiency is represented by the Cordier diagram. In 1953, Cordier [5] linked the operating conditions Q and Δp_t with the impeller outside diameter D_2 and angular velocity ω, for single-stage machines operating at the best efficiency point, by means of the specific speed Ω and specific diameter Δ. He calculated Ω and Δ for a large number of single-stage fans, compressors and pumps in service and plotted the results in a logarithmic diagram. Cordier noted that experimental data are grouped in this diagram into specific machine types. Axial machines possess high specific speeds Ω and low specific diameters Δ, whereas radial machines are characterized by low Ω and high Δ. Mixed-flow machines are in the range of medium values of Ω and Δ. The author also observed that data lie in a relatively narrow band, which is referred to as the “Cordier line” in the literature [6]. As stressed by Casey et al. [7], the Cordier line is a mean curve based upon experimental results obtained from a large number of machines, so it should not be represented as a single line, but much more like a band of points as there is considerable scatter in the data, especially in the axial region. The Cordier diagram reported by

Dixon and Hall [8] is depicted in Fig. 1a. As explained by Epple et al. [9], from a practical point of view, the Cordier diagram can be used for the preliminary design of a fan in two different ways. On the one hand, given the prescribed operating point of the fan (Q, Δp_t) and a motor drive (i.e. a rotational speed), using the Cordier diagram one can obtain the outside diameter D_2 of the impeller which would meet this operating point at the best efficiency. On the other hand, if a certain operating point (Q, Δp_t) has to be achieved with an impeller of a given outside diameter D_2, the Cordier diagram provides the required rotational speed of the fan to meet this operating point at the best efficiency. Following Lewis [10], an alternative presentation of the Cordier diagram is proposed by Dixon and Hall [8], which expresses the Cordier line data in terms of flow coefficient ϕ and pressure coefficient ψ. In this way, a new and more definite shape of the Cordier line results, depicted in Fig. 1b. Casey et al. [7] showed that the s-shape of the Cordier line and the two distinct parts of the curve in Fig. 1b are caused by the variation in centrifugal effects in the different machine types. In radial machines, almost all the pressure change is due to the centrifugal effects generated by a change in flow radius, whereas these effects are absent in axial machines.

The above considerations explain the reasons for which the Cordier diagram is still today a fundamental tool for the preliminary design of high-efficiency fans. As previously mentioned, the Cordier diagram, as well as most of the empirical correlations routinely employed for the preliminary design of fans, was determined in the middle of the last century upon experimental data obtained from different machine types. In light of the strong boost provided to fan technology by recent European Directives on eco-design demands, this paper presents a statistical survey on the performance of radial-flow fans based on market data. A large number of existing centrifugal fans is analyzed and their overall performance at the best efficiency point, taken from manufacturers' product catalogues, is expressed in non-dimensional form by means of characteristic non-dimensional parameters. For completeness, the analysis is performed considering definitions for non-dimensional parameters from both English and German literature. Either single- or double-inlet centrifugal fans with backward- or forward-curved blades are taken into consideration for the study. The obtained results are then compared with literature correlations, both empirical and analytical, in order to investigate their mutual relations. The trends outlined for non-dimensional parameters can help designers in the preliminary design of high-efficiency radial-flow fans.

ANALYSIS DEFINITION

Considered fan categories

Five categories of radial-flow fans have been defined on the basis of fan type (centrifugal fans with

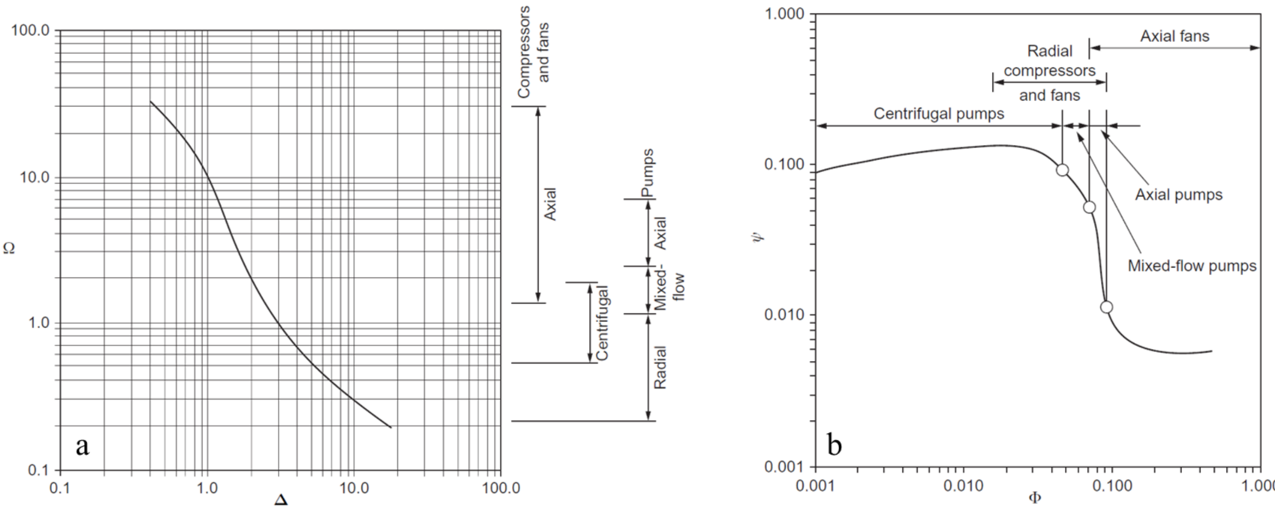


Figure 1: Cordier diagram: a) specific speed Ω Vs specific diameter Δ and b) pressure coefficient ψ Vs flow coefficient ϕ [8]

backward- or forward-curved blades and squirrel-cage fans) and suction configuration (single- or double-inlet centrifugal fans):

- backward-curved single-inlet centrifugal fans (BC SI),
- forward-curved single-inlet centrifugal fans (FC SI),
- squirrel-cage single-inlet fans (SC SI),
- backward-curved double-inlet centrifugal fans (BC DI),
- squirrel-cage double-inlet fans (SC DI).

Figure 2 shows typical impeller geometries for the defined categories of fans. A detailed description of the peculiar characteristics of each considered fan type, in relation to geometry, performance, noise levels, applications, materials and manufacturing technologies, can be found in Bleier [11]. However, additional clarification is needed on the definition of "forward-curved fan" adopted in the present analysis. In literature, in fact, this term denotes a squirrel-cage type fan [11]. In this work, "forward-curved fan" indicates instead a centrifugal fan whose impeller geometry, in terms of inlet-to-outlet diameter ratio D_1/D_2, width-to-diameter ratio at impeller outlet b_2/D_2, blade number and shape, is similar to that of a backward-curved fan, but the blades are curved forward with respect to the direction of rotation.

In order to collect the performance data necessary for the analysis, three different European fan manufacturers have been taken into consideration and the data have been extracted from their product catalogues. The total number of fans analyzed for each of the previously defined categories is reported in Table 1.

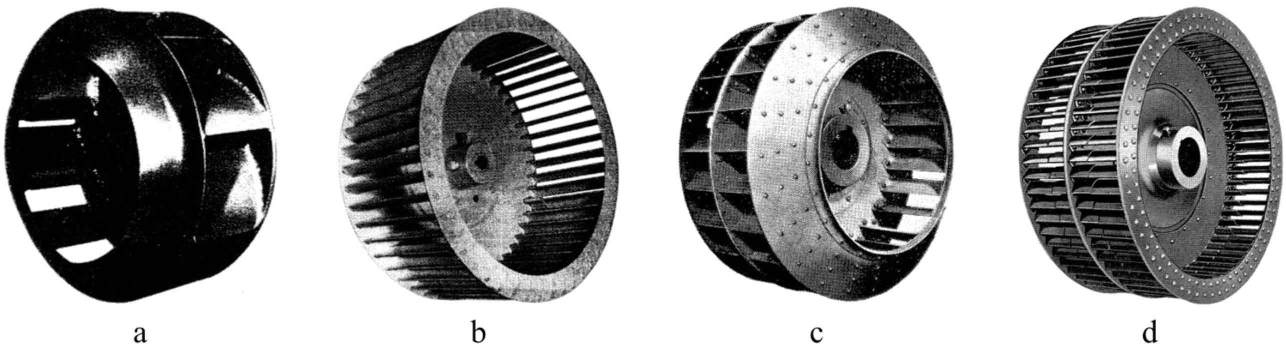
The FC SI fans analyzed belong to the product catalogue of a single manufacturer. In addition to the fan type, the geometry of each fan is characterized by the impeller outside diameter D_2, which is the only relevant geometric feature reported by manufacturers. Regarding fan performance, the following data were collected at the fan best efficiency point: the volume flow rate Q, the total pressure rise Δp_t, the static pressure rise Δp_s, the shaft power P and the rotational speed n.

Non-dimensional performance parameters

Several different definitions can be found in literature for the specific speed and specific diameter. Some of these are not even truly non-dimensional as they make use of inconsistent units. In general,

Table 1: total number of fans analyzed for each defined category

BC SI	FC SI	SC SI	BC DI	SC DI
422	97	63	70	27



a b c d

Figure 2: typical impeller geometries: a) backward-curved single-inlet impeller, b) squirrel-cage single-inlet impeller, c) backward-curved double-inlet impeller and d) squirrel-cage double-inlet impeller [11, 12]

two different strands of literature can be identified with respect to definitions of non-dimensional parameters used to characterize fan performance. Specific speed Ω and specific diameter Δ are mainly used in English literature, as can be seen in the classic books of Csanady [4] and Balje [6]. In German literature, speed number σ and diameter number δ are used instead, since they date back to the original publication of Cordier [5]. These latter definitions are adopted by Eck [13]. Table 2 gives an overview of the different expressions used for non-dimensional performance parameters according to English and German literature and clarifies the relations between "English" and "German" parameters.

In Table 2, German parameters have a subscript "C", indicating that they are defined in accordance with Cordier [5]. Furthermore, U_2 is the blade speed at the impeller outlet, ρ is the fluid density and the rotational speed n is expressed in revolutions per second. For completeness, the present analysis is carried out considering both English and German definitions for non-dimensional parameters. Fan performance data extracted from manufacturers' product catalogues have been expressed in non-dimensional form by means of the above parameters. Tables 3, 4 and 5 summarize the range of variation of these parameters for each defined fan category. The reported range of variation represents useful information for the preliminary design of centrifugal fans.

RESULTS

Cordier diagram analysis

Cordier experimental findings were published by the author in 1953 [5]. Later on, similar experimental analyses were proposed in a number of textbooks on turbomachinery. Typical

Table 2: non-dimensional performance parameters according to English and German literature

Parameter	English literature	German literature	Relation
flow coefficient	$\phi = \dfrac{Q}{\omega D_2^{\,3}}$	$\phi_\mathrm{C} = \dfrac{4Q}{\pi U_2 D_2^{\,2}}$	$\phi_\mathrm{C} = \dfrac{8}{\pi}\phi$
total pressure coefficient	$\psi_\mathrm{t} = \dfrac{\Delta p_\mathrm{t}/\rho}{\omega^2 D_2^{\,2}}$	$\psi_\mathrm{t,C} = \dfrac{2\,\Delta p_\mathrm{t}/\rho}{U_2^{\,2}}$	$\psi_\mathrm{t,C} = 8\psi_\mathrm{t}$
static pressure coefficient	$\psi_\mathrm{s} = \dfrac{\Delta p_\mathrm{s}/\rho}{\omega^2 D_2^{\,2}}$	$\psi_\mathrm{s,C} = \dfrac{2\,\Delta p_\mathrm{s}/\rho}{U_2^{\,2}}$	$\psi_\mathrm{s,C} = 8\psi_\mathrm{s}$
power coefficient	$\lambda = \dfrac{P}{\rho\omega^3 D_2^{\,5}}$	$\lambda_\mathrm{C} = \dfrac{8P}{\pi\rho D_2^{\,2} U_2^{\,3}}$	$\lambda_\mathrm{C} = \dfrac{64}{\pi}\lambda$
total efficiency	$\eta_\mathrm{t} = \dfrac{Q\Delta p_\mathrm{t}}{P} = \dfrac{\phi\psi_\mathrm{t}}{\lambda} = \dfrac{\phi_\mathrm{C}\psi_\mathrm{t,C}}{\lambda_\mathrm{C}}$		
static efficiency	$\eta_\mathrm{s} = \dfrac{Q\Delta p_\mathrm{s}}{P} = \dfrac{\phi\psi_\mathrm{s}}{\lambda} = \dfrac{\phi_\mathrm{C}\psi_\mathrm{s,C}}{\lambda_\mathrm{C}}$		
specific speed speed number	$\Omega = \dfrac{\omega Q^{1/2}}{(\Delta p_\mathrm{t}/\rho)^{3/4}} = \dfrac{\phi^{1/2}}{\psi_\mathrm{t}^{\,3/4}}$	$\sigma = 2\sqrt{\pi}\,\dfrac{nQ^{1/2}}{(2\,\Delta p_\mathrm{t}/\rho)^{3/4}} = \dfrac{\phi_\mathrm{C}^{\,1/2}}{\psi_\mathrm{t,C}^{\,3/4}}$	$\sigma = \dfrac{1}{2^{3/4}\sqrt{\pi}}\Omega$
specific diameter diameter number	$\Delta = \dfrac{D_2(\Delta p_\mathrm{t}/\rho)^{1/4}}{Q^{1/2}} = \dfrac{\psi_\mathrm{t}^{\,1/4}}{\phi^{1/2}}$	$\delta = \dfrac{\sqrt{\pi}}{2}\,\dfrac{D_2(2\,\Delta p_\mathrm{t}/\rho)^{1/4}}{Q^{1/2}} = \dfrac{\psi_\mathrm{t,C}^{\,1/4}}{\phi_\mathrm{C}^{\,1/2}}$	$\delta = \dfrac{\sqrt{\pi}}{2^{3/4}}\Delta$

Table 3: range of variation of non-dimensional performance parameters for each defined fan category: English parameters

	Ω	Δ	ϕ	ψ_t	ψ_s	λ
BC SI	0.178 – 2.081	1.620 – 13.450	0.002 – 0.113	0.085 – 0.184	0.073 – 0.174	0.001 – 0.014
FC SI	0.118 – 0.512	4.552 – 20.567	0.001 – 0.021	0.165 – 0.209	0.130 – 0.194	0.001 – 0.005
SC SI	1.014 – 1.298	1.496 – 1.812	0.166 – 0.237	0.264 – 0.328	0.221 – 0.289	0.075 – 0.114
BC DI	1.670 – 2.171	1.372 – 1.838	0.096 – 0.179	0.104 – 0.121	0.097 – 0.115	0.012 – 0.024
SC DI	1.396 – 1.567	1.119 – 1.295	0.330 – 0.455	0.298 – 0.328	0.264 – 0.290	0.160 – 0.244

Table 4: range of variation of non-dimensional performance parameters for each defined fan category: German parameters

	σ	δ	ϕ_C	$\psi_{t,C}$	$\psi_{s,C}$	λ_C
BC SI	0.060 – 0.698	1.708 – 14.176	0.006 – 0.288	0.680 – 1.469	0.581 – 1.394	0.016 – 0.285
FC SI	0.039 – 0.172	4.798 – 21.675	0.002- 0.053	1.324 – 1.673	1.042 – 1.552	0.009 – 0.106
SC SI	0.340 – 0.435	1.577 – 1.910	0.421 – 0.604	2.114 – 2.622	1.766 – 2.313	1.528 – 2.330
BC DI	0.560 – 0.728	1.446 – 1.937	0.243 – 0.455	0.828 – 0.969	0.777 – 0.920	0.254 – 0.499
SC DI	0.468 – 0.526	1.180 – 1.365	0.840 – 1.159	2.383 – 2.625	2.112 – 2.317	3.251 – 4.967

Table 5: range of variation of total and static efficiencies for each defined fan category

	η_t	η_s
BC SI	0.507 – 0.897	0.498 – 0.834
FC SI	0.372 – 0.775	0.336 – 0.726
SC SI	0.579 – 0.744	0.495 – 0.641
BC DI	0.706 – 0.858	0.673 – 0.816
SC DI	0.568 – 0.724	0.512 – 0.647

examples are the books of Eck [13] on fans and Balje [6] on various types of turbomachines. Several authors have tried to define an analytical relationship for the Cordier line. A short review of published equations for the Cordier line can be found in [7]. On the basis of data from mixed-flow fans and pumps, Casey et al. [7] developed a new equation for the Cordier line for mixed-flow compressors, in order to provide some useful guidelines for their preliminary design. Focusing on radial- and axial-flow fans, Epple et al. [9] proposed a theoretical derivation of the Cordier diagram by applying basic fluid mechanics considerations to process the overall performance of such machines. Willinger [14] gave a theoretical interpretation of the Cordier lines for squirrel-cage and cross-flow fans based on velocity triangles and energy transfer considerations. The same approach has been recently applied by Willinger and Köhler [15] to the theoretical derivation of the Cordier line for axial-flow fans.

Figure 3 reports the distributions of collected data on the Cordier diagram for the defined fan categories. Separate graphs have been realized for backward- and forward-curved fans (Fig. 3a) and squirrel-cage fans (Fig. 3b), respectively, since they occupy different regions of the Cordier diagram. For the sake of comparison, the Cordier lines provided by Balje [6] and Eck [13] have

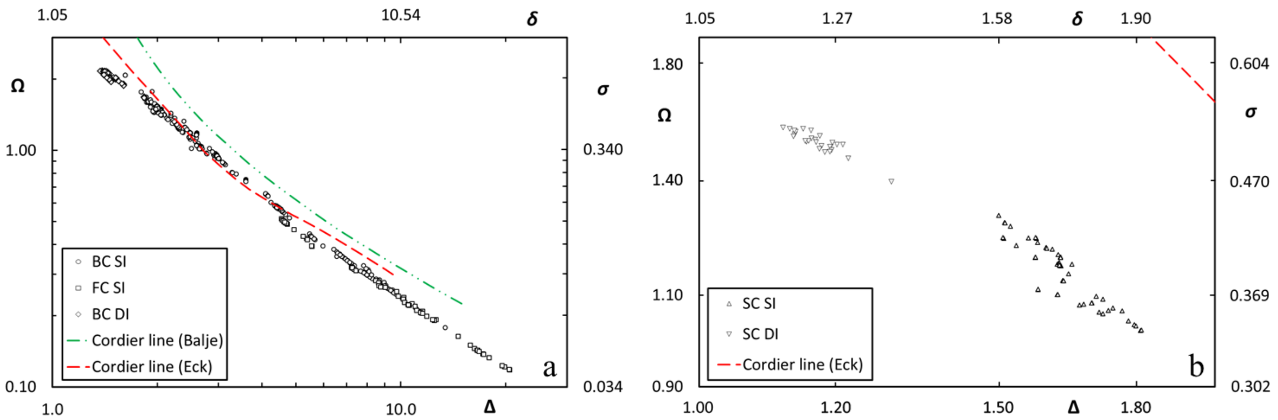


Figure 3: distributions of collected data on the Cordier diagram for the defined fan categories: a) BC SI, FC SI and BC DI fans and b) SC SI and SC DI fans

been superimposed to the data in Fig. 3.

As pointed out by Willinger [14], the combination of specific speed and specific diameter for squirrel-cage fans (Fig. 3b) deviates considerably from that of common radial-flow fans (Fig. 3a). Two different reasons have been identified by the author. The first reason is that squirrel-cage fans have forward-curved blades performing relative flow angles at the impeller outlet $\beta_2 < 90°$. This is in contrast to traditional centrifugal fans with backward-curved blades resulting in $\beta_2 > 90°$. The second reason is their relatively high width-to-diameter ratio at the impeller outlet b_2/D_2. For squirrel-cage fans, the maximum value is $b_2/D_2 \approx 0.50$ compared to a much lower value $b_2/D_2 \approx 0.25$ for common radial-flow fans with backward-curved blades [14]. Both reasons are responsible for the fact that squirrel-cage fans show higher specific speeds and lower specific diameters with respect to traditional centrifugal fans with backward-curved blades. Therefore, squirrel-cage fans can be used for high volume flow rates at relatively low impeller outside diameters.

As can be noted from Fig. 3a, the agreement between collected data and Cordier lines from Balje [6] and Eck [13] is not so fitting, especially for the Cordier line provided by Balje [6]. This may be due to the fact that the Balje curve was obtained from experimental tests on various types of machines, whereas the Eck curve was derived specifically from fan data. In general, the data show lower specific speeds and specific diameters compared to Cordier lines. With regards to this evidence, it is possible to make two different considerations. On the one hand, given the prescribed operating point of the fan (Q, Δp_t) and a motor drive (i.e. a rotational speed), according to data trend, a lower impeller outside diameter D_2 is required to meet this operating point at the best efficiency. On the other hand, if a certain operating point (Q, Δp_t) has to be achieved with an impeller of a given outside diameter D_2, in accordance with data trend, a lower fan rotational speed is required to meet this operating point at the best efficiency.

Regarding the double-inlet centrifugal fans analyzed (i.e. BC DI and SC DI), the values of specific speed and specific diameter shown by these fans are consistent with their peculiar characteristics. As stated by Bleier [11] in fact, with respect to a single-inlet centrifugal fan of the same size, a double-inlet fan delivers about 1.9 times the volume flow rate against the same static pressure rise, while consuming approximately double the shaft power.

Flow coefficient, pressure coefficient and power coefficient analyses

Figures 4, 5 and 6 report the distributions of collected data on the specific speed-flow coefficient, specific speed-total pressure coefficient and specific speed-power coefficient diagrams, respectively, for the defined fan categories. Separate graphs have been realized for backward- and forward-curved fans (Figs. 4a, 5a and 6a) and squirrel-cage fans (Figs. 4b, 5b and 6b). For the sake

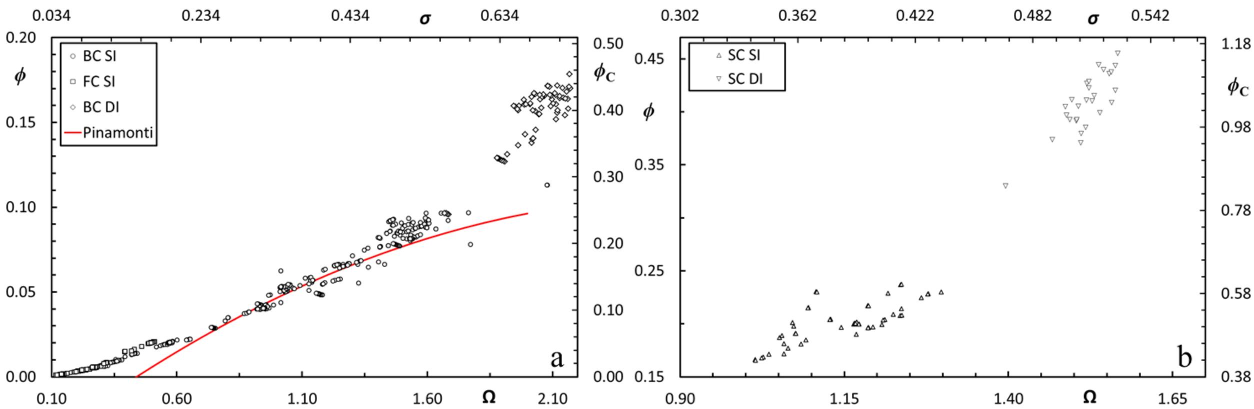


Figure 4: distributions of collected data on the specific speed-flow coefficient diagram for the defined fan categories: a) BC SI, FC SI and BC DI fans and b) SC SI and SC DI fans

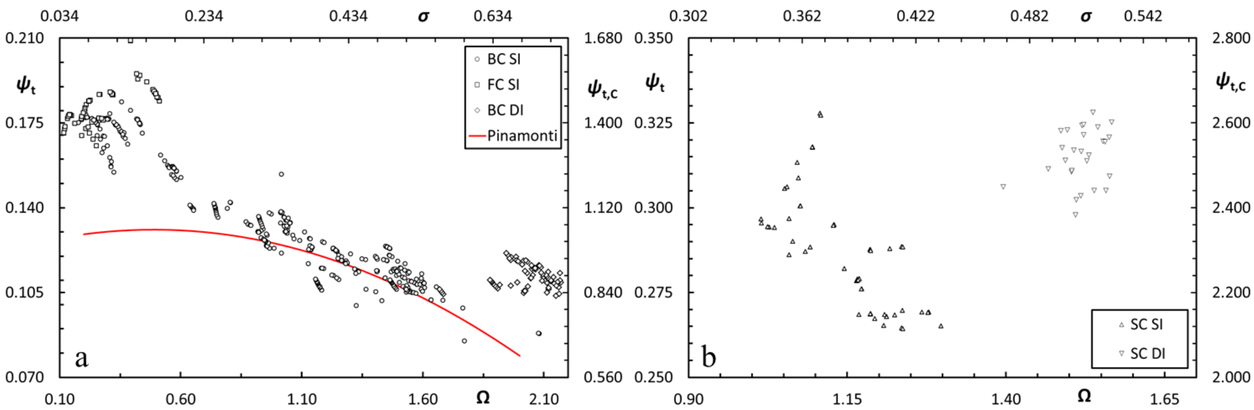


Figure 5: distributions of collected data on the specific speed-total pressure coefficient diagram for the defined fan categories: a) BC SI, FC SI and BC DI fans and b) SC SI and SC DI fans

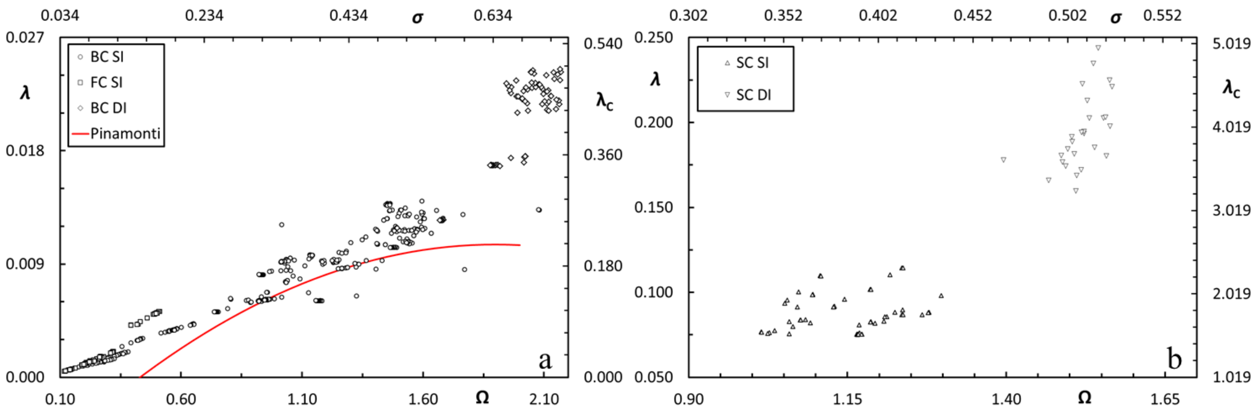


Figure 6: distributions of collected data on the specific speed-power coefficient diagram for the defined fan categories: a) BC SI, FC SI and BC DI fans and b) SC SI and SC DI fans

of comparison, the quadratic regression curves provided by Pinamonti [16] have been superimposed to the data in Figs. 4a, 5a and 6a. Pinamonti [16] performed a statistical investigation on the performance and dimensions of high-efficiency backward-curved single-inlet centrifugal fans and obtained several regression curves, both linear and quadratic, that fit the data.

As can be observed from Figs. 4a, 5a and 6a, the data distributions for BC SI fans are in agreement with the Pinamonti curves. Therefore, though Pinamonti's work dates back to the 1980s, his regression curves are still representative of the performance of high-efficiency BC SI centrifugal fans and can be used for the preliminary design of this fan type.

Figure 7 reports the distributions of collected data on the specific speed-total efficiency diagram for the defined fan categories. Separate graphs have been realized for backward- and forward-curved fans (Fig. 7a) and squirrel-cage fans (Fig. 7b), respectively.

From Fig. 7a, it can be noted that most of the backward-curved centrifugal fans analyzed (both single- and double-inlet) present peak total efficiencies ranging from 0.70 to 0.90. Their peak efficiencies η_t decrease as the specific speed decreases, with values almost equal to 0.50 for the lowest specific speeds. The FC SI fans are characterized by the lowest specific speed values and the corresponding peak efficiencies η_t range from about 0.35 to 0.75. Finally, peak efficiencies η_t of most of squirrel-cage fans (both single- and double-inlet) range from about 0.60 to 0.70 (Fig. 7b). Their peak efficiencies, in fact, are somewhat lower than those of other types of radial-flow fans in the same size class.

Final analyses

Figures 8 and 9 report the distributions of collected data on the flow coefficient-total pressure coefficient and flow coefficient-power coefficient diagrams, respectively, for the defined fan categories. Separate graphs have been realized for backward- and forward-curved fans (Figs. 8a and 9a) and squirrel-cage fans (Figs. 8b and 9b).

Figure 9a shows a common linear trend for power coefficients of backward- and forward-curved fans, both single- and double-inlet. The same linear behavior can be detected also for squirrel-cage fans (Fig. 9b). In this case, however, the slope of the power coefficient trend is different from that of backward- and forward-curved fans.

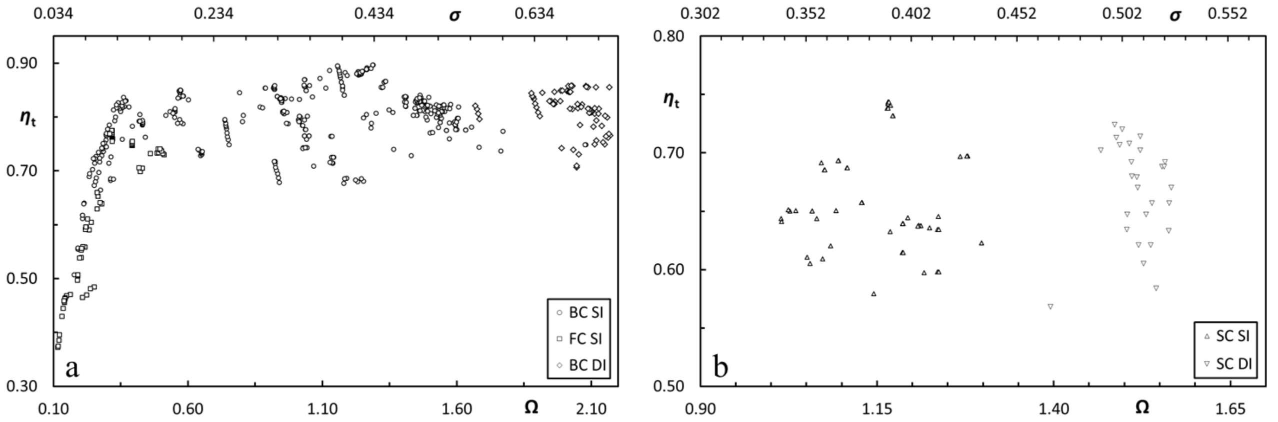


Figure 7: distributions of collected data on the specific speed-total efficiency diagram for the defined fan categories: a) BC SI, FC SI and BC DI fans and b) SC SI and SC DI fans

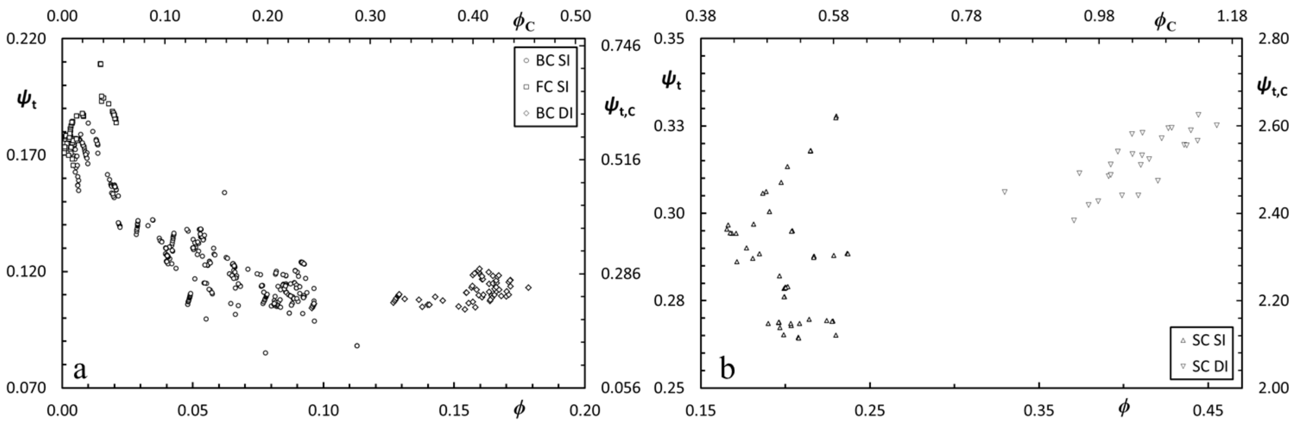


Figure 8: distributions of collected data on the flow coefficient-total pressure coefficient diagram for the defined fan categories: a) BC SI, FC SI and BC DI fans and b) SC SI and SC DI fans

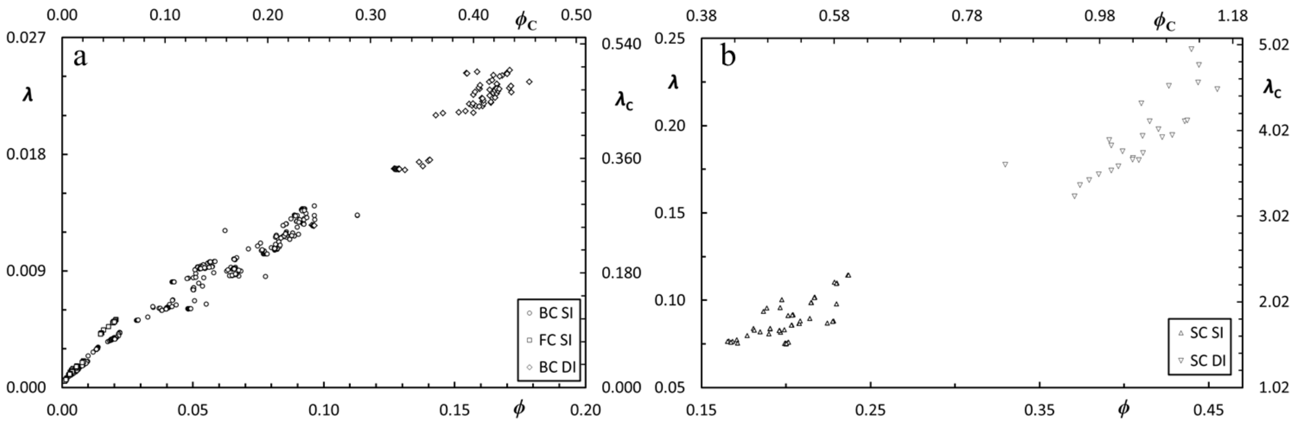


Figure 9: distributions of collected data on the flow coefficient-power coefficient diagram for the defined fan categories: a) BC SI, FC SI and BC DI fans and b) SC SI and SC DI fans

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