

DESCRIPTION

5 The present invention relates to an axial ventilator in accordance with the preamble of claim 1.

10 Such axial fans are used in a preferred manner for cooling electrical and electronic equipment and installations, particularly where very compact fan dimensions and low noise with high air output are required. The dimensions of the fans which can be used in such equipment are limited by the available structural space. Thus, it is not possible to increase the dimensions in order to increase the output of such fans.

20 An axial fan of the aforementioned type is known (DE-OS 29 40 650), in which relatively high air outputs are obtained in that on the intake side the impeller hub is provided with a ring surface tapering towards the end face. The housing shell is cylindrical in the vicinity of the axial median plane while leaving a small gap to the fan blades and widens to a square shape by walls sloping in the corner areas both towards the intake side and towards the outlet side and

extending symmetrically to the axial median plane.

05 In view of the fact that in the equipment in question ever smaller electrical and/or electronic components are being increasingly used, the casings or housings of such equipment can also be made ever smaller. This means for the small fans of the type considered here that their efficiency is limited by the opposing dynamic pressures in the installation areas.

10 The object of the present invention is therefore to provide an axial fan, which has a higher efficiency than the known fans of this type and in which this is achieved without increasing the external dimensions of the known axial fans.

15 According to the invention, this object is achieved in an axial fan according to the preamble of Claim 1 in that, with respect to the axial median plane, the housing shell is formed asymmetrically in the corner areas and is formed cylindrically from the axial median plane over a greater distance to the intake side than to the outlet side.

20 It has been discovered that these measures lead to a considerable improvement in the performance, without having to modify the external dimensions of axial fans. It has also been found that the improved action does not occur to any noticeable extent unless the axial length of the conical ring surface corresponds to at least one third of the total hub length. Only the
25 combination of the two features leads to the surprising improvement, which will be further explained below on the basis of graphs. As a result of the aforesaid measures on the intake side a wall is formed over a relatively long path (in the vicinity of the cylindrical configuration of the housing shell), which

surrounds the fan blades in a circular manner even in the corner areas, so that the air drawn in, even in the case of high dynamic pressures on the outlet side, cannot leave the fan blades in a radially outward direction before reaching the half of the air guidance path located on the outlet side.

According to an advantageous further development, the cylindrical portion of the housing shell merges into a rounded-off intake portion at the outer edge area directed towards the intake side. This leads to a wider intake cross section on the intake side, which merges narrowing only gradually into the flow duct portion bounded by the cylindrical part of the housing shell. It is thereby advantageous if the radius of curvature of the rounded intake portion is chosen relatively large and in fact somewhat larger or equal to $1/3$ of the distance between the axial median plane and the intake side.

A similar action can also be obtained if the cylindrical portion of the housing shell merges into the intake side by way of a chamfer. It has, finally, also been found that if the housing shell is cylindrical over the entire distance between the axial median plane and the intake side, a considerable improvement in output can still be achieved.

The invention will now be described and illustrated on the basis of embodiments shown in the drawings, wherein:

Figure 1 is a view of the intake side of an axial fan according to the present invention;

Figure 2 is a sectional view along line II-II through the axial fan of Figure 1 but enlarged with respect to Figure 1;

Figure 3 is a cross-sectional view of a detail of a corner area of another form of embodiment;

05 Figure 4 is a cross-sectional view of a detail of a corner area of another form of embodiment;

Figure 5 is a graph illustrating the curves of the quantity of air delivered against static pressure measured in an axial fan according to the invention and also in an axial fan of the prior art;

10 Figure 6 is a longitudinal axial sectional view, along the angular line E-N in Figure 1, through an axial fan illustrating two modifications with important features, while

Figure 7 is a similar view of a smaller version;

15 Figure 8 is a further modified embodiment;

Figure 9 shows the air output curves for two different rotational speed ranges, as developed by modifying the design and altering the inlet port according to Figures 2, 3 and 4, and

20 Figure 10 is a radial plan view on a rotor blade but in which only a single blade is shown in order to clearly define the blade contour.

25 In Figure 1 an axial fan is designated by reference numeral 1 and which, as a result of its small external dimensions and compact construction, is preferably used for cooling purposes in electronic or electrical equipment. The impeller is installed in a housing shell 2. The fan blades 4 and motor casing 3 can also be seen in Figure 1. The fan shown preferably has a square contour of less than 100 mm side length and an axial length of less than 32 mm. Viewed over

the axial depth, the housing shell 2 has a partial area 5, in which it extends cylindrically and concentrically to the fan axis B. From the cylindrical area 5, the housing shell 2 merges into a square form towards the intake side and also towards the outlet side, mounting holes 7 being provided in the resultant corner areas 6.

The cross section of Figure 2 through half of an axial fan according to the present invention shows that the motor casing 3, serving as a hub, is provided with an annular surface 10 conically tapering towards the intake side 9, wherein the fan blades 4 also extend over this annular surface 10.

The motor is constructed in a manner known per se. The coil ends 13, which are mounted on a ring 12 and which are sealed off at the front by means of a protective cover 14 made from an insulating material, surround the stator 19, which in turn is supported by means of a tubular member 23 and a sliding bearing 22 with respect to shaft 15. The shaft 15 is fixed to the motor casing 3 by means of the bush 11. Externally, the casing 3 is constructed in one piece as a short-circuited rotor motor, whose bars pass in a manner known per se through the laminated plates 16 of the rotor. The fan blades 4 are then fixed to the outer periphery of the motor casing 3, which is constructed as a hub.

The cylindrically extending area 5 of the housing shell 2 delimits the inner space in which the impeller is arranged, in the axial median plane A. Towards the intake side 9, housing shell 2 extends over a length a_2 which is greater than half the distance a_1 from the axial median plane A to the intake side 9. Towards the outlet side 8, the housing shell in the corner areas extends from the axial median plane A with a

sloping wall 17, so that an overall asymmetrical construction with respect to the axial median plane A between the intake side half and the outlet side half of the axial fan results. The cylindrical area 5, in
05 the embodiment of Figure 2, merges into a rounded-off intake portion 18, whose radius of curvature R is approximately one third of the distance a_1 from the axial median plane A to the intake side 9. It has been found that the fan performance can be improved if the cylindrically extending area 5 is advanced relatively far up to the intake side. This can be attributed to
10 the fact that in the construction according to the present invention, the fan blades 4 are surrounded over a relatively large axial area by the cylindrical area 5 of the housing shell 2, so that the air taken up by the fan blades ahead of the cross-sectional narrowing in the axial median plane, when dynamic pressures prevail on the outlet side 8, is unable to avoid this
15 counterpressure by avoiding the movement through the fan blades 4 radially outwards and would thus be forced again toward the intake side, as is the case with the known constructions.

In the embodiment of Figure 3, which also shows a
20 cross section through a corner area of the housing shell, the cylindrical area 5a extends over the entire distance between the axial median plane A and the intake side 9. The output can be increased in this case also with an axial fan, whose cylindrical area 5b merges into the outer edge area towards the intake side 9 by way of a bevel 24 (cf. Figure 4), compared with
25 known axial fans.

Figure 5 shows the characteristic curves 20a and 20b of the novel axial fan compared with characteristic curves 21a and 21b of a state of the art axial fan (DOS 29 40 650); it is apparent from these curves that, particularly in the case of high counterpressures (the

pressure is plotted along the ordinate), a considerable improvement in output can be achieved with the axial fan according to the invention. The curve pairs 20a and 21a as well as 20b and 21b which belong together, here represent comparative measurements carried out with two different impeller frequencies. The curves 20a and 21a are plotted at a frequency of 60 Hz, while the curves 20b and 21b are plotted at a frequency of 50 Hz. The quantity of air delivered is plotted along the abscissa axis.

It has also been found that an even more marked improvement of the characteristics can be obtained at higher speeds, which is apparent from the greater deviations in the curves 20a and 21a plotted at a frequency of 60 Hz, as compared with the deviations of curves 20b and 21b, plotted at a frequency of only 50 Hz and thus at lower fan speeds. Consequently, this effect occurs in particular in high-speed fans, especially direct-current fans having a speed of rotation higher than 3000 rpm. These fans are particularly suitable for cooling and ventilating electrical and electronic equipment, as described above.

The inflow channel according to Figure 2 has a circumferential rounded edge with a relatively large inlet radius R while in the construction according to Figure 3 the inlet radius has a theoretical value equal to zero. Figure 4 illustrates only a small widening angle of the cylindrical surface 5b toward the inflow side in the vicinity of the cone surface 24; it starts, for example, at 1/8 of the axial housing length from the inlet plane 9 and has a value of about 60°.

Figure 6 illustrates a modification of the tapering annular surface 10 of Figure 2.

In the case of Figure 6 the latter is a surface cylindrically offset in axial direction in the vicinity of the closed outer rotor bottom, with a portion 65 reduced diameter d_1 of length l_1 which merges
05 steplike into a portion 66 with the full rotor diameter d_2 . The blades 63 are butt-welded, for example, by capacitor discharge on to a rotor cap 61 which is deep-drawn of soft magnetic metal so that a larger inlet cross section is present in inflow direction 60 (as in the case of the conical annular surface 10 of Figure 2). A rotor pot section 65, which is reduced in
10 diameter, thus acts like a strongly defined inlet cone of the rotor hub or of the flange.

A cylindrical part 67 of the port wall extends toward the intake side from the centre plane A by a distance a_2 which is significantly larger than the
15 remaining axial length of the port wall between its cylindrical part and the intake opening. On the outlet side, the port wall merges into the square corner areas, with a wall 68 enlarging the port. An inclined wall 68 (which is a coaxial cone surface only in special cases) can be replaced by a wall with
cylindrical surfaces, the diameter of which is enlarged
20 in steps towards the outlet surface, as indicated by a wall 69 to form complete corner pockets, whereas in the case of the gradual transition by the inclined corner walls 68, something like a "half-corner pocket" is formed.

It is to be noted that a maximum cross section is
25 available in the inflow area, above all that a cross-sectional enlargement about the rotor hub is effected, and that on the outlet side as large as possible an enlargement of the flow cross section is effected by the port 68 or 69 and the hub portion 66, wherein the ratio of the lengths $a_2 + a_3/2a_1$ should have a minimum value of 0.3 but, preferably, a

value of 0.5. Here a_3 corresponds to the distance between the centre axis A and the start of the widened port of the port wall on the outlet side and $2a_1$, the total axial length of the fan. If, however, for
05 manufacturing reasons, one cannot make the diameter of the rotor, i.e. the outer diameter of the rotor blades 63, sufficiently close to the inner diameter of the cylindrical part 67 of the port wall, the ratio $a_2 + a_3/2a_1$ must be larger; its optimum value may then be about 0.5.

10 The ratio $l_1/2a_1$ has a minimum value of 0.3.

Figure 7 illustrates an embodiment in which the housing with an outer annular wall 74, which merges into square fastening lugs corresponding to the illustration in Figure 1 is formed in one-piece with
15 fastening webs 75 and a flange 76 as well as the bearing tube of parts 77, 78 and 79, it is thus a one-piece plastics injection or compression moulded part, preferably with an interior width enlarged in diameter with respect to the centre part 78 in the vicinity of bearings 72 and 73 for installation of the bearing, because the fan illustrated is driven by a
20 commutatorless d.c. motor, especially a so-called two-pulse motor. Particularly if the fan is very small it is not easy to accommodate the electronics in the motor area, i.e. either in the flange 76 or, as in the case of Figure 7, within the area of the closed rotor bottom 71.

25 Preferably, the motor can be a commutatorless d.c. motor, in particular a 1 or 2-pulse motor with a permanent magnet rotor, wherein the electronic components are mounted for commutation in the vicinity of the motor.

However, with commutatorless d.c. fans with one or

two-pulse operation, one uses particularly few electronic components and the latter can be advantageously accommodated in the drive hub of the fan. At the same time, they produce relatively little heat so that with a commutatorless d.c. motor for the drive of the fan, the bearing tube 77, 78 and 79 can be made of plastics material, particularly with a one or two-pulse commutatorless d.c. motor. The plastics bearing tube has sufficient durability and maintains the tolerances over a long service life because heating from motor losses and the electronics is slight. The plastics bearing tube can be made very advantageously and economically in one piece with the rest of the housing.

Figure 8 illustrates a similar construction to Figures 6 and 7 with stepped pockets 81 and 89 on the inlet and outlet side, with additional sloping corner walls 84 and 88 on the inlet side and outlet side. It is again critical here that a point of transition 85 from the corner wall 84 into a cylindrical wall 87 on the inlet side is at a greater distance from the centre plane A than the point 86 where the transition takes place on the outlet side from the cylindrical wall 87 into the corner wall 88. The stepped recesses which lead to the formation of the corner pockets 81 and 89 are favourable from a manufacturing point of view and assure better adherence to the dimensions of the one-piece plastics housing which, otherwise, is constructed similar to Figure 7; namely, in one piece including the bearing tube and consisting of plastics material.

Figure 9 illustrates clearly that a small widening into the corners on the inflow side, as shown in Figure 4, entails a very advantageous configuration of the curve at a slightly higher pressure requirement while at a specific maximum pressure, the differences from

the different contour of the inflow opening disappear and in the middle pressure range, where the practical applications lie, a relatively large radius of curvature of the circumferential contour (as indicated in Figure 2) clearly still produces an additional improvement.

In particular, it can be seen that the asymmetry of the outer housing, as viewed from the centre plane A, in conjunction with the inlet cone (for example according to Figure 2 or also in the variant according to Figures 6, 7 and 8) is very favourable (difference of curve α from curves β , δ and ξ). The curve α corresponds to a symmetrical housing, wherein a pronounced cone hub was also provided in the inflow area, e.g. as described in Japanese Patent Application 54/133638 or in U.S. Patent 4,373,861.

One obtains generally a further improvement in the characteristic curve and also in the noise level if the contour of the blades 23, respectively 83, 63, 73 or 123 as shown in Figure 10 is so shaped that the angle at the blade root on the inlet side ϵ_i which is formed by the tangent to the blade root on the inflow side (i.e., at the blade surface thereof) and the inflow plane (as also to any of the other planes which are parallel to the inflow plane which may be E, E₁, E₂) is smaller than the angle α_i on the outflow side (at the blade root). The angle of incidence ϵ_a at the radial outside on the inflow side (formed again by the tangent to this blade edge and its angle to the inlet plane) is smaller than the angle of incidence at the radial outer edge on the outflow side α_a . In other words ϵ_i is smaller than α_i and ϵ_a is smaller than α_a , wherein the conditions, as illustrated in Figure 10, are optimal for a fan according to Figure 2, whereas similar conditions apply in the case of Figures 6, 7 and 8; however, i is

approximately equal to α_a in that case.

05 The blade curvature is approximately that of a cylindrical surface. In all of these cases, the angle ϵ_i passes over continuously into the angle ϵ_a and the angle α_i into the angle α_a in the radially directed path of the inlet and outlet edges. The true radial extent of the blades has to be considered here.

10 The values of ϵ_i, ϵ_a preferably lie within the range of between 30° and 50° and values for α_i, α_a within the range of between 40° and 60° , wherein α_i is approximately equal to α_a .

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CLAIMS

1 An axial ventilator, especially a small ventilator or
5 fan of axial compactness, comprising a housing shell (2)
 which surrounds the impeller of the ventilator and the
 inner contour of which is formed cylindrical in the region
 of the axial centre plane (A), and is enlarged to a
10 polygonal, especially square profile circumscribing the
 diameter of the impeller toward the exhaust side (8) and
 toward the intake side (9) while defining corner regions
 (6), and further comprising a central coaxial core which
 is defined by the drive motor, the hub of the impeller and
15 the mounting flange for the drive motor, and which
 includes an outer annular surface tapering toward the
 front side of the intake side, thereby forming an intake
 port which tapers in the flow direction, with the axial
 length of the tapered annular surface being at least equal
20 to 1/3 of the length of the hub, and wherein even with
 relatively high dynamic pressures on the outflow side the
 intake air cannot leave the fan blades radially outwards
 before reaching that half of the air circulation path
 situated on the outflow side, characterized in that in the
 corner regions (6) the housing shell (2) is formed
25 asymmetrically with respect to the axial centre plane (A),
 while it is formed cylindrically (region 5, 5a, 5b) from
 the axial centre plane (A) toward the intake side (9) over
 a long distance (a_2) than toward the exhaust side.

30 2 The axial ventilator according to Claim 1,
 characterised in that the distance (a_2) corresponds to at
 least one-half of the distance ($a_1/2$) between the axial
 centre plane (A) and the intake side (9).

35 3 The axial ventilator according to Claim 1 and 2,
 characterised in that the housing shell (2) changes from



its cylindrical region (5) to a circumferential intake rounding (18) in the outer edge portion facing the intake side (9).

5 4 The axial ventilator according to Claim 3, characterised in that the radius of curvature (R) of the intake rounding (18) is greater than, or equal to, $1/3$ of the distance (a_1) between the axial centre plane (A) and the intake side (9).

10 5 The axial ventilator according to Claims 1 and 2, characterised in that the cylindrical region (5b) of the housing shell (2) merges into a chamfer or bevel (24) in the corner regions (6) toward the intake side (9).

15 6 The axial ventilator according to Claims 1 and 2, characterised in that the housing shell (2) is formed cylindrically across the full distance (a_1) between the axial centre plane (A) and the intake side (9).

20 7 The axial ventilator according the Claim 1, characterised in that the drive motor is a commutator-less D.C. motor, especially a one or two-pulse operated motor with a permanent-magnet rotor, in which the electronic means for commutation is disposed in the vicinity of the motor.

30 8 The axial ventilator according to Claim 7, characterised in that the parallelepipedal housing has, in axial plan view, a square contour of less than 100 mm side length, and an axial length of 32 mm or less.

35 9 The axial ventilator according to Claim 1, characterised in that the central core on the inflow side exhibits a stepped reduction of its outer diameter in the region of the closed rotor bottom surface.



10 The axial ventilator according the Claims 1 or 9, characterised in that the contour of the housing inner wall has, on the outflow side, step-like enlargements defining stepped corner pockets.

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11 The axial ventilator according to Claim 1, characterised in that stepped recesses (81, 89) are provided in the mounting corner portion of the housing on the inflow side and/or outflow side.

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12 The axial ventilator according to Claim 1, characterised in that the angles of incidence ϵ_i ϵ_a on the intake side are smaller than the angles of incidence α_i α_a on the outflow side, with ϵ_i and ϵ_i representing the angles at the vane root on the inflow and outflow sides, respectively, which are defined by the tangents to the vane root and the inflow and outflow planes, respectively, and with α_a and α_a representing the angles of incidence at the radial outer side on the inflow and outflow sides, respectively, which are defined by the tangents to the respective vane edges and the inflow and outflow planes, respectively.

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13 The axial ventilator according to Claim 12, characterised in that ϵ_i' and ϵ_a are in the range of between 30° and 50°, and α_i' α_a are in the range of between 40° and 60°, and that α_i is approximately equal to α_a .

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14 The axial ventilator according to any one of Claims 1 to 13, characterised in that the external rotor bottom (61) included in the central coaxial core includes a cylindrically reduced surface (65) which merges in stepped fashion into a portion (66) of the full rotor diameter.

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15 The axial ventilator according to Claim 14,



characterised in that the external rotor bottom (61) is deep drawn from a soft magnetic material.