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(71) Applicant: **SAME DEUTZ-FAHR S.P.A.**
24047 Treviglio (Bergamo) (IT)

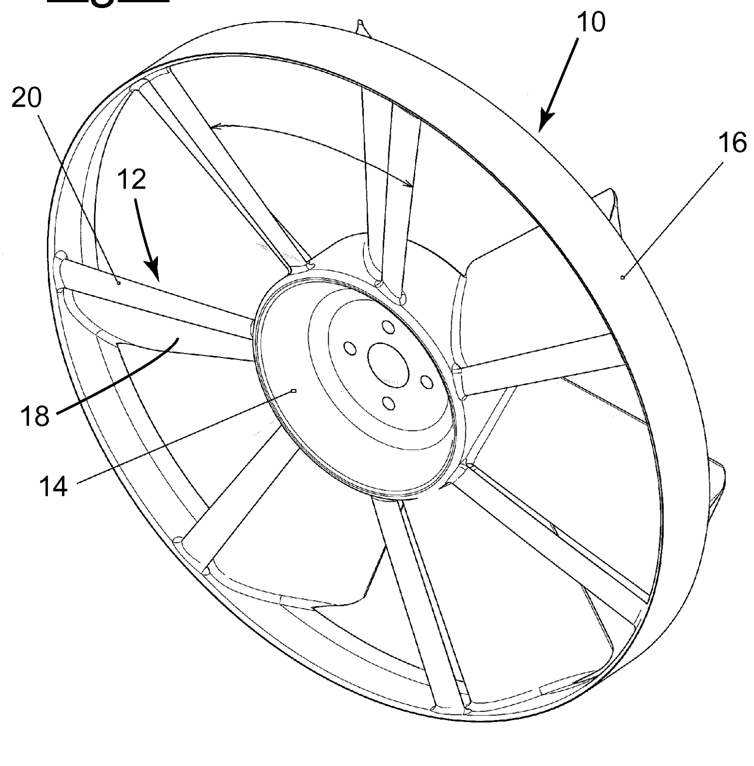
(72) Inventors:
- **Paggi, Bruno**
  24047 Treviglio (Bergamo) (IT)
- **Benzoni, Giuseppe**
  24020 Fino del Monte (Bergamo) (IT)

(74) Representative: **De Gregori, Antonella et al**
Ing. Barzano' & Zanardo Milano S.p.A.
Via Borgonuovo 10
20121 Milano (IT)

(54) **Axial fan for cooling the underhood of a farm machine**

(57) Herein described is an axial fan for cooling the underhood of a farm machine, of the type made up of a rotor (10) provided with a plurality of blades (12) with a wing profile and a substantially radial development constrained against the central hub (14) of the rotor (10) at one of their first ends. Each blade (12) is made up of a first element or main profile (18) and by at least a second element or flap (20) arranged in proximity to the inlet edge (18') of the first element or main profile (18). A blading profile of such type allows pitch setting the blades with considerable angles of attack and obtaining high dynamic values, avoiding the occurrence of separations of the boundary layer from the top layer of the blade profile itself at the same time.

Fig. 1



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Description

[0001] The present invention refers to an axial fan for cooling the underhood of farm machines and earthmovers in general.

[0002] The underhood of modern farm machines, especially referring to the medium-high series, is characterised by the simultaneous presence of several radiating elements not only intended for cooling the engine, but also for cooling the oil present in the transmission and hydraulic system. In addition, the condenser of the cockpit conditioning system is almost always present on the farm machines.

[0003] An element becoming more and more indispensable to allow farm machines equipped with turbocharged engines meet the requirements provided for by the antipollution directives in force is the so-called "intercooler", that is an air/air or air/liquid heat exchanger of a quite considerable size, in that the temperature of the comburant air entering the engine must not exceed the value of 50°C.

[0004] Lastly, another element more and more common on the modern farm machines is the diesel fuel radiator, which is essential to ensure that the power supplied by the engine is the most constant possible, regardless of the high temperatures that the modern internal combustion engines are capable of reaching during their operation cycle.

[0005] Therefore, if on one hand the performances of the farm machines nowadays are decidedly higher with respect to the past, on the other the dissipation of an ever-growing amount of heat developing during the operation of the machine themselves is required. For such purpose, the ventilation of the underhood of the machine is of crucial importance. However, according to experimental tests it was observed that the traditional cooling fans are inefficient, that is they are not capable of processing a sufficient amount of air due to the high pressure drops caused by the underhood.

[0006] Use of higher rotation speed and/or increasing the diameter of the rotors partially improves the situation strictly in terms of the flowing capacity, but to the detriment of silence and absorption of the mechanical power and, thus, of the aeraulic performance of the fan. The performances required of the cooling fans according to the prior art are thus restricted to the operation field of the axial fans.

[0007] Use of families of turbomachines different from the axial ones, such as for example the radial ones or the mixed flow ones, gives rise to distribution of flow speeds, both when suctioning and discharging, of decidedly low advantage. In particular, detected in the suctioning section are high values of air speed (and thus pressure drop) concentrated in proximity to the periphery of the radiating pack, arranged upstream of the fan, while the central zone of the radiating pack itself is barely crossed by the air-flow, thus leading to the drop of the heat exchange efficiency. Downstream the fan, the high values of the radial speed component upon discharge complicate the evacuation of hot air from the underhood and cause further pressure drops which weigh, in turn, on the fan. Lastly, further problems related to the use of fans of the abovementioned types are linked to the large axial dimensions, greater manufacture difficulties and total costs.

[0008] Therefore, an object of the present invention is that of overcoming the problems of the prior art, by providing a fan of the axial type for cooling the underhood of a farm machine, capable of providing better performances, considering the same air-flow, with respect to fans of the traditional type.

[0009] Another object of the invention is that of providing an axial fan for cooling the underhood of a farm machine, capable of containing - during its operation - the occurrence of turbulent phenomena in the cooling air-flow, with the consequent reduction of the noise and the power absorbed.

[0010] Still, another object of the invention is that of providing an axial fan for cooling the underhood of a farm machine which is simple and inexpensive to manufacture.

[0011] These objects, according to the present invention, are attained by providing an axial fan for cooling the underhood of a farm machine as described in claim 1.

[0012] Further characteristics of the invention are highlighted by the subsequent claims.

[0013] Characteristics and advantages of an axial fan for cooling the underhood of a farm machine according to the present invention shall be clearer from the following exemplifying and non-limiting description with reference to the attached schematic drawings wherein:

figure 1 is a perspective view of an embodiment of an axial fan for cooling the underhood of a farm machine according to the present invention;

figure 2A is a partial plan view of the axial fan of figure 1;

figures 2B and 2C are sectional views, obtained along line A-A of figure 2A, of the blade profile of the fan shown in figure 1;

figure 3 illustrates the trend of the motion field around the blade section indicated in figures 2B and 2C, the distribution of speed having been determined through a CFD (Computational fluid dynamics) simulation on computer by setting, as rotation speed, the one of the project equivalent to 3120 revolutions per minute;

figures 4 and 5 are diagrams showing the comparison between the aeraulic performances (work and efficiency) of an axial fan for cooling the underhood of a farm machine according to the present invention and the ones characterising a fan of the traditional type; and figures 6-11 are diagrams showing the comparative noise measurements between an axial fan for cooling the underhood of a farm machine according to the present invention and a fan of the traditional type.

[0014] Referring to figure 1, shown is an example of an embodiment of an axial fan according to the present invention, indicated in its entirety with reference number 10. The fan 10, made in the form of a disk or rotor, is configured in particular but not exclusively to carry out the cooling of the underhood (not shown) of a generic farm machine, such as for example a farm tractor, a combine harvester or the like.

[0015] The rotor 10 is provided with a plurality of blades 12, with a wing profile and a substantially radial development, constrained against the central hub 14 of the rotor 10 itself at one of their first ends. Furthermore, the rotor 10 has an external circumferential edge 16 constrained against which is the other end of each blade 12.

[0016] According to the invention, each blade 12 is made up of a profile of the multiple type, that is characterised by the presence of a first element or main profile 18 and at least the presence of one further element or flap 20 arranged in proximity to the inlet edge 18', also referred to by the term of aviation derivation "slat", of the main profile 18 itself.

[0017] Figures 2B and 2C both show a sectional view of the blade profile 12, obtained at a distance from the central axis of the rotor 10 equivalent to 1.6 times the chord C in the axial direction of the rotor 10 itself. The chord C is defined as the linear distance between the inlet edge 20' of the flap 20 and the outlet edge 18" of the main profile 18. According to figure 2A it can be observed that, according to a particularly preferred embodiment, the radius of the rotor 10 is equivalent to about two times the length of the chord C, while the radius of the central hub 14 is equivalent to about 75% of the length of the chord C itself.

[0018] As shown in the sections of figures 2B and 2C, the main profile 18 is characterised by a median line 22 having a relative maximum deflection fm equivalent to about 6% with respect to the length of the chord C, wherein the point Xfm of maximum bending is identified at 50% or, in other words, in proximity to the centreline point of the chord C itself. Regarding the thickness Sp, the main profile 18 has a maximum value equivalent to 12% of the length of the chord C in proximity to 30% of the length of the chord C itself, with measurement taken starting from the inlet edge 20' of the flap 20. The good characteristics of the main profile 18 in terms of bending and thickness make it particularly functional and, above all, sturdy, in that stall phenomena occur only in case of particularly negative flow angles of incidence.

[0019] The purpose of the flap 20 is that of extending the field of correct functionality of the main profile 18 with considerably critical angles of attack. Research for elevated work values per mass unit required to the fan 10, within the scope of applications herein referred to, requires very high blade keying angles of the blade profile 12 (35°÷42° at the root of the blade) and, thus, the use of more sophisticated wing sections like the one according to the present invention. The valuable function of "controlling the boundary layer" performed by the blade profile 12 made up of two distinct elements 18 and 20 is not only useful from an aerodynamic point of view, but also from an acoustic point of view (discussed hereinafter), in that the reduction of important turbulent structures at the edge of the outlet 18" of the blade profile 12 has a positive impact on the aero-acoustic behaviour of the fan 10.

[0020] Regarding determining the optimal number of blades 12 and, thus, the best pitch P/chord C ratio of the fan 10 (figure 1), reference was made to Weinig's theory, usually used when designing axial turbomachines such as compressors and fans considerably charged from the aerodynamic point of view. In addition to the abovementioned theory, the optimisation of the pitch P/chord C ratio was also performed in the light of considerations based on the results of previous experiments regarding the acoustic effects linked to the number of blades 12 of the machine 10.

[0021] On the other hand, figure 3 allows showing the considerable acceleration received by the air-flow when crossing the space comprised between the two wing elements. Such increase of speed allows the flow along the side in depression to advance, without any risks of separation, towards the outlet edge 18". As a matter of fact, it is clear that the trail zone (at a low speed) is slightly thicker than the outlet edge itself, to the advantage of the aeraulic performances. Furthermore, the low turbulence of the flow upon discharge is an indicator of low noise as a consequence of a lower dissipation of energy to the detriment of the processed flow.

[0022] Figures 4 and 5 illustrate the distributions of the work and efficiency, depending on the amount processed, of a traditional rotor with respect to the rotor 10 according to the present invention. The results observed from the acoustic calculations carried out shall be discussed hereinafter. In particular, figure 3 shows the trend of the coefficient of static work Ψ depending on the flow coefficient Φ. Such parameters respectively describe the rise of static pressure and the volumetric flow nondimensionalized, that is:

$$\Psi = \frac{\Delta P_S}{\rho n^2 D^2} \qquad \text{and} \qquad \Phi = \frac{Q}{nD^3}$$

wherein:

ΔP_s = static pressure difference [Pa]
ρ = air density [kg/m^3]
n = revolutions per second
D = external diameter [m]
Q = volumetric flow [m^3/s]

[0023] In detail, figure 4 allows comparing two different distributions of parameters Ψ and Φ defined beforehand, according to the dimensions measured through experimental surveys carried out on the fan 10 according to the invention and on a fan of the conventional type but with good performances. As well visible in figure 4, the characteristic curve of the new fan 10 indicates better performances with respect to the ones of the traditional fan, with differences in terms of work (and thus in terms of static pressure) whenever higher that 20% considering the same amount of processed flow.

[0024] Major differences between the fan 10 according to the invention and a fan of the conventional type are also visible in figure 5, which shows a comparison in terms of total efficiency and from which it can be observed that the margin between the performance curves of the two fans is close to ten points percentage at the point of maximum efficiency. Also in the two diagrams of figures 4 and 5, is evident the substantial constancy of the trend of the curves characteristic of the fan 10 according to the invention, which indicates a good operational flexibility of the blade profile 12 under different load conditions.

[0025] Represented in figures 6-11 are the diagrams regarding the comparative noise measurements between the fan 10 according to the invention and the reference axial fan. The abovementioned experimental surveys were obtained by positioning the fans into the underhood of a purposely equipped farm tractor. In particular, subject of the survey, in a temporary and direct manner, was the rotation speed of the engine and of the fan alongside the dimensions characterising the behaviour of the system according to the acoustic point of view. The determination of the aforedescribed dimensions was carried out by means of various phonometers with the respective microphones arranged at different positions. In particular, detailed measurements were performed at the front of the machine and in proximity to the point conventionally defined as the "guide ear". The subsequent step for processing and reducing of the data allowed summarising, in figures 6-11, the acoustic behaviour of the machines subject of the comparison.

[0026] Figure 6 shows, referring to the noise detected in proximity to the front zone, substantial differences regarding the noise intensity especially within the interval of the rotation speeds of most interest (torque and maximum power: 2400÷3100 revolutions per minute). Within the considered interval, the difference of the quantity in question between the rotor of the new generation and the conventional one is greater than 5 dB. Furthermore, the two subsequent figures 7 and 8 allow observing, at a rotation speed of 2000 and 3100 revolutions per minute, the spectra (in one-third octaves) regarding the analysis in frequency. As clearly observed, especially at a higher rotation speed, the rotor of the new generation (continuous section line) shows maximum noise values decidedly lower at 1kHz and 500 Hz frequencies.

[0027] Figure 9, regarding the noise in proximity to the guide ear, allows showing significant difference entities (averagely greater than 5 dB) in the interval of the engine-rotation speeds of most interest (identical to the ones of the previous case). Furthermore, in particular, it is observed that, at such position, the noise spectra (usually in one-third octaves) defined at 2000 and 3100 revolutions per minute (figures 10 and 11) are more "constant" and of lower intensity within the interval of frequencies considered.

[0028] It has thus been observed that the axial fan for cooling the underhood of a farm machine according to the present invention attains the objects described previously.

[0029] In particular, providing a blading with a wing profile made up of two distinct elements allows an efficient control of the trend of the boundary layer along the profile itself, thus reducing the tendency to stall in case of high blade keying angles required for producing considerable values of the specific work, required to the fan. The correct evolution of the flow through the blading has an additional effect related to the reduction of turbulent structures source of the main fluidodynamic drop and noise phenomena.

[0030] The axial fan for cooling the underhood of a farm machine of the present invention thus conceived is susceptible to various modifications and variants, all falling within the same invention concept; furthermore, all the details can be replaced by technically equivalent elements. In practice, the materials used, alongside the shapes and dimensions, may vary depending on the technical requirements.

[0031] Therefore, the scope of protection of the invention is defined by the attached claims.

Claims

1. Axial fan for cooling the underhood of a farm machine, of the type made up of a rotor (10) provided with a plurality of blades (12) with a wing profile and a substantially radial development constrained against the central hub (14) of said rotor (10) at one of their first ends, **characterised in that** each of said blades (12) is made up of a first element or main profile (18) and by at least a second element or flap (20) arranged in proximity to the inlet edge (18’) of said

first element or main profile (18).

2. Axial fan according to claim 1, **characterised in that** said rotor (10) has an external circumferential edge (16) constrained against which is a second end of each of said blades (12).

3. Axial fan according to claims 1 or 2, **characterised in that** said main profile (18) is **characterised by** a median line (22) having a maximum relative deflection (fm) equivalent to 6% with respect to the length of the chord (C) of said blade (12), said chord (C) being defined as the linear distance between the inlet edge (20') of said flap (20) and the outlet edge (18") of said main profile (18).

4. Axial fan according to claim 3, **characterised in that** the point of maximum bending (Xfm) of said main profile (18) is arranged in proximity to the centreline point of said chord (C).

5. Axial fan according to claims 1 or 2, **characterised in that** the thickness (Sp) of said main profile (18) has a maximum value equivalent to 12% with respect to the length of the chord (C) of said blade (12), said chord (C) being defined as the linear distance between the inlet edge (20') of said flap (20) and the outlet edge (18") of said main profile (18).

6. Axial fan according to claim 5, **characterised in that** said maximum value of the thickness (Sp) of said main profile (18) is obtained in proximity to 30% of the length of said chord (C), with measurement carried out starting from said inlet edge (20') of said flap (20).

7. Axial fan according to claims 1 or 2, **characterised in that** the radius of said rotor (10) is equivalent to two times the length of the chord (C) of said blade (12), said chord (C) being defined as the linear distance between the inlet edge (20') of said flap (20) and the outlet edge (18") of said main profile (18).

8. Axial fan according to claim 7, **characterised in that** the radius of said central hub (14) is equivalent to 75% of the length of said chord (C).

Fig. 1

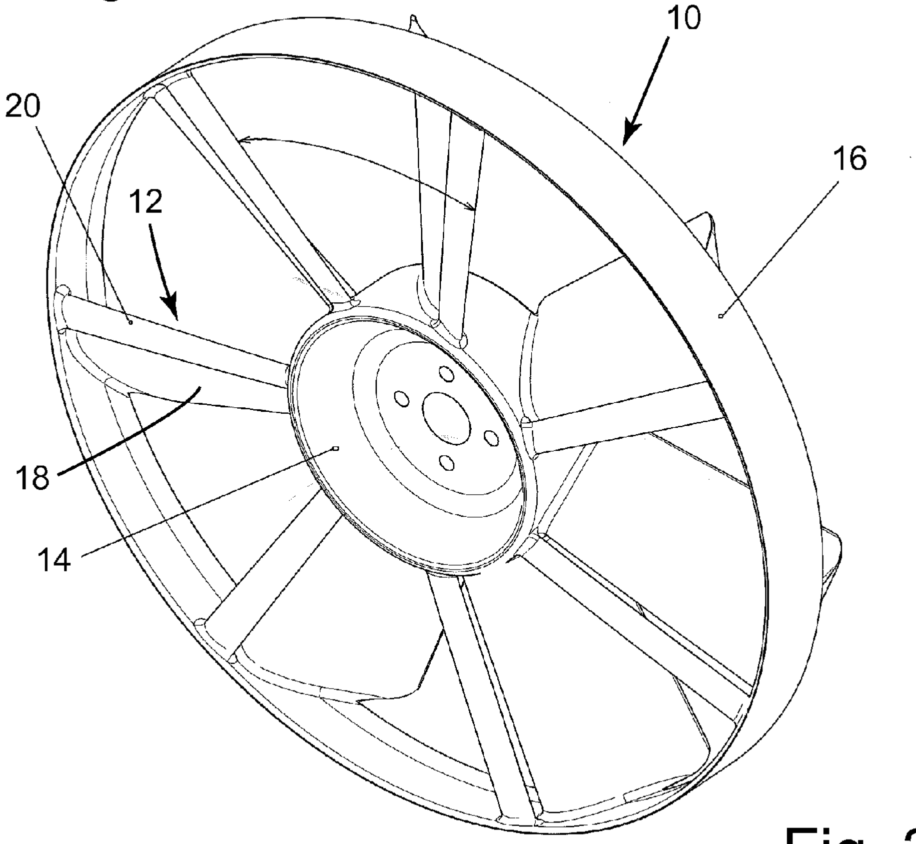


Fig. 2A

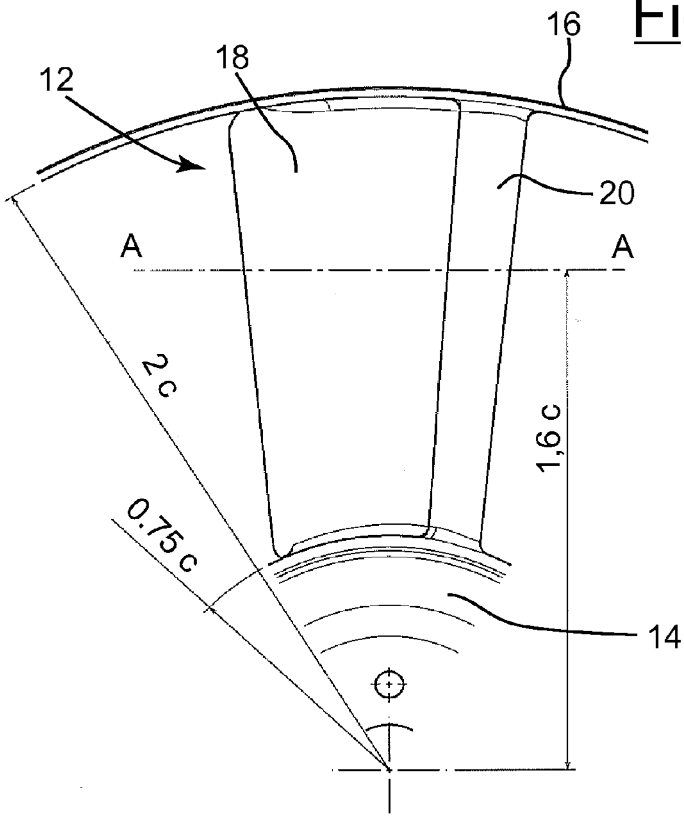


Fig. 2B

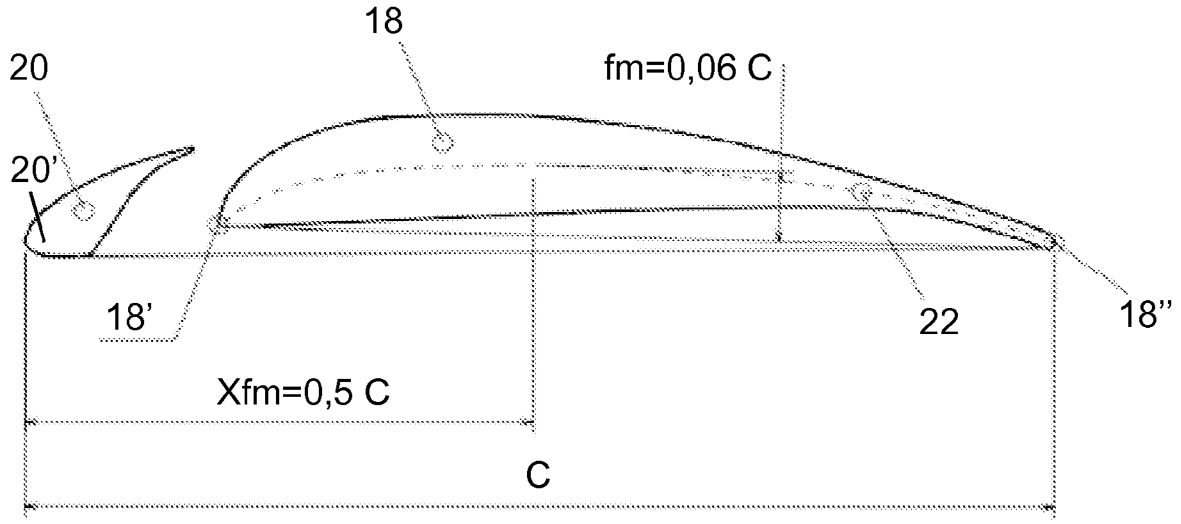
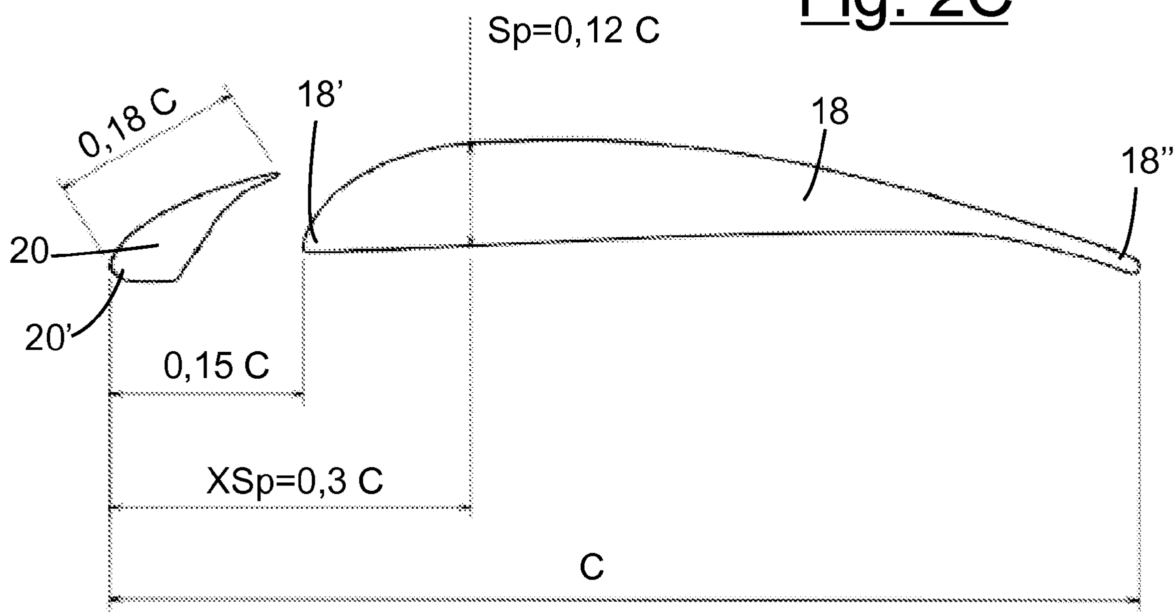
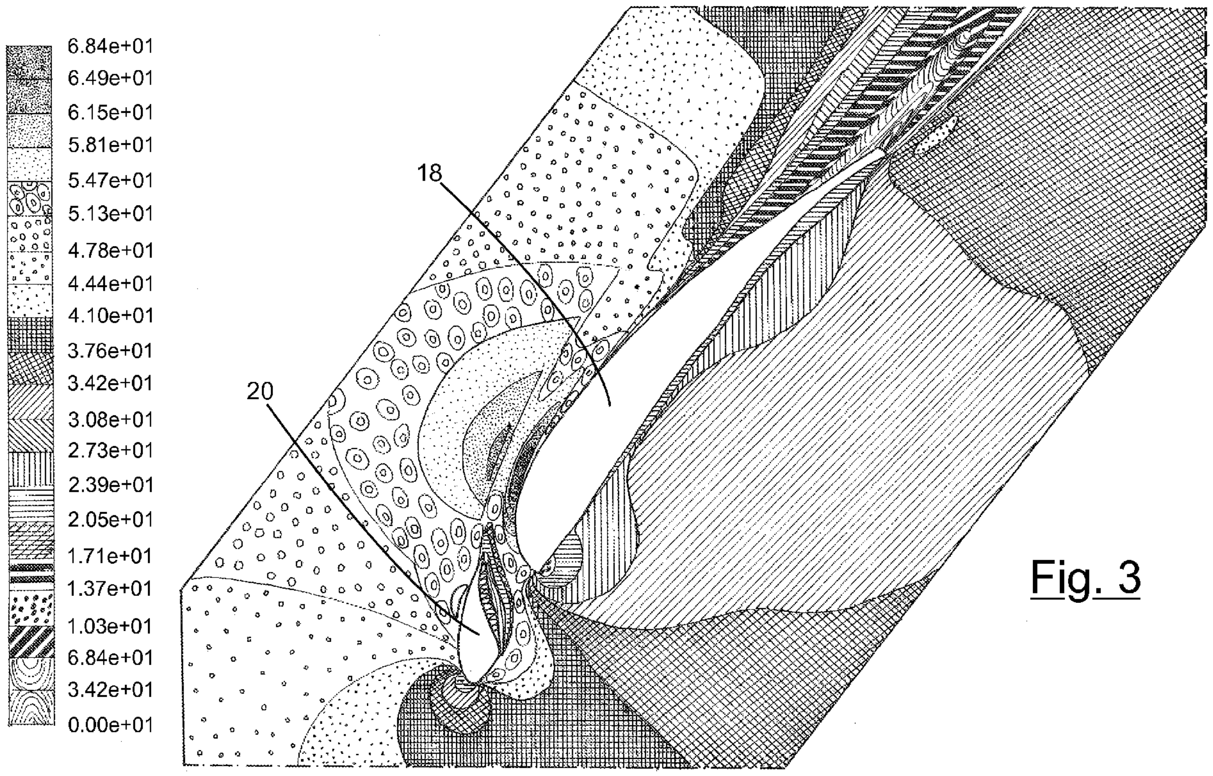

18
20
fm=0,06 C
20'
18'
22
18''
Xfm=0,5 C
C

Fig. 2C


Sp=0,12 C
0,18 C
18'
18
18''
20
20'
0,15 C
XSp=0,3 C
C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2.05e+01
1.71e+01
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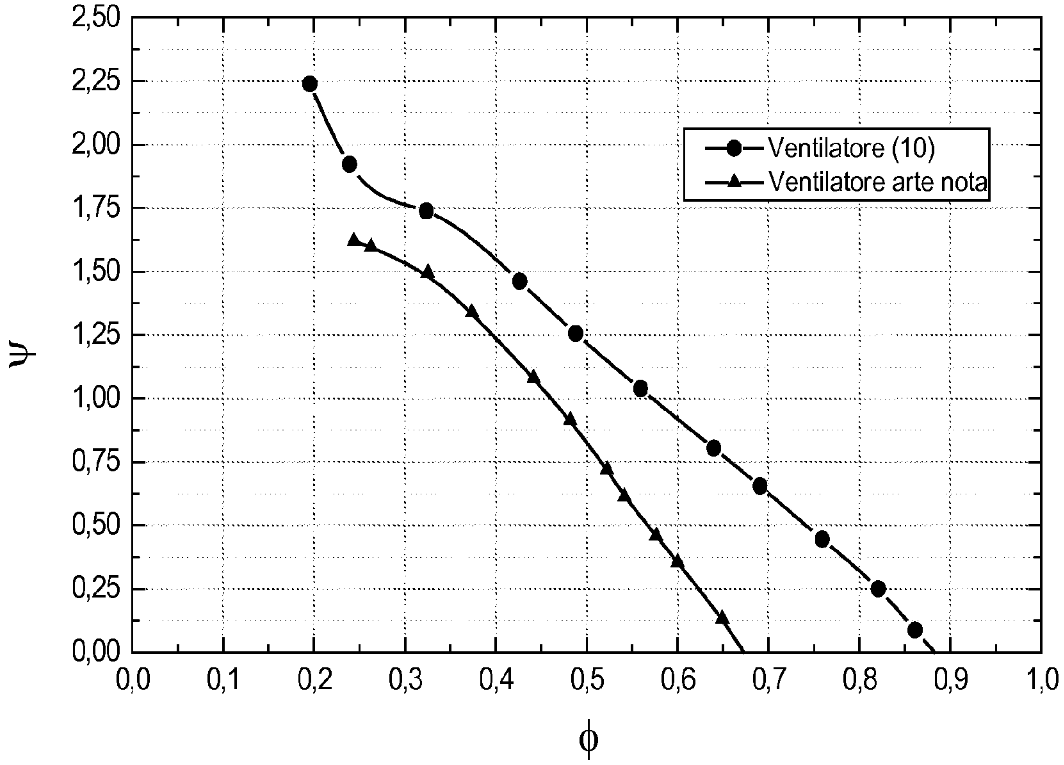
18
20

Fig. 3

Fig. 4


Ventilatore (10)
Ventilatore arte nota
ψ
φ
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2,00
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1,00
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0,50
0,25
0,00
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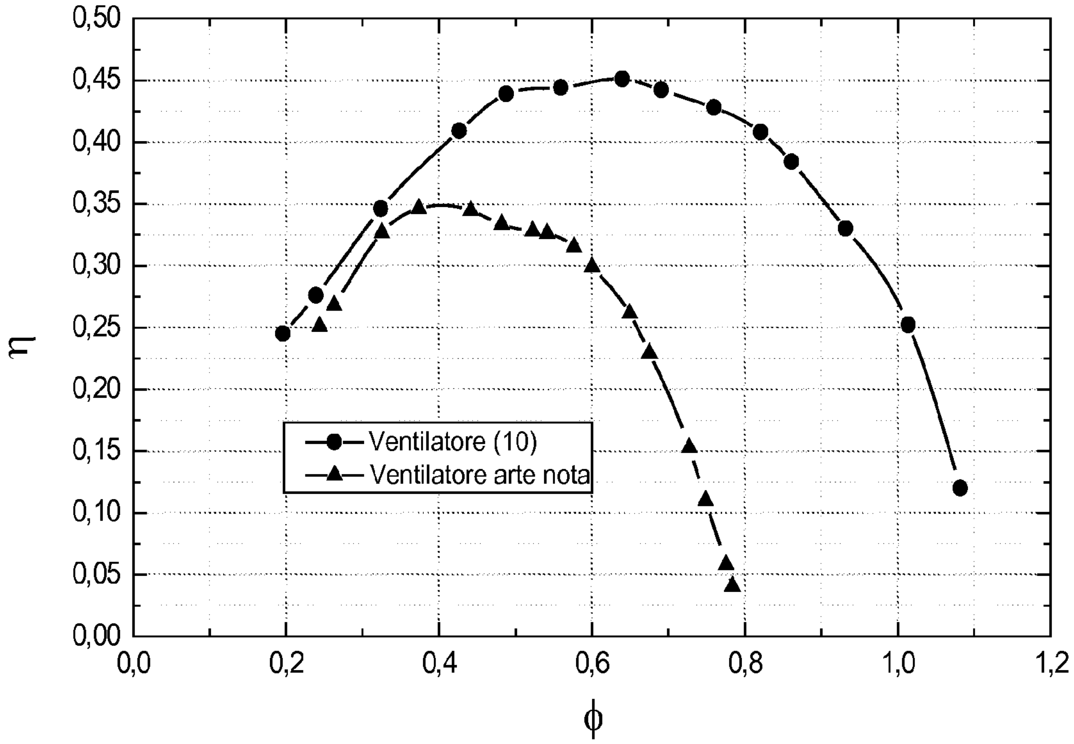
Fig. 5


Ventilatore (10)
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1,2

Fig. 6

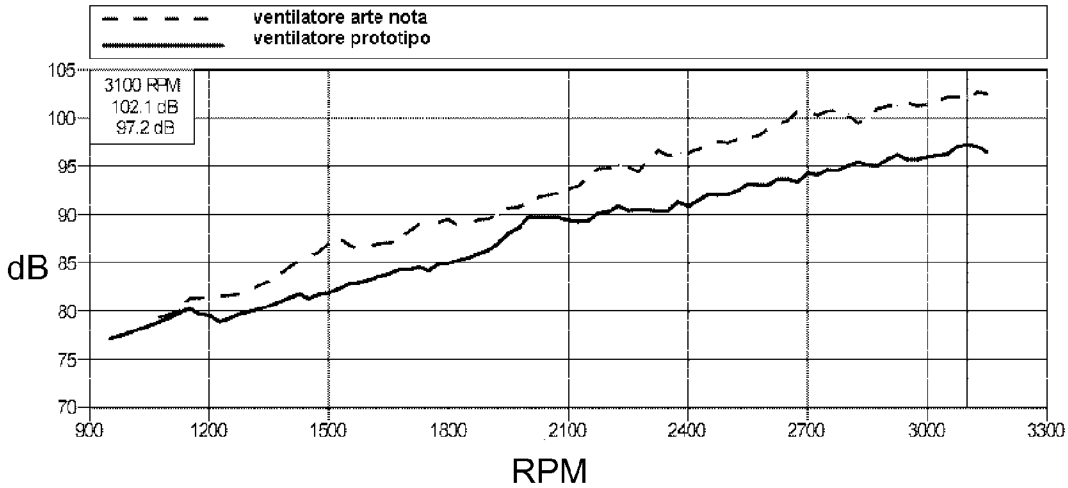

Fig. 7

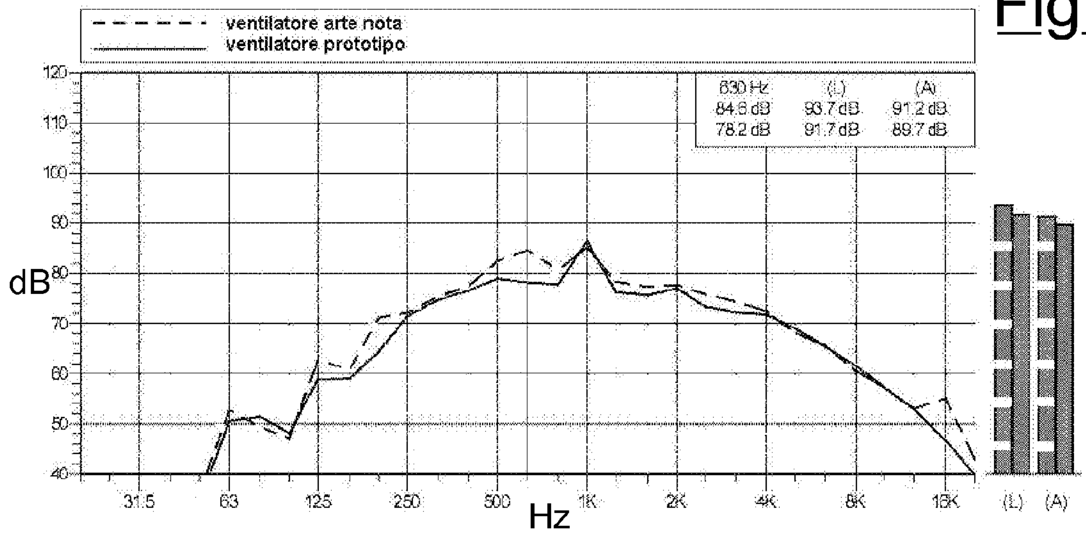

Fig. 8

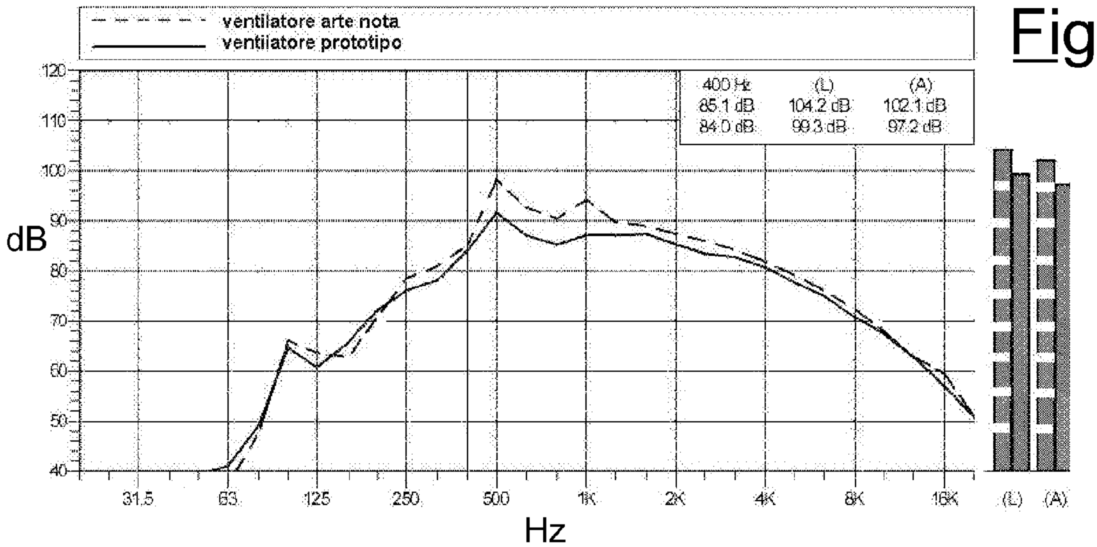

Fig. 9

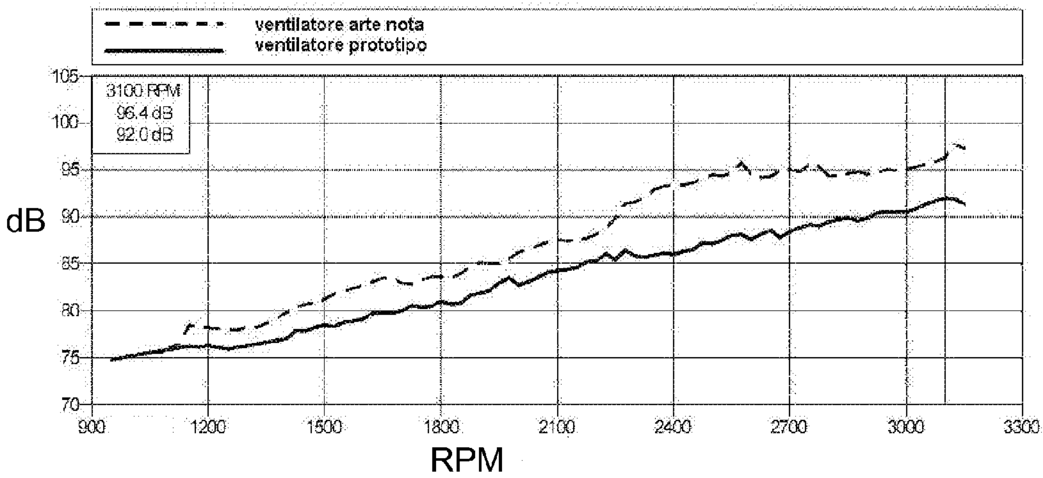

Fig. 10

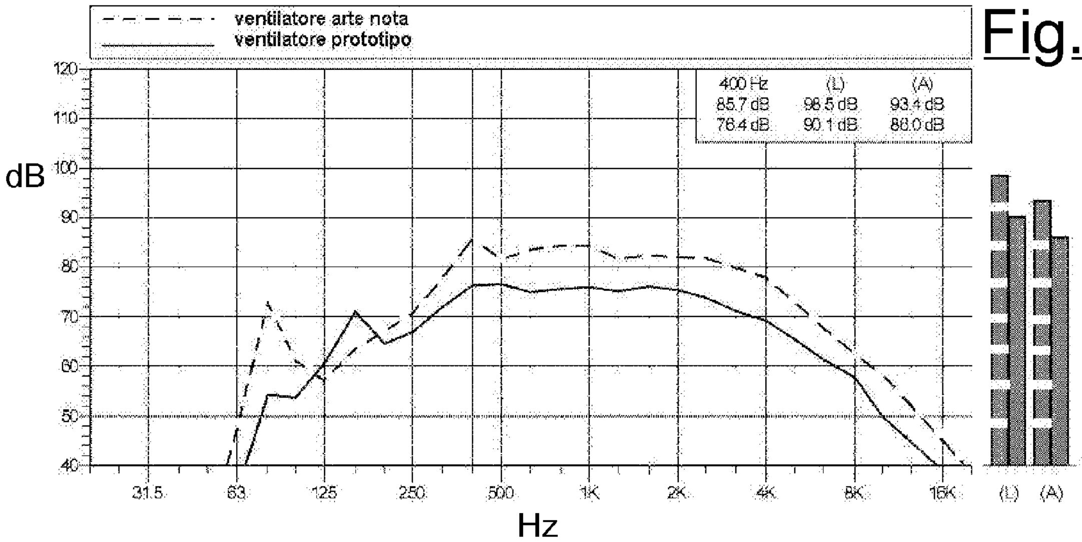

Fig. 11