July 23, 1940. F. L. WATTENDORF 2,208,615

AXIAL FLOW TYPE FAN OR PUMP

Filed July 27, 1939 2 Sheets-Sheet 1

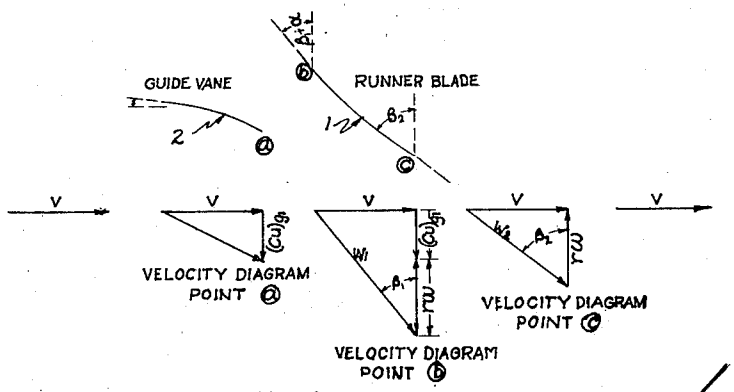


FIG. 1

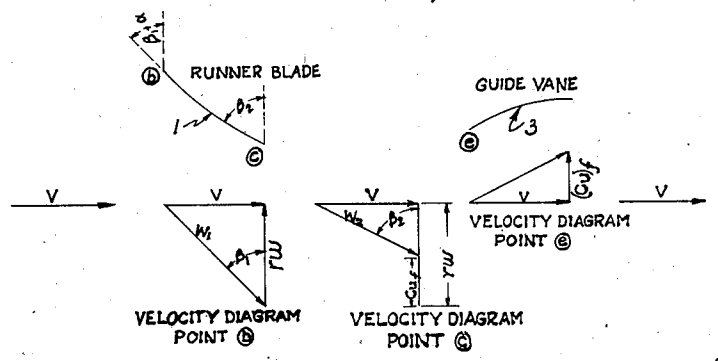


FIG. 2

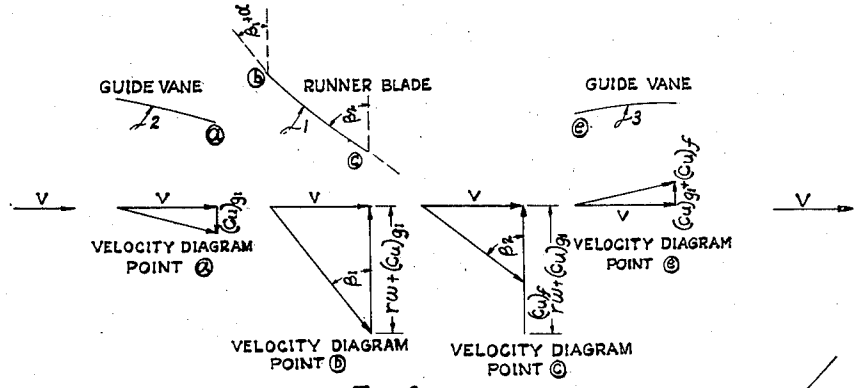


FIG. 3

INVENTOR
Frank L. Wattendorf
BY W. Glenn Jones
ATTORNEY

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AXIAL FLOW TYPE FAN OR PUMP

Filed July 27, 1939 2 Sheets-Sheet 2

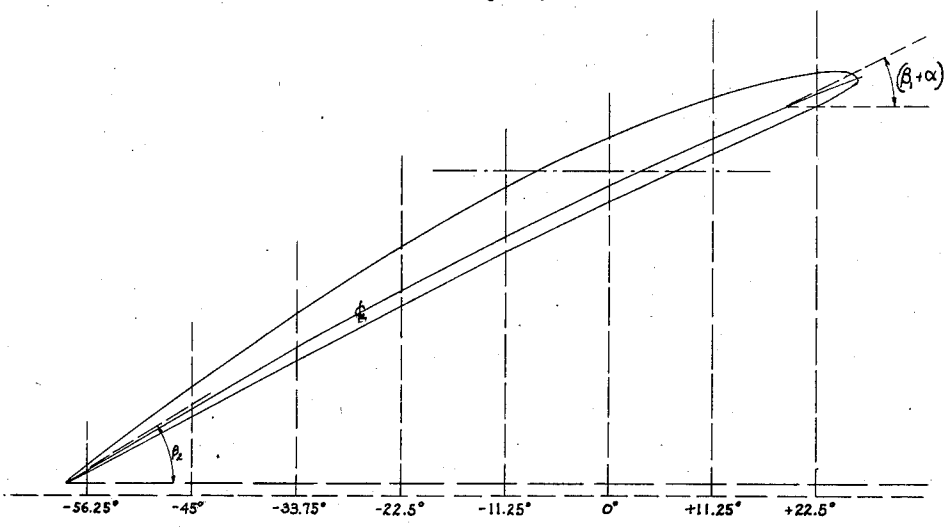


Fig. 7

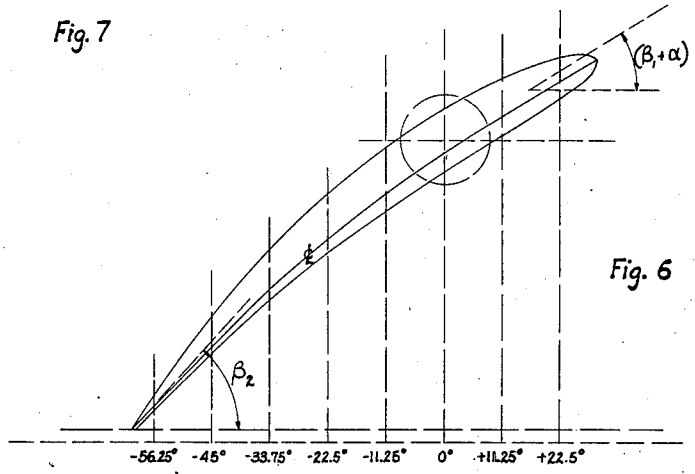


Fig. 6

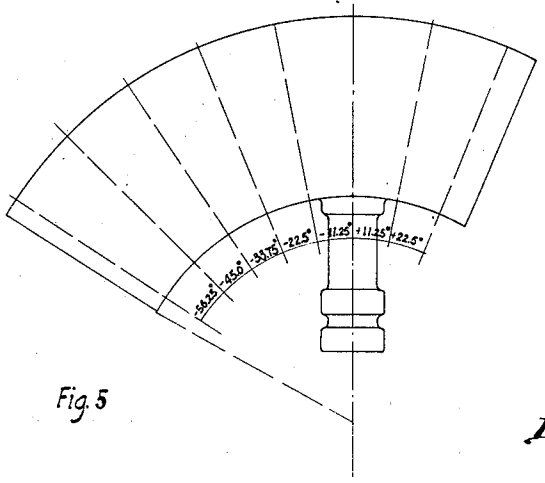


Fig. 5

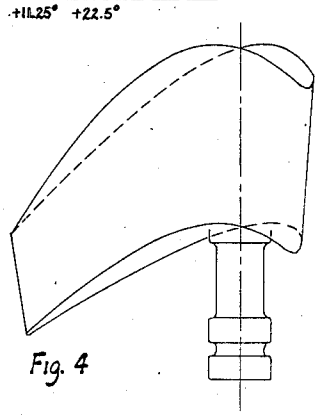


Fig. 4

INVENTOR

Frank L. Wattendorf

BY W. Glenn Jones

ATTORNEY

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UNITED STATES PATENT OFFICE

2,208,615

AXIAL FLOW TYPE FAN OR PUMP

Frank L. Wattendorf, Hull, Mass.

Application July 27, 1939, Serial No. 286,877

3 Claims. (Cl. 230—120)

(Granted under the act of March 3, 1883, as amended April 30, 1928; 370 O. G. 757)

This invention relates to a fan or pump of the axial flow type capable of developing high pressures while maintaining low rotating tip speeds with resulting low noise development in the case of fans and minimum danger of cavitation in the case of pumps.

It is an object of this invention to provide an axial flow type fan or pump having a pressure coefficient larger than 0.5 with blade spacing ratios larger than 1.

Fan characteristics are preferably expressed in the form of non-dimensional coefficients for the pressure and capacity. The following notation will be used:

p=total head developed by the fan or pump
R=radius of the blade tips
ω=angular velocity
ρ=mass density of the fluid (air, water, oil, etc.) in which the fan is operating
A=net area of the fluid passage through the fan proper
Q=quantity of fluid delivered per unit time
η=total efficiency of the unit for each radius
ψ=the pressure coefficient
λ=the volume coefficient
C_L=the lift coefficient
r=the radial distance to any blade element
β=the mean angle of pitch
z=the number of blades
b=the blade chord
α=the angle of attack
$S=bz/2\pi r$=the spacing ratio, i. e., the ratio of the blade chord b to the spacing between two blades
Γ=the circulation
w=the average velocity of the fluid relative to each blade element

The pressure coefficient ψ may be expressed by the following formula:

$$\psi=\frac{p}{\frac{1}{2}\rho(\omega R)^2}$$

and the volume coefficient λ by:

$$\lambda=\frac{Q/A}{\omega R}$$

Axial flow fans described in technical literature are seldom found to exceed a pressure coefficient of 0.5 for a single stage. The reason for this limitation can be found in the methods used for the design calculations. For the usual design the theory of a single airfoil is used as a basis, whereby each blade is assumed to produce the same force as an equivalent single airfoil in the same relative flow. The axial component of the force acting on the blades is responsible for the pressure increase of the fans. Application of airfoil theory gives the following approximate relationship between the pressure coefficient and the blade characteristics:

$$\psi=C_L S(r/R)\lambda\left(\frac{1}{\sin\beta}\right)\eta$$

For optimum efficiency of a blade element the best mean angle of pitch β is 45°, while the flow coefficient λ, for a given quantity Q, is limited for reasons of efficiency, noises or cavitation. The lift coefficient is limited by the flow separation (stall) phenomenon, and often has to be kept very small for reduction of noise in the case of fans, and for minimizing the danger of cavitation in the case of pumps.

The only means, therefore, to increase the pressure coefficient ψ is by increasing the spacing ratio S. However, when S approaches and exceeds unity the behavior of the fan blades deviates essentially from that of a single airfoil, so that the usual methods of design do not apply for values of S greater than one.

The present invention provides means for utilizing a spacing ratio greater than 1, and obtaining at the same time a pressure coefficient ψ greater than 0.5, with the accompanying advantages of a relatively small fan diameter and low rotating tip speed.

This is accomplished by proper bending of the leading and trailing edges of the blades in the following manner:

The trailing edge of any blade element is bent at each radius in such a way that the zero lift line of the corresponding element forms with the plane of rotation an angle β_2 which is calculated as follows:

Case I—In which guide vanes are mounted in front, i. e., upstream, of the runner:

$$\tan\beta_2=\frac{Q/A}{\omega r}=\lambda\frac{R}{r}$$

Case II—in which guide vanes are mounted behind, i. e., downstream, of the runner:

$$\tan \beta_2 = \lambda \frac{1}{\left(\frac{r}{R} - \frac{\psi}{\eta}\frac{R}{r}\frac{1}{2}\right)}$$

Case III—In which guide vanes are located both upstream and downstream of the runner. In this case the trailing edge of the blade element at each radius is bent in such a direction that the zero lift line of the corresponding element forms with the plane of rotation an angle β_2 such that:

$$\tan \beta_2 = \lambda \frac{1}{\left(\frac{r}{R} + (k-1)\frac{\psi}{2\eta}\frac{R}{r}\right)}$$

where k is a constant depending on the distribution of the rotation added to the fluid by the guide vanes in front and substracted by the guide vanes behind.

In all three cases the leading edge of any blade element has to be bent in such a way that the flow enters the runner sufficiently without shock that flow separation is avoided over the entire blade, and particularly at the leading edge.

The direction of flow relative to the blade element at the leading edge forms with the plane of rotation an angle β_1 of the following magnitudes:

Case I;

$$\tan \beta_1 = \lambda \frac{1}{\left(\frac{r}{R} + \frac{\psi}{\eta}\frac{R}{r}\frac{1}{2}\right)}$$

Case II;

$$\tan \beta_1 = \lambda \frac{R}{r}$$

and

Case III;

$$\tan \beta_1 = \lambda \frac{1}{\left(\frac{r}{R} + k\frac{\psi}{\eta}\frac{R}{r}\frac{1}{2}\right)}$$

In all three cases the tangent to the zero lift line of each element forms an angle $(\beta_1 + \alpha)$ with the plane of rotation.

α denotes the angle of attack of the selected blade profile of the runner, corresponding to the desired C_L for that profile. The profile may be selected from a large number of tested profiles well known in the art. α is defined by the expression

$$C_L = \frac{L}{\text{Area}\left(\frac{\rho}{2}\right)w^2} = B2\pi \sin \alpha$$

(within limits), where B varies with the shape of the profile selected from tested profiles, the term "area" refers to the area of a blade element or b times unit length in the radial direction, and L refers to the total anodynamic lifting force on the same element, said force acting in a direction perpendicular to the mean flow.

The chord length b and the lift coefficient C_L of each element are selected in such a way that the circulation, $\Gamma = C_L b w/2$, is approximately independent of the radius r.

For the three cases considered, the guide vanes must impart to the fluid a rotational component C_u of the velocity, determined as follows:

Case I

$$C_u = \frac{\psi}{2\eta}\frac{R}{r}\omega R$$

Case II—Since the guide vanes are downstream, they must remove from the fluid a rotational component of the same magnitude.

Case III—The upstream guide vanes may produce a rotational component equal to an arbitrary factor times the expression for Case I. The downstream guide vanes then will have to remove the balance of rotational component remaining in the flow leaving the fan.

For all three cases the guide vanes may be located to suit the individual design. They may be used to regulate the flow at operating conditions removed from the design point. The runner blades may be made controllable to obtain a high efficiency at operating conditions removed from the design point.

In the drawings:

Fig. 1 is a diagrammatic sectional view taken through a runner blade and one of the guide vanes associated therewith in the arrangement constituting Case I, and showing velocity diagrams for this arrangement.

Fig. 2 is a similar view of the arrangement constituting Case II.

Fig. 3 is a similar view of the arrangement constituting Case III.

Figs. 4 and 5 are side and rear views respectively, of a typical runner blade arranged according to the invention.

Figs. 6 and 7 are sectional views taken along the shortest and longest radii respectively of the runner blade of Figs. 4 and 5.

In each of Figs. 1, 2 and 3 the fluid acted upon enters the fan or pump from the left and the direction of travel of the runner blade is toward the top of the drawing. The velocity diagrams in each figure show the composition of the velocities imparted to the fluid at various points of its travel through the fan. In these figures the runner blade is indicated by reference numeral 1, the upstream guide vane by 2 and the downstream guide vane by 3. The relations expressed by the velocity diagrams are as described in the following explanation. When an arrow appears above the plus or minus sign it indicates vectorial addition or subtraction.

At the entrance to the front guide vanes:

V=axial velocity

At the exit from the front guide vanes:

$\vec{V} + (C_u)_{g1}$=absolute velocity

C_u =the rotational velocity component of the absolute velocity, and is counted positive when in the direction of the runner rotation.

$(C_u)_{g1}$=rotational velocity component imparted by the front guide vanes.

$(C_u)_f$ =rotational velocity component imparted by the runner.

$(C_u)_{g2}$=rotational velocity component imparted by the rear guide vanes.

At the entrance to the runner:

$$\vec{V} + (C_u)_{g1} = \vec{W}_1 + r\omega$$

where W_1 is the relative velocity at the entrance of the blade element.

At the exit from the runner:

$$\vec{V} + [(C_u)_{g1} + (C_u)_f] = \vec{W}_2 + r\omega$$

where W_2 is the relative velocity at the exit of the blade element.

At the entrance to the rear guide vanes:

$$\vec{V}+[(C_u)_{g1}+(C_u)_f]=\text{absolute velocity}$$

At the exit from the rear guide vanes:

$$\vec{V}+[(C_u)_{g1}+(C_u)_f+(C_u)_{g2}]=\text{absolute velocity}$$

For Case I (Fig. 1):

$$-(C_u)_{g1}=(C_u)_f$$

$$(C_u)_{g2}=0$$

For Case II (Fig. 2):

$$(C_u)_{g1}=0$$

$$-(C_u)_{g2}=(C_u)_f$$

For Case III (Fig. 3):

$$-(C_u)_{g1}=k(C_u)_f$$

$$-(C_u)_{g2}=(1-k)(C_u)_f$$

$$-[(C_u)_{g1}+(C_u)_{g2}]=(C_u)_f$$

Figs. 4 and 5 show side and rear views of a typical blade while Figs. 6 and 7 show the blade profiles along the shortest and longest radii respectively. The plane of rotation is the plane of the paper for Fig. 5, is normal to the plane of the paper in Fig. 4, and lies in the horizontal for Figs. 6 and 7.

The following is a description of the application of the formulas described herein to the development of the typical blade illustrated in Figs. 4 to 7.

Consider that a blade form is to be selected for an axial fan unit to meet the following requirements:

Q=10,000 cu. ft./min.
p=3 inches water=15.6 lbs./sq. ft.
R=12½ inches
ωR=125 ft./sec.
A=2.33 sq. ft.
e=.0024 slugs/cu. ft.
λ=.57
ψ=.83
η=83%

Calculation of blade angles for Case I, guide vanes upstream.

At blade tip, R=12½ inches and the angle of relative flow at blade entrance is given by the formula of page 2 column 1 line 30:

$$\tan\beta_1=\frac{\lambda}{\left(\frac{r}{R}+\frac{\psi}{\eta}\frac{R}{r}\frac{1}{2}\right)}=.38$$

$$\beta_1=20.8°$$

At blade root, r=7 inches

$$\text{Tan } \beta_1=.393$$
$$\beta_1=21.5°$$

For quiet fan operation, choose low values of the lift coefficient, C_L, say 0.5 at R=12.5 inches and 0.8 at r=7 inches.

The angle of attack α of the blade entrance with respect to the relative flow may be taken from published curves of profiles well known to the art, for instance the N. A. C. A. 2409, see National Advisory Committee for Aeronautics, Report No. 460, Fig. 17. The corresponding angles determined from said curves are 3° for C_L=0.5 and 6° for C_L=0.8.

The blade entrance settings are:

$\beta_1+\alpha=20.8°+3°=23.8°$ at R=12½ inches
$\beta_1+\alpha=21.5°+6°=27.5°$ at r=7 inches

The blade exit settings are defined by the formula of page 1 column 2 line 50:

$$\text{Tan } \beta_2=\lambda R/r$$

for the present example the values are:

Tan β_2=.57 β_2=29.7° at R=12½ inches
Tan β_2=1.02 β_2=45.6° at r=7 inches

The corresponding profile shapes and settings are shown in Figs. 4 to 7 inclusive, Fig. 6 representing a typical section at r=7 inches and Fig. 7 showing the corresponding blade section at R=12½ inches.

It is to be understood that the invention is limited only by the scope and limitations of the appended claims.

The invention described herein may be manufactured and/or used by or for the Government of the United States of America for governmental purposes without the payment of any royalties thereon or therefor.

I claim:

1. An axial flow type fan or pump comprising a multi-bladed runner and guide vanes located upstream of said runner, the elements of said blades being curved in such a manner that the tangent to the zero lift line of any blade element forms at the leading edge of said element an angle $(\beta_1+\alpha)$ with the plane of rotation, such that;

$$\tan\beta_1=\lambda\frac{1}{\left(\frac{r}{R}+\frac{\psi}{2\eta}\frac{R}{r}\right)}$$

and at the trailing edge thereof an angle β_2, such that;

$$\tan\beta_2=\lambda\frac{R}{r}$$

where

$$\lambda=\frac{Q/A}{\omega R}$$

$$\psi=C_L S(r/R)\lambda\left(\frac{1}{\sin\beta}\right)\eta=\frac{p}{\frac{1}{2}\rho(\omega R)^2}$$

Q=quantity of fluid delivered per unit time
A=net area of the fluid passage through the fan proper
ω=angular velocity
r=radius of any blade element
R=radius of the tip of any blade
η=total efficiency of the fan
β=the mean pitch angle of the blades
S=the ratio of the blade chord to the spacing between two blades
ρ=the mass density of the fluid operated on by the fan
C_L=the lift coefficient, and
p=the total head developed by the fan or pump, the angle α being determined by the expression C_L=B2π sin α, where B varies with the shape of the selected profile.

2. An axial flow type fan or pump comprising a multi-bladed runner and guide vanes located downstream of said runner, the elements of said blades being curved in such a manner that the tangent to the zero lift line of any blade element forms at the leading edge of said element an angle $(\beta_1+\alpha)$ with the plane of rotation, such that;

$$\tan\beta_1=\lambda\frac{R}{r}$$

and at the trailing edge an angle β_2, such that;

$$\tan\beta_2=\lambda\frac{1}{\left(\frac{r}{R}-\frac{\psi}{2\eta}\frac{R}{r}\right)}$$

where the meanings of the terms of said equations are as defined in claim 1.

3. An axial flow type fan or pump comprising a multi-bladed runner and guide vanes located both upstream and downstream of said runner, the elements of said blades being curved in such a manner that the tangent to the zero lift line of any blade element forms at the leading edge of said element an angle $(\beta_1+\alpha)$ with the plane of rotation of said runner, such that;

$$\tan \beta_1=\lambda\frac{1}{\left(\frac{r}{R}+k\frac{\psi}{2\eta}\frac{R}{r}\right)}$$

and at the trailing edge an angle β_2, such that;

$$\tan \beta_2=\lambda\frac{1}{\left(\frac{r}{R}+(k-1)\frac{\psi}{2\eta}\frac{R}{r}\right)}$$

where the meanings of the terms of said equations are as defined in claim 1, the term k being a constant depending on the distribution of the rotation added to the fluid by said upstream vanes and subtracted by said downstream guide vanes.

FRANK L. WATTENDORF.