Joint impeller/scroll sizing of squirrel cage fans using alternative non-dimensional head and flow rate coefficients.

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Abstract: - The published performance results of squirrel cage fans with known casing sizes are reviewed and some new performance data are reported. The effects of volute size on the flow and performance of squirrel cage fan is outlined. It is concluded that average casing diameter is a relevant length scale for squirrel cage fans which combines both rotor and overall fan dimensions with superior performance and aerodynamic flow inside the fan. This is then employed to define new non-dimensional head and flow rate coefficients. The performance curves of squirrel cage fans, plotted with these non-dimensional co-ordinates, reveal that there is a joint optimum for impeller diameter and the volute spread angle. This optimum volute angle, sizes the impeller diameter in a given overall fan volume such that maximum head and efficiency will be achieved.

Key-words: - squirrel-cage fan, casing size, optimum design, non dimensional coefficients.

NOTATION

b	impeller width
B	volute width
D	diameter
N	rotational speed
p	pressure
Q	volumetric flow rate
r	curvature
Re	machine Reynolds number = ND2/ν
s	inlet gap
Z	number of blades
α_s	volute spread angle
β	blade angle
γ	cut-off angle
δ	cut-off clearance
ϕ	flow coefficient
ρ	density
ψ	head coefficient
υ	kinematics viscosity

Subscripts

1	impeller inner diameter
2	impeller outer diameter
c	casing based
c-o	cut-off
n	inlet nozzle
t	total
s	static

1 Introduction

1.1 The fan and its flow field

The squirrel cage fan is slower than other fans for the same flow rate, or is smaller for the same head. This feature plus simple construction and low manufacturing cost, makes it very popular in comfort industry and house hold appliances. A schematic of the fan is given in fig. 1. This fan is forward curved centrifugal, with typical impeller diameters that range from 4 to 80 cm. The width to diameter ratio is about 0.5. The impellers have 30-40 short chord blades that are manufactured from thin sheet plates.

The flow enters the fan axially from the shroud side and gradually turns radial. Even the best available configuration gives poor flow guidance and inefficient energy exchange inside the fan. High angle of incidence, the mismatch between the inlet and the impeller and non-uniform flow at impeller inlet, result in extensive unsteady separation over the blade suction side and inside the volute [1]. From the shroud to one third of the impeller width, there are regions of two and three dimensional separation with no through-flow [2-4]. At the other two thirds, through flow actually happens and a jet-and-wake pattern occurs after the impeller [4, 5]. Even at the best efficiency point, the velocity components and their fluctuations in and out of the impeller, depend on axial and circumferential locations and the recirculation around the cut off from the scroll to the impeller [2, 6].

This introduction started with a summary of the flow field inside the fan. It now presents the sizing method for the impeller and the volute and then argues that in this way available performance optimizations do not lead to superior aerodynamic flows.

1.2 Geometric optimization

To date only variations of individual parameters have led to experimental optimization of the squirrel cage fan [7, 8]. Experimental studies either: 1) modify the geometry to enhance performance or 2) try to bridge the gap between geometry and performance through flow field studies. The former approach either aims at superior flow/head production [7] or attempts to reduce noise [6, 9]. These are customer based criteria and it is therefore natural that any other approach should lead to improvements in them.

The approach to squirrel cage fan sizing starts from the impeller diameter and then other dimensions are defined, non-dimensionalized and optimized accordingly. Unlike other centrifugal machines, different performances do not come about through a modification in impeller relative dimensions or blade angles, but with a change of impeller overall size, speed or volute spread angle.

The basic theory of the fan provides a simple guideline for selecting the impeller geometry [10]. It is evident that a one dimensional approach is not able to take the complex three-dimensional flow field with separation into account. A review of the experimental investigations in Germany [7] shows a selection for parameters such as impeller diameter ratio, shroud and cut off geometries. The effects of changes in blade angles, blade numbers, and impeller width to diameter ratio as well as casing width and spiral angle on the fan performance are then studied. A comparison between this recommended impeller configuration and a deviation from it has resulted in inferior performances for the latter [2].

After an optimum configuration for impeller dimensions is decided, a change in the spiral spread angle of the fan from 3 to 11 degrees shows that a larger spiral angle allows a larger flow rate but the maximum efficiency occurs at 5 degree. Seven degrees is the recommendation for a compromise between high efficiencies and a wide operating range [7]. It might be argued that a larger range for flow rate is due to larger allocated space to the fan. If this point is taken into account then a smaller but more efficient spiral angle could be of benefit.

The data associated with the above guidelines matches the recommended geometry and must be renewed for any future modifications. Recent research has shown that a squirrel cage fan could be sensitive to modifications like a removal of the shroud [11] or a change in the inlet nozzle curvature [3]. Work therefore attempts to find sensitive trends and geometries that must be included in any optimization program. These might act differently from fan to fan but at least are certain to be important.

1.3 Performance and aerodynamic flow

The flow field and the generated turbulence translate any geometric modification to a superior/inferior performance or noise characteristic. The line of research that develops from such justification is to study the flow and in this way relate geometric parameters of a fan to its flow field and then make relevant conclusions for noise and performance. This approach could also comply with the Euler energy equation, in which blade angles control developed head and performance.

The literature on squirrel cage fan research does not link the above sequence. Although there is an advantage that for a constant impeller diameter, a larger volute spread angle produces more flow [7], but flow field studies do not show that better aerodynamics ensue. The first evidence for this statement is from measurements of slip factor for squirrel cage fans which is an important parameter in theoretical performance prediction of centrifugal machines [11]. The calculated local slip factor from measured velocity components at the impeller exit for two different fan sizes showed larger values of slip factor for fans with smaller volute spiral angles. This means that for a large value of volute spread angle which is superior in producing larger flow rates, the short blade chord of squirrel cage fan is less able to guide the flow.

The second evidence is that the performance of a squirrel cage fan with larger volute spread angle (and therefore a larger flow rate) is less sensitive to modifications to the inlet shape or the impeller inlet spacing. The research that leads to this conclusion starts from a study of the flow inside the volute of a squirrel cage fan which showed there is a stationary stall cell downstream of the inlet separating zone [5].

A suggestion was that an outward curved inlet nozzle could remove this stall cell. The effect on the velocity profile for the inlet to a small volute of 3 degrees spiral angle was examined. The outward curved inlet nozzle resulted in a uniform flow at the impeller exit with no inlet flow separation that is commonly observed for the case of inward curved nozzles [3]. This improvement did not come about in subsequent studies with a larger volute spiral angle of 8 degrees [8]. The difference was not resolved then, but the above influence of volute spread angle on fan performance parameters was again repeated in another study [11]. The optimization of other inlet geometries such as the inlet gap, nozzle diameter and its curvature was also effective for smaller volute spread angles [8] and did not occur otherwise [12, 11].

The conclusion is that a selection of the impeller followed by an accordingly sized volute might not result in an aerodynamic flow and therefore is inefficient. Since there is a strong interaction between the inlet, the impeller and the volute, only a joint sizing of the three would result in optimum overall dimensioning of the fan. A condition that opens this point of view is the selection of a fan for an application with overall space limitation. When the total size is defined, two alternatives are:

A fan with a smaller volute spread angle and a larger impeller.

A fan with a larger volute spread angle and therefore a smaller impeller.

The required flow rate could therefore be developed through a larger rotor tangential velocity combined with a small volute spread angle or through a larger volute with a small impeller radius. The selection could only be resolved through comparative rotor/volute sizing of the fan. This is done in this paper through the introduction of a proper length scale that highlights volute as well as impeller sizes. The correct combination of rotor and volute length scales could lead to a more aerodynamic flow, less noise and could produce a more efficient fan, with no loss of the volumetric flow rate. Alternative flow and head coefficients are then introduced that include the total size of the fan, and their relevance and ability to compare the performance of different designs of squirrel cage fans are shown.

2 Casing based non-dimensional coefficients

It is common to employ rotor based non-dimensional coefficients to compare different designs of a fan type. The flow rate and head coefficients for squirrel cage fans are then defined as follows:

$$f = \frac{Q}{pD_2 b(pND_2)} \tag{1}$$

$$y = \frac{\Delta p}{\frac{1}{2} r (pND_2)^2} \tag{2}$$

The impeller diameter appears as a length scale in the reference velocity and the reference flow area. There is no doubt that this diameter is a representative length scale, especially for axial flow type machines, as it defines the effective size of the inlet, the impeller and the outlet channels.

This was argued in the previous section that in squirrel cage fans the volute geometry and the impeller diameter have comparable effects. Nevertheless, when squirrel cage fan performance is compared through the above non dimensional flow and head coefficients, volute size are treated as a secondary parameter. A proposal in this paper is to use the geometrical average of the two perpendicular casing diameters, fig. 1, as a new length scale:

$$D_c = \sqrt{D_{c1} D_{c2}} \tag{3}$$

This average diameter represents the sum of the impeller diameter and the average height of volute channel. Table 1 shows a range for impeller to casing diameter ratios for various spiral angles. The assumed cut off clearance to impeller diameter is taken to be 0.1 [13]. The recommended values of impeller width to diameter ratio, and scroll to impeller width ratio are used to find a corresponding casing width to diameter ratio in the third row. This could then be used to select or evaluate the allocated space for fan at an early stage of design.

Casing based non-dimensional coefficients are therefore defined when instead of the impeller diameter; this new length scale is used:

$$f_c = \frac{Q}{pD_c b(pND_c)} \tag{4}$$

$$y_c = \frac{\Delta p}{\frac{1}{2} r (pND_c)^2} \qquad (5)$$

The denominators of these flow rate and head coefficients are proportional to overall fan size, and therefore these non-dimensional coefficients could each be interpreted as the volume effectiveness of the fan. This is a meaningful concept as it compares fan performance curves on an overall size basis. Large φ_c means more flow is displaced and large ψ_c means higher head is developed by the fan in a given space.

3 Application and discussion

It is possible to further develop on these definitions when performance curves of different fans are compared. Fig.2 shows characteristic curves for a number of different fan configurations that are drawn using the common definition of non-dimensional head and flow rates. The test procedure for all these performances was according to B.S. 848 with an open inlet and a ducted outlet [14]. Pitot tube was used in specified locations of the rectangular outlet duct to measure pressure for the calculation of developed head and produced flow rate. Three different duct sizes were constructed to allow for variations in outlet cross section. Other outlets were matched to the ducts with appropriate nozzles/diffusers. A part of these performances were measured during a wide ranging research program on the effects of inlet nozzle geometry on squirrel cage fan [3, 5, 8, 11, and 15]. The specifications of the fans are in tables 2 to 4. All impellers had optimum dimensions [7] and modifications were only in the limits of documented research. These fans had similar inlet nozzles and impeller diameter ratios. Although curves vary from fan to fan but those of c, c-1 and E with a small scroll spread angle group together as they have a similar range for flow coefficient (ϕ). The same occurs for fans with a large scroll spread angle.

Fig. 3 shows the same performance results that are redrawn using the coefficients of equations 4 and 5. In this figure the range of variation of flow coefficient is almost the same, irrespective of the volute spiral angles. It means that total fan size is the controlling parameter for flow rate and therefore in order to correlate the performance data, the coefficient of equation 4 (casing based) is superior to that of equation 1 (impeller based).

Different types of rotor based non-dimensional coefficients are employed in open literature to represent the fan performance. Fig. 4 shows a reproduction of a published data [7] employing coefficients of equations 1 and 2. These data are the only ones on the subject with known volutes sizes. It might be concluded that a wider spread angle has an advantage since it produces a larger flow rate but this overlooks the drawback of a larger size and a less aerodynamic flow. Fig. 5 is the re-production of the same data using the coefficients of equations 4 and 5.

Two conclusions are to be made here:

It is evident that for all volute spread angles of 5 degree and larger the flow coefficients that are based on casing size, fall in a band of up to 0.2. The operational range is within 0.08 to 0.16 and maximum efficiencies are expected to happen at about a flow coefficient of 0.1.

Volute spread angle of 5o offers the largest head and efficiency in the available space. It is important to note that this optimum is achieved with no loss in flow rate since for the same overall space a larger rotor with the same speed would make up for the reduced flow rate of a small volute spread angle. The slip factor out of the impeller would in this case be larger and the performance is more susceptible to geometric optimizations [3, 8].

4 Justification for an optimum volute to impeller diameter ratio

The argument for a joint rotor/volute aerodynamic dimensioning of the squirrel cage fan led to optimum characteristics in the previous section. Although the procedure was for the present impeller, but this could be repeated if, for any research or industrial purpose, a different impeller is selected. A comparison of total head coefficients in fig. 6 shows that the optimum value for volute to impeller ratio is 0.65 to 0.69 that corresponds to a volute spread angle of 4 [15] to 5 degrees [7]. This is the ratio for which, the space effectiveness of energy exchange inside the fan is the highest. The difference in angle (4-5o) could be attributed to the difference in casing to impeller width ratio that was 1.3 for the former and 1.08 in the latter experiments. The optimum extension index of 0.1, that minimizes the fan noise level [12], also corresponds to the above mentioned spiral angles.

It was previously argued that a larger volute spread angle would be advantageous in producing

a larger flow coefficient and an actually higher flow rate but for a bigger allocated space [7]. For squirrel cage fan, the impeller occupies a narrow annular space. The volute spiral angle changes the relative volume of the spiral zone after the impeller to that of the cylindrical inlet zone. When in a given space for the fan, a smaller volute spiral angle is considered, the larger developed head coefficient is primarily due to a larger relative available space left for the impeller. There is also a larger slip factor that results when this relatively smaller spiral angle is employed [11].
For a smaller volute spiral angle than the optimum, the outlet space after the impeller is relatively small. The dominant friction resistance in such a volute blocks the flow, and the re-circulating flow back into the rotor reduces head and efficiency.
When for a given fan space spiral angle is larger than the optimum, the inlet zone would be relatively smaller. This does not reduce the flow into the fan since the performance results show that the casing based flow coefficient has a constant range and the flow rate only depends on the overall size. Higher velocity and vacuum at the entrance of such a fan, results in a wider inlet separating zone and more non uniform velocity profile after the blades. Again such loss making phenomena lead to energy dissipation in the volute and reduces fan head and efficiency.

5 Application to design

The advantage that for all selections of volute spread angle the new non-dimensional flow rate has the same range, leads to another positive point. The maximum head coefficient of equation 4 now occurs at almost similar flow coefficient for different squirrel cage fans. A similar argument is true for maximum efficiency. When the available space is fixed through the total casing size, the impeller speed could be found from equation 4, as the required flow rate is known. The required head and efficiency would then decide the impeller diameter and the volute spread angle. The following procedure is envisaged for selecting the impeller diameter when the available space is a constraint:
Define and evaluate the available space using table 1.
Use the optimum value for the new non-dimensional flow coefficient (0.1 for the present configuration) to select the speed and fulfil the required flow.
Select the optimum volute spread angle.
Find the impeller diameter for the optimum cut-off clearance and impeller diameter ratio.
Evaluate the head from the curves of the rotor in use, and compare it with the required head.
If the required head is lower, a larger spiral angle and smaller impeller diameter can be selected to match the fan with the system.
The details of impeller geometries and the casing geometries could follow the recommendations such as those reviewed in this paper.

6 Conclusion

It is shown that the average casing diameter is a controlling length scale of squirrel cage fan and it may be used to define its corresponding non-dimensional head and flow rate coefficients. The new coefficients convey the space effectiveness of the fan. In this way a better comparison of squirrel cage fan is possible when the overall fan size is a constraint.
It is also shown that for the present optimum impeller and casing configurations, a range of volute spiral angles combined with different rotor diameters can provide the same flow rate for a given overall fan volume. The maximum head and efficiency are achieved when an optimum for impeller diameter (or volute spread angle) is selected in the available space. Therefore the average casing diameter that is related to the impeller diameter for an optimum performance point is able to translate a more aerodynamic flow inside the fan to superior performance parameters.

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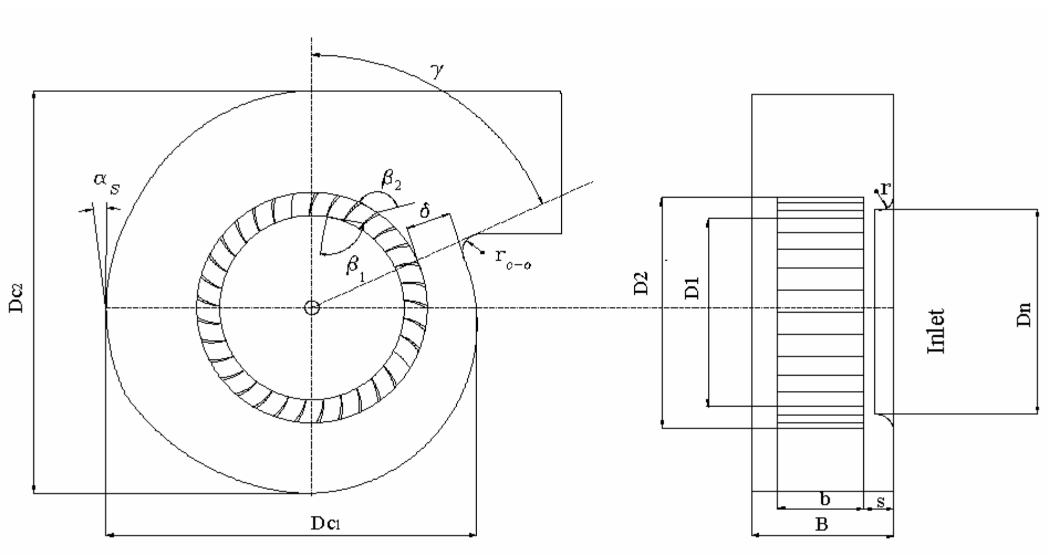


Fig. 1 A general schematic and geometries of squirrel-cage fan

Table 1. Fan dimensions for different spread angles.

so	3	4	5	6	7	8	9	10
D/Dc	0.72	0.69	0.65	0.62	0.59	0.56	0.53	0.5
maximum B/Dc	0.43	0.41	0.38	0.34	0.33	0.29	0.29	0.3

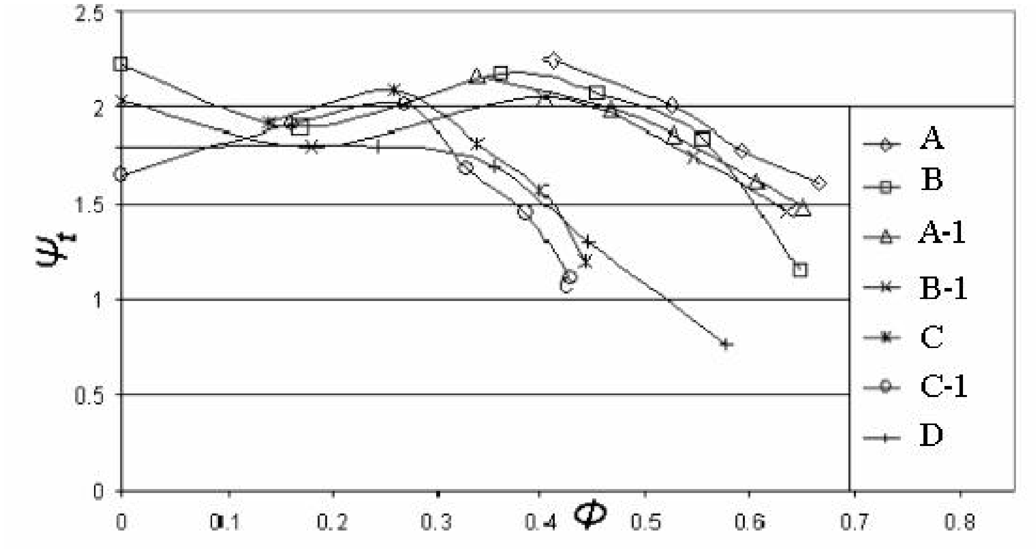


Fig. 2: Grouping of squirrel cage fan performance curves according to volute size using common definitions of head and flow rate coefficients.

Table 2. Common fan specifications

Dr1(cm)	b(cm)	β_1(degrees)	β_2(degrees)	Z	rc(cm)
29	15	85	175	36	1.5

Table 3. Rotor specifications.

	Fan A*	Fan B*	Fan C*	Fan D
Dr2 (cm)	33.5	33.5	33.5	33.5
Re	2.73x105	2.73x105	2.73x105	3x105
N (rpm)	745	745	745	820

Table 4. Casing specifications.

	Fan A*	Fan B*	Fan C*	Fan D
α_s°	9	9	5	8.5
Dc1 (cm)	56	56	46	53.9
Dc2 (cm)	70	70	53	64.2
Dc (cm)	62.6	62.6	49.4	58.8
B(cm)	19.5	19.5	21	21.5
Dn(cm)	29	29	29	30
δ(cm)	3.5	3.5	3.5	3.5
γ°	65	65	65	76
s(cm)	3	1	4	3

* suffix 1 in fan designations in different figures, refers to the same configuration but with adjustable shroud.

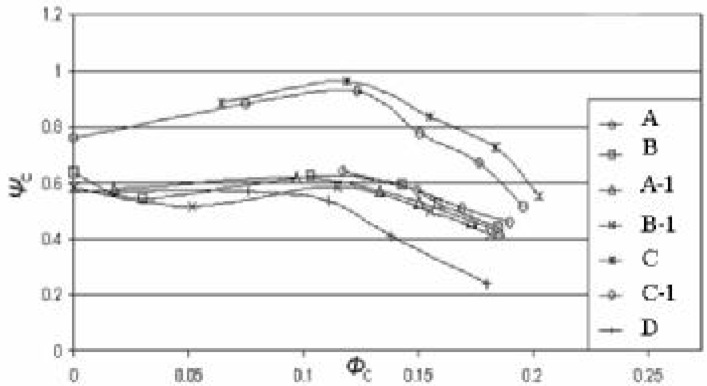
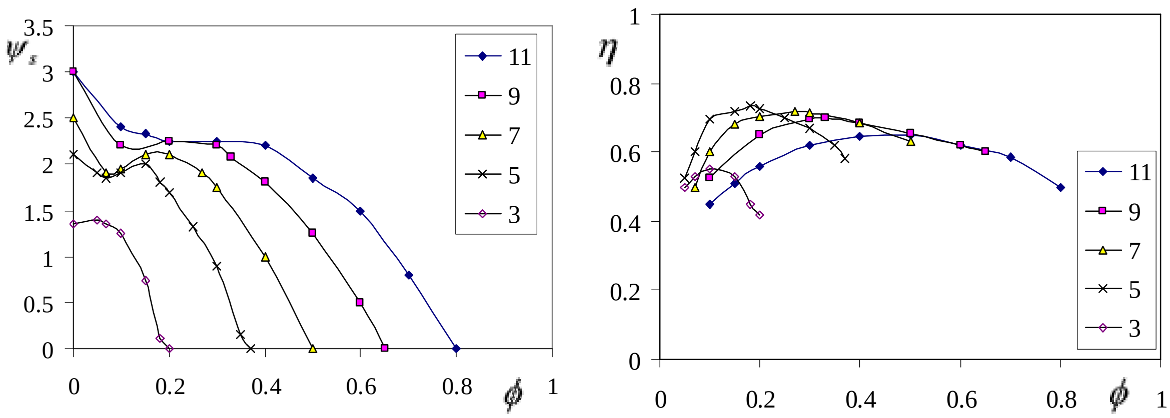
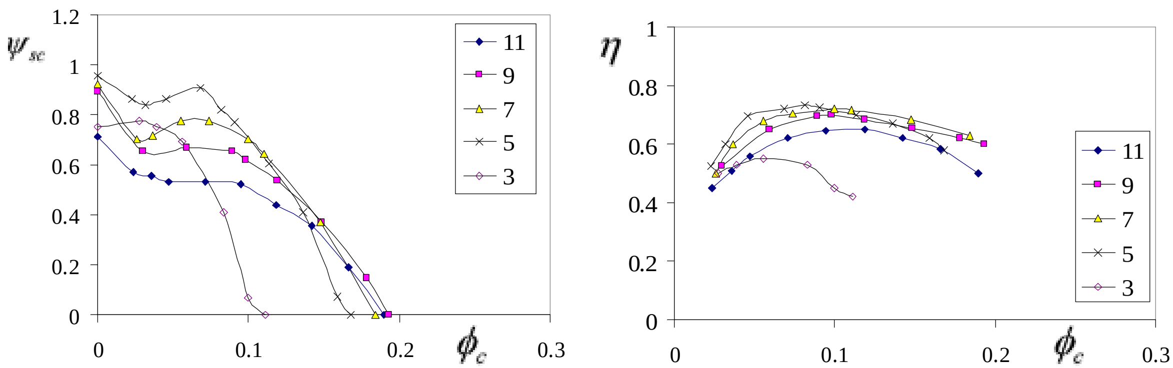


Fig. 3: Grouping of squirrel cage fan performance curves according to volute size using casing based non dimensional coefficients



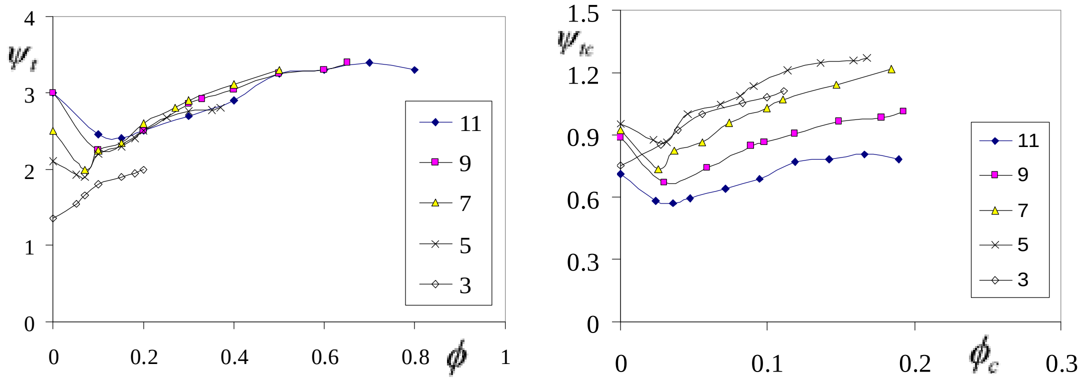
(a) (b)

Fig. 4: Fan performance curves for different volute spread angles (degrees) using common definition of the flow coefficient. [7]. a): static head coefficient. b): efficiency.



(a) (b)

Fig. 5: A comparison between squirrel cage fan performance curves using casing based non dimensional coefficients. a) static head. b) efficiency.



(a) (b)

Fig. 6 The advantage of the new definition of non dimensional head in showing an optimum for the volute spread angle. a) rotor based total head coefficient [7]. b) casing based total head coefficient.