May 30, 1967 N. LAING 3,322,932

PORTABLE ELECTRIC FAN HEATER

Original Filed Sept. 5, 1962 2 Sheets-Sheet 1

FIG. 1.

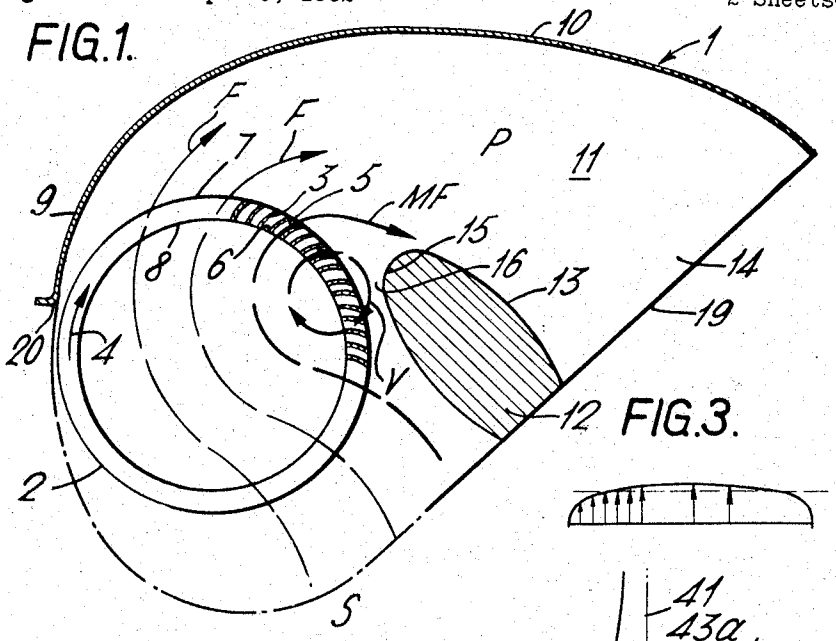


FIG. 3.

FIG. 2.

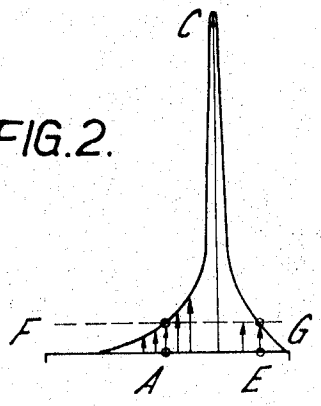


FIG. 4.

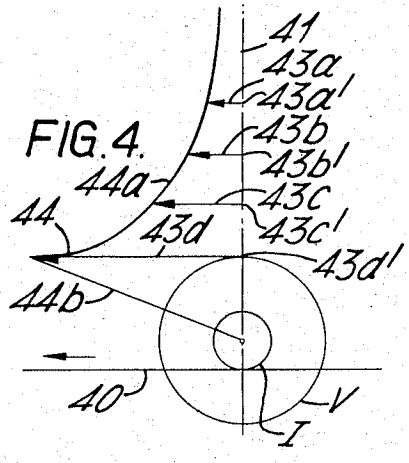


FIG. 5.

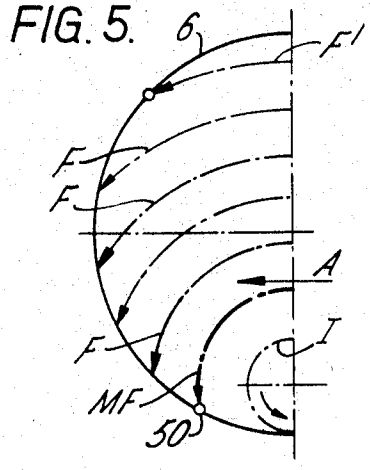
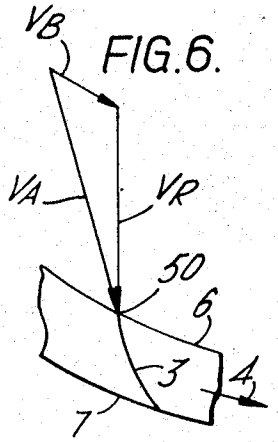


FIG. 6.



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FIG. 7.

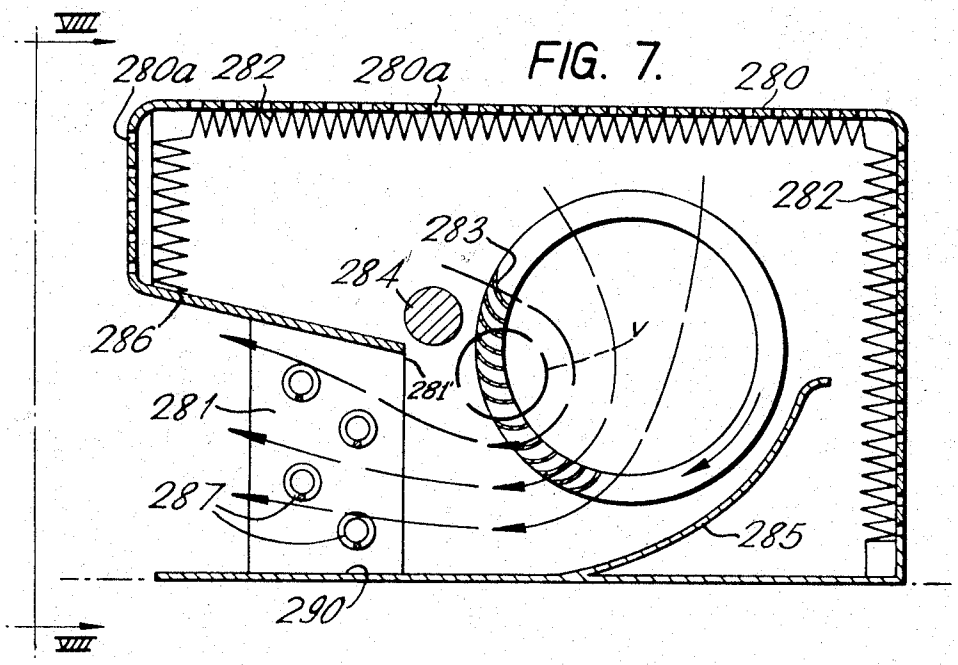
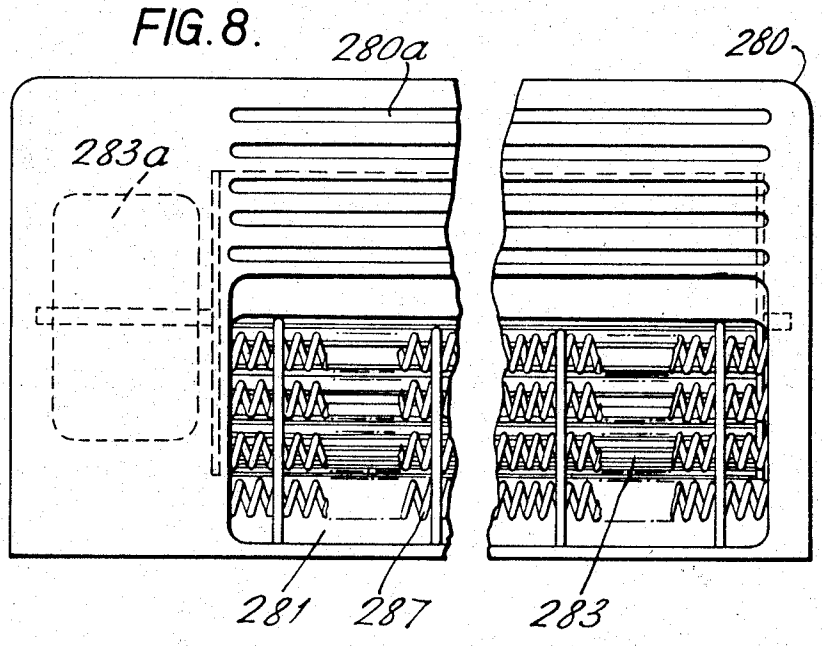


FIG. 8.



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PORTABLE ELECTRIC FAN HEATER

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Original application Sept. 5, 1962, Ser. No. 221,621. Divided and this application July 26, 1965, Ser. No. 474,874
Claims priority, application Germany, Dec. 7, 1956, L 26,388
7 Claims. (Cl. 219—370)

This invention relates to fan heaters, more especially of the electric type for domestic use, which may be portable or fixed. This application is a division of my copending application Ser. No. 221,621, filed Sept. 5, 1962, now Patent No. 3,232,522, granted Feb. 1, 1966, which is itself a continuation-in-part of application Ser. No. 671,114, filed July 5, 1957, now abandoned.

One main use of an electric fan heater is to provide additional heat in a room when the central heating is inadequate, for example in excessively cold weather. Another use is to heat a room in unseasonably cool weather in late spring or early fall when the central heating is not operating, or to provide heat in some enclosure which is not normally heated. In the usual fields of use the most important requirement for a domestic fan heater is that it should heat up an enclosure quickly. At the same time, for reasons of safety and comfort the fan heater must not generate heat in such a way that the heater itself or the air stream from it become unduly hot. In addition the fan heater must be capable of economic quantity production.

A variety of fan heaters have been proposed incorporating fans of the axial-flow type. This type of fan can be cheaply made and is not unduly bulky for the air flow it produces. However, the air flow is annular in cross-section. Though this is not objectionable for some uses, it will be seen from the following that this annular flow is a disadvantage in a fan heater. It will not normally be practicable to provide a fan heater with a centrifugal fan since such fans are inadmissably bulky for domestic use.

If the fan heater has to be portable as opposed to being built into a wall of a room or the like, then it is particularly important that it should be light, compact, steady on its supports and of pleasing exterior shape.

The main object of the invention is to provide a new and improved domestic fan heater better capable of satisfying the requirements above set forth than those previously proposed.

The invention is directed to the realization that the speed with which a room can be heated up by a fan heater depends on the direction and penetration of the warm air jet of the fan heater. An object of the invention therefore is to provide a means to direct a flat jet of warm air over the floor, the jet having as little turbulence as can be managed. With the bottom side of the jet guided by the floor its penetration is surprisingly increased as compared with a similar jet in free air. It might be thought that a jet of circular section as produced by an axial fan could be directed along the floor with the same effect as for a rectangular jet but experiments have shown that this is not so. In the first place the jet is found always to be retarded as if in free air owing precisely to its circular shape. In the second place, due mainly it is believed to the vortices left by the blade tips, an axial fan produces a jet which is inherently very turbulent and therefore readily retarded by the surrounding air. Quite apart from this, because of the turbulence of the jet it tends to stir up dust, and is thus not suitable for direction along the floor.

To produce the above-mentioned flat jet of warm air directed over the floor, the invention has recourse to the

cross-flow fan. The cross-flow fan comprises a bladed cylindrical rotor and guide means co-operating therewith on rotor rotation to induce a flow of air from an inlet region through the path of the rotating blades to the interior thereof and thence again through the path of the rotating blades to an outlet region. One form of cross-flow fan was disclosed by a Frenchman Mortier in the 1890's (see United States Patent 459,135), and a number of such fans have been proposed since. However, these proposals have mostly concerned relatively large machinery and the only commercial use of cross-flow fans, so far as is known, has been for ventilating mines in France about the year 1900: this use was abandoned after at short time. The present inventor is believed to be the first to use cross-flow fans in portable domestic equipment.

Not all forms of cross-flow fan are suitable for carrying out the invention and in contrast to many earlier proposals the invention requires the rotor to define an interior space clear of stationary guides and the guide means to comprise a pair of guide walls extending the length of the rotor and spaced exteriorly therefrom at opposite sides thereof. The invention further provides a casing for the rotor having an inlet and an outlet and heater means between the rotor and the outlet: the rotor and guide walls are positioned directly opposite the outlet and disposed for direct ejection of air from the rotor through said outlet in the form of a flat jet without substantial change of direction, and the casing is adapted to be positioned for direct ejection of the flat jet over the floor. It will commonly be preferred to have the heater means on the outlet side of the rotor.

A further important appreciation of the invention lies in making the outlet walls diverge and in arranging the heater means therein. The heating of the air gives it an increased volume. In a duct of constant cross-section the heating would cause an acceleration of the air proportional to the absolute temperature and would impose an equivalent back pressure on the fan leading to reduced throughput. By making the outlet diverge the acceleration of the air can be reduced or eliminated according to the designer's requirements, since the increasing volume of air being heated can be given an increasing space to accommodate it. The diverging outlet reduces the pressure drop across the heater means and accordingly allows for increased throughput of air. Advantageously the outlet can be so designed in relation to the fan that the accelerating effect of the heating means is nearly but not completely eliminated so that the outlet due to the presence therein of the heater means and notwithstanding its divergence, functions as a slightly converging nozzle with the effect of rendering flow more laminar as well as faster and therefore better able to penetrate as a jet into the room being heated.

In what follows a particularly efficient and easily manufactured form of cross-flow fan is described as preferred for use in the portable fan heater of the invention. This form of cross-flow fan can be made more efficient and quiet in operation, and also much less bulky than a comparable centrifugal fan: it can be made to produce a jet of remarkably little turbulence and excellent penetration properties with little tendency to pick up dust.

In the drawings:

FIGURE 1 is a cross-sectional view of a fan which with the following explanatory FIGURES 2 to 6 illustrates certain features of construction and flow pattern which are recommended in the fan heater according to the invention;

FIGURE 2 is a graph illustrating velocity of air flow at the outlet of the FIGURE 1 fan;

FIGURE 3 is a graph illustrating velocity of air flow at the outlet of a conventional fan;

FIGURE 4 is a graph illustrating air velocity within the field and core of a Rankine vortex;

FIGURE 5 illustrates, somewhat idealized, the flow occurring in one half of the cross-sectional area of the rotor shown in FIGURE 1;

FIGURE 6 is a vector diagram illustrating air flow at one blade of the FIGURE 1 rotor where the air is passing from the interior of the rotor to the outlet region;

FIGURE 7 is a somewhat diagrammatic view in section transverse to the rotor axis of one form of fan heater according to the invention, and

FIGURE 8 is a diagrammatic side elevation of the fan heater of FIGURE 7, seen in the direction of the arrow VIII in that figure.

Referring to the drawings, the fan illustrated in FIGURE 1 comprises a cylindrically bladed rotor **2** having thereon a plurality of blades **3** concavely curved in the direction of rotation of the rotor indicated by the arrow **4** wherein the blades **3** have their outer edges **5** leading their inner edges **6**. The outer edges define an outer envelope **7** while the inner edges define an inner envelope **8** when the rotor is rotated. The rotor is mounted, by means not shown, whereby it will rotate about its axis. A guide wall **9** extends the length of the rotor and merges with a wall **10** to form one side of an exit duct **11** of the machine. A vortex-forming and stabilizing means **12** also extends the length of the rotor and is positioned exteriorly thereof and has thereon a wall **13** which forms part of the exit duct and which more particularly forms part of a diffuser section **14** as is more fully explained hereafter.

The vortex-forming and stabilizing means **12** has a rounded end **15** which has a portion extending towards the rotor in the direction of rotation to form a converging gap **16** which, as more fully explained hereafter, serves to form and stabilize a fluid vortex when the rotor is rotated. The means **12** also serves to separate the suction side S from the pressure side P of the machine and defines with the wall **9** an entry and an exit region to the rotor. End walls **19**, only one of which is shown, substantially cover the ends of the machine.

The wall **9** terminates a point **20** which is spaced from the rotor a minimum of one-third the blade depth and not more than three times the blade depth of the blades **3** in order to minimize interference which causes an undesirable noise when the machine is operated while at the same time the wall provides a means to guide the flow leaving the machine. Wall **15** of the vortex-forming and stabilizing means **12** likewise is spaced a substantial distance from the rotor, in this instance a distance equal to a minimum distance of at least one-third the blade depth of the blades of the rotor. Because both the wall **9** and the vortex-forming and stabilizing means **12** are spaced from the latter a substantial distance, close manufacturing tolerances do not have to be observed when assembling the machine and, as such, the machine lends itself to economical construction such as is achieved when sheet metal stampings are utilized.

In operation of the fan illustrated in FIGURE 1, a fluid vortex having a core designated by the line V approximating a Rankine type vortex is formed wherein the core is positioned eccentrically with respect to the rotor axis and wherein the core will interpenetrate the path of the rotating blades of the rotor. The whole throughput of the machine will then flow twice through the blade envelope in a direction perpendicular to the rotor axis indicated by the flow lines F, MF.

FIGURE 4 illustrates an ideal relation of the vortex to the rotor **2** and the distribution of flow velocity in the vortex and in the field of the vortex. The line **40** represents a part of the inner envelope **6** of the rotor blades **3** projected onto a straight line while the line **41** represents a radius of the rotor taken through the axis of the vortex

core V. Velocity of fluid at points on the line **41** by reason of the vortex is indicated by the horizontal lines **43***a*, **43***b*, **43***c*, and **43***d*, the length of these lines being the measure of the velocity at the points **43***a*^{1}, **43***b*^{1}, **43***c*^{1}, and **43***d*^{1}. The envelope of these lines is shown by the curve **44** which has two portions, portion **44***a* being approximately a rectangular hyperbola and the other portion, **44***b*, being a straight line. Line **44***a* relates to the field region of the vortex and the curve **44***b* to the core. It will be understood that the curve shown in FIGURE 4 represents the velocity of fluid where an ideal or "mathematical" vortex is formed, and that in actual practice, flow conditions will only approximate these curves.

The core of the vortex is a whirling mass of air with no translational movement as a whole and the velocity diminishes from the periphery of the core to the axis **42**. The core of the vortex intersects the blade envelope as indicated at **40** and an isotach I within the vortex having the same velocity as the inner envelope contacts the envelope. The vortex core V is a region of low pressure and the location of the core in a machine constructed according to the invention can be determined by measurement of the pressure distribution within the rotor.

The velocity profile of the air where it leaves the rotor interior and passes through the path of the rotating blades will be that of the vortex. In the ideal case of FIGURE 4, this profile will be that of the Rankine vortex there shown by curves **43***a* and **43***b*, and in actual practice, the profile will still be substantially that shown in FIGURE 4 so that there will be in the region of the periphery of the core V shown in FIGURE 1 a flow tube MF of high velocity and the velocity profile taken at the exit of the rotor will be similar to that shown in FIGURE 2 where the line FG represents the exit of the rotor and the ordinates represent velocity. The curve shown exhibits a pronounced maximum point C which is much higher than the average velocity represented by the dotted line.

It will be appreciated that much the greater amount of fluid flows in the flow tubes in the region of maximum velocity. It has been found that approximately 80% of the flow is concentrated in the portion of the output represented by the line AE which is less than 30% of the total exit of the rotor. A conventional velocity profile for fluid flow in a defined passage is illustrated by way of contrast in FIGURE 3 where the average velocity of flow is represented by the dotted line. Those skilled in the art regard this profile as being aproximately a rectangular profile which following the principle generally adhered to is the sort of profile heretofore sought in the outlet of a fan.

The maximum velocity C shown in FIGURE 2 appertains to the maximum velocity flow tube indicated as MF in FIGURE 1. With a given construction the physical location of the flow tube MF may be closely defined. The relative velocity between the blades and fluid in the restricted zone of the rotor blades **3** through which the flow tube MF passes is much higher than it would be if a flow machine were designed following the conditions adhered to heretofore in the art respecting the desirability of a rectangular velocity profile at the exit arc and even loading of the blades.

Under low Reynolds number conditions, which are the conditions obtaining in a domestic fan heater, this unevenness of the velocity profile leads to beneficial results in that there will be less separation and energy loss in the restricted zone through which the flow tube MF passes than if that flow tube had the average velocity of throughput taken over the whole exit of the rotor. There is a more efficient transfer of momentum to the air by the blades in this restricted zone and while the transfer of momentum in the flow tubes travelling below the average velocity will be less efficient, nevertheless when all of the flow tubes are considered, there is a substantial gain in efficiency.

The Reynolds number in a given fan is a dimensionless number indicative of the flow conditions therein, and is herein defined as

$$Re=\frac{d.c}{\gamma}$$

where d is the blade depth measured radially of the rotor, c is the peripheral speed of the rotor, and γ the kinematic viscosity of the air, equal to the quotient of dynamic viscosity and density. A Reynolds number is considered herein to be low if, as above defined, it is less than 5×10^4. By way of example, if the fan of FIGURES 7 and 8 has a rotor of 60 mm. diameter and is driven to rotate at 1500 r.p.m., which with a rotor length of 180 mm. would be satisfactory for a 2 kw. heating element, the Reynolds number as above defined will be about 1.2×10^4.

FIGURE 5 illustrates the ideal distribution of flow tubes F occurring within one half the rotor area defined by the inner envelope **6**, it being understood that the flow tubes in the other half of the rotor are similar. The maximum velocity flow tube MF is shown intersecting the envelope **6** at point **50** and the isotach I as being circular when the whole rotor is considered. It is seen that ideally the maximum velocity flow tube MF undergoes a change of direction of substantially 180° from the suction to the pressure sides when the flow in the whole rotor is considered. It is also to be noted that the major part of throughput, represented by the flow tube MF, passes through the rotor blades where they have a component of velocity in a direction opposite to the main direction of flow within the rotor indicated by the arrow A.

FIGURE 6 is a diagram showing the relative velocities of flow with respect to a blade at the point **50** referred to in FIGURE 5. In this figure V_B represents the velocity of the inner edge of the blade **3** at the point **50**, V_A the absolute velocity of the air in the flow tube MF at the point **50**, and V_R the velocity of that air relative to the blade as determined by completing the triangle. The direction of the vector V_R coincides with that of the blade at its inner edge so that fluid flows by the blade substantially without shock.

The character of a vortex is considered as being determined largely by the blade angles and curvatures. The position of the vortex, on the other hand, is considered as being largely determined by the configuration of the vortex forming means which forms and stabilizes a vortex in co-operation with the bladed rotor. The particular angles and curvatures in any given case depend upon the following parameters: the diameter of the rotor, the depth of a blade in a radial direction, the density and viscosity of the fluid, the disposition of the vortex forming means and the rotational speed of the rotor, as well as the ratio between over-all pressure and back pressure. These parameters must be adapted to correspond to the operating conditions in a given situation. Whether or not the angle and shape of the blades have been fixed at optimum values is to be judged by the criterion that the flow tubes close to the vortex core are to be deflected substantially greater than 90°.

It is to be appreciated that the flow lines of FIGURE 1 do not correspond exactly to the position of the vortex core V as illustrated in FIGURES 4 and 5 which represent the theoretical or mathematical flow. These latter figures show that it is desirable to have the axis of the core of the vortex within the inner blade envelope **6** so that the isotach within the core osculates that envelope. Although this position is desirable, it is not essential, and in fact, is not achieved in the structure shown in FIGURE 1.

It is to be further appreciated that despite the divergence of the flow in FIGURE 1 from the ideal, the maximum velocity flow tube MF with which is associated the major part of the throughput is nevertheless turned through an angle of substantially 180° in passing from the suction

to the pressure side of the rotor and that this maximum flow tube passes through the rotor blades where the blades have a velocity with a component opposite to the main direction of flow through the rotor as indicated by the arrow A.

The fan heater according to the invention illustrated in FIGURES 7 and 8 comprises a generally box-shaped casing **280** which has one side cut away to form an outlet duct **281** having an entry portion **281'**. The remainder of the side of the casing on which the outlet duct **281** is located, the top wall of the casing and the opposite or rear wall of the casing from the outlet, are perforated as shown at **280***a* to form an inlet and support a filter **282** on their inside areas. A bladed rotor **283** is mounted within the casing for rotation about a horizontal axis and driven by a motor **283***a*. A small cylinder **284** is mounted parallel and adjacent to the rotor **283** and serves as a vortex forming and stabilizing means. A guide wall **285** is positioned radially opposite the rotor from the cylinder **284**. In operation of the device, a vortex having a core V is formed and air flows through the rotor **283** along the flow lines indicated. A wall **286** extends from the casing **280** adjacent the outlet **281** to a point close to the cylinder **284** to form an upper portion of the outlet duct and heating coils **287** are located within the outlet duct. In operation, air passes through the complete area of the filter **282** at a low velocity, through the rotor, and is then discharged outward through the outlet **281** past the heating coils **287** whereby the air may be heated.

It is to be understood that rotor **283**, vortex forming and stabilizing means **284** and guide wall **285** are designed on the same principles and function in the same way as the corresponding parts **2**, **12** and **10** of FIGURE 1. Thus, for a further understanding of the fan heater of FIGURES 7 and 8, FIGURES 1 to 6 should be referred to.

It will be seen from FIGURE 7 that the guide wall **285** presents to the air flow a surface which merges with that of the base **290** of the casing **280**, which forms the lower boundary of the outlet duct **281**. When the fan heater rests on the floor as it is adapted to do the air leaves the outlet duct **281** in a flat jet directed over the floor. It will be seen that the rotor **283** and guide means **284**, **285** are positioned directly opposite the outlet and positioned for direct ejection of air from the rotor **283** past the heating coils **287** and through the outlet duct parallel to the floor. By this means there is provided a jet of good penetration properties; the low turbulence of the jet enhances its penetrative character and prevents excessive stirring up of dust.

It will be noticed that the casing wall **286** and base **290** which define the outlet duct **281** diverge in the direction of air flow through the outlet duct, which accordingly forms a diffuser, and accommodates at least in some measure the increasing volume of air as it becomes heated in flowing past the heating coils **287**, thereby reducing back pressure on the fan, as explained in the foregoing.

I claim:

1. A fan heater comprising a substantially rectangular casing having a top wall, a flat bottom wall, a rear wall and a front wall, an air inlet in said front, top and rear walls, a rectangular outlet in said front wall, an electric motor driven cylindrical bladed rotor of the cross-flow type extending the length of and disposed in alignment with and parallel to said outlet, said rotor defining an interior space clear of stationery guides, a pair of guide walls extending the length of the rotor and spaced exteriorly thereof and at opposite sides thereof with one of said guide walls extending upwardly from said bottom wall, heater means positioned between the rotor and the outlet and extending across the width of said outlet with the rotor and guide walls being positioned directly opposite the outlet and disposed for direct ejection of air from the rotor past said heater means and through said outlet in the form of a flat jet without any substantial change of direction.

2. A fan heater as claimed in claim 1, wherein a vertically diverging outlet duct extends between the outlet and the rotor diverging vertically in a direction away from the rotor and wherein the heater means are located in said diverging duct.

3. A fan heater as claimed in claim 1, wherein an upper surface of the bottom wall forms the lower boundary of said outlet.

4. A fan heater as claimed in claim 1, wherein the top and rear walls are apertured over the greater part of their total area to provide said inlet.

5. A fan heater as claimed in claim 4, wherein the casing carries a filter on the inside of said apertured area.

6. A fan heater as claimed in claim 1, wherein the outlet is formed in the lower part of the front wall.

7. A fan heater comprising a rectangular casing having a top wall, a bottom wall, a rear wall and a front wall; an outlet in said front wall; an inlet in at least one of said rear, top and front walls; an electric motor driven bladed cylindrical rotor of the cross flow type rotatably mounted within the casing and defining an interior space; an outlet duct, having a top and bottom wall, extending the length of the rotor and having an entry portion adjacent the rotor with said duct extending to the outlet, the top wall of said duct being more upwardly inclined than the bottom wall of said duct so that the diverging upward top duct wall produces a substantially uniform velocity of air passing through the duct; vortex forming and stabilizing means adjacent the top of the entry portion of the outlet duct extending the length of the rotor and exterior

thereof and cooperating therewith on rotor rotation to form and stabilize a fluid vortex having a core which interpenetrates the path of the rotating blades to induce a flow of air through the inlet, through the path of the rotating blades of the rotor to the interior thereof, through the path of the rotating blades to the exterior thereof and thence through the diverging outlet duct to the outlet; and heating means in and extending substantially completely across the width of the vertically diverging outlet duct; said vertically diverging outlet duct serving to reduce back pressure of air in said duct caused by increase of the temperature of air passing over the heating means.

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