TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 927

CONSTANT-PRESSURE BLOWERS

By E. Sörensen

Zeitschrift des Vereines Deutscher Ingenieure
Vol. 83, No. 32, August 12, 1939

Washington
January 1940

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 927

CONSTANT-PRESSURE BLOWERS*

By E. Sörensen

The conventional axial blowers operate on the high-pressure principle. One drawback of this type of blowers is the relatively low pressure head, which one attempts to overcome with axial blowers producing very high pressure at a given circumferential speed. The Schicht constant-pressure blower affords pressure ratios considerably higher than those of axial blower of conventional design with approximately the same efficiency.

INTRODUCTION

The same principle is applied to axial and radial fans as to water turbines and centrifugal pumps.(figs. 1 and 2). For a specified delivery volume, the axial blower is much smaller in diameter and consequently lighter and less expensive than the radial type. Being, in addition, easy to install in many cases, its general use is much favored. Unfortunately, its range of application is quite restricted, since it can overcome only small pressure difference.

According to the laws of hydraulic similitude, the delivery head or lift H produced by a blower (in m) is proportional to the square of the circumferential speed u_a of the blade (in m/s).

$$H = \psi \frac{u_a^2}{2g}$$

The nondimensional quantity ψ denotes the so-called pressure ratio of the blower, because the delivery head usually agrees with the pressure difference $\Delta p/\gamma$ (γ =

---

specific weight of delivered gas) produced in the blower inclusive of diffuser. Hence the produced pressure in any blower is controllable within wide limits by changing the rotational speed. For this reason, the mere statement of the pressure produced in a blower is insufficient to characterize its value or special features. The required circumferential speed or the pressure ratio must also be stated.

The pressure ratio of a blower depends, first, on the shape and number of the guide and rotor blades, and secondly, on the method of arrangement of guide vanes and rotor. To illustrate: Figure 1 shows a radial blower and figure 2, an axial blower with the guide vanes in flow direction behind the rotor. Figure 3, on the other hand, shows guide vanes fitted before and behind the rotor. The pressure ratios obtainable with such blower are very unlike. To illustrate this fact, let us assume four bladings for an axial blower:

Rotor A, with blade angle of 90° at discharge;

Rotor B, the blading of which completely cancels a certain helix in the inflowing air;

Guide vanes C, the blading of which completely neutralizes the helix created by rotor A;

Guide vanes D, whose blades can produce a helix equal to the helix behind rotor A.

The losses are assumed as follows:

The blade losses are zero;

The axial component of the discharge velocity is without loss converted to pressure in the following diffuser;

The circumferential component of the discharge velocity is lost completely.

On these premises, the pressure ratio is as follows:

For blade arrangement A + C (guide vanes behind rotor) at $\psi = 2$;

For blade arrangement D + B (guide vanes before rotor) at $\psi = 2$;

For blade arrangement   D + A   (guide vanes before rotor) at $\psi = 3$;

For blade arrangement   D + A + C   (guide vanes before and behind rotor) at $\psi = 4$;

For blade arrangement   D + A + A + C   (two oppositely rotating rotors; guide vanes before and behind rotors) at $\psi = 8$.

Fundamentally, arrangements   D + B   and   D + A   are identical (guide vanes before rotor), differing only in the blade angles.   But arrangement   D + A   purchases the higher pressure ratio with supplementary losses.   Even so, somewhat higher pressure ratios are attainable in practice with guide vanes fitted before the rotor without impaired efficiency than with guide vanes arranged behind.

The greatest drawback of the axial against the radial blower so far is its very low pressure ratio.   Its pressure for a given circumferential speed is much lower than that of the radial blower.   Only a small part of the delivered gas passes the rotor of the axial blower entirely outside where its circumferential speed u_a is maximum. Another portion passes inside along the hub where the blading has its lowest circumferential speed, and this low speed must suffice to produce the required delivery head. The axial rotor could produce a far higher delivery head outside than inside.   Since, for reasons of good efficiency, the delivery head must evidently be the same for every part of the blade, the circumferential speed u_a cannot be utilized at all for producing pressure.   Logically, the pressure ratio of the axial blower should be referred to its circumferential speed at the hub.

In the radial blower, the total delivery discharges outside on the rotor circumference.   Thus, every gas particle has the speed u_a available for producing pressure. In this case, ψ must be absolutely referred to u_a.   But the value ψ for the axial blower is also always computed with reference to u_a.   For this very reason, the pressure ratio and consequently the produced pressure is always much lower than on the radial blower.

This drawback of the axial blower could be balanced by correspondingly higher circumferential speed if the problems of strength and noise did not intervene.   The higher the circumferential speed of the rotor the greater

the stress due to centrifugal force. Better material
would have to be used, thus raising the cost of the blower.
The noise at high speed may reach proportions which make
operation utterly impossible.

Hence, to succeed in raising the pressure ratio would
mean a substantial step forward. This was temporarily
achieved by enlarging the hub diameter. Then u_i itself
increases and is not quite as much at variance with u_a.
In consequence, ψ itself must increase. The hub diameter
D_i on modern high-pressure axial blowers is usually much
greater than half the outside diameter

$$D_a: 0.5 < \frac{D_i}{D_a} < 0.7.$$

A further substantial raise in pressure ratio accrues
from the application of the constant-pressure method to
the axial blower.

PHYSICAL PRINCIPLES

Figure 3 is a section through an axial blower with
rotor located between two guide vanes. It serves to illus-
trate the most general case.

The flow energy of the delivered gas consists of the

speed energy $\frac{c^2}{2g}$ and the pressure energy $\frac{p}{\gamma}$, amounting

at point 1 (before the front guide vanes) to figure 3.

$$H_1 = \frac{c_1^2}{2g} + \frac{p_1}{\gamma}$$

The speed c consists of the velocity component c_m,
shown in figure 3, and the circumferential component c_u
at right angle to it. Since the gas flow approaches the
first guide vanes axially, it is

$$c_{1u} = 0 \quad \text{and} \quad c_1 = c_{1m}$$

In the first guide vanes the gas is deflected, creating a
circumferential component c_{2u}, which is usually opposite
to the circumferential speed u of the rotor. Since the

speed increases in these guide vanes $(c_2 > c_1)$, whereas the flow energy, discounting the losses, does not change, the pressure must drop $(p_2 < p_1)$.

In the rotor the energy (delivery head) H is transmitted to the gas; whence

$$\frac{c_3{}^2}{2g} + \frac{p_3}{\gamma} = \frac{c_2{}^2}{2g} + \frac{p_2}{\gamma} + H$$

In the conventional blowers the speed as well as the pressure energy $(c_3 > c_2; p_3 > p_2)$ increases in the rotor. Fundamentally, it is equally possible that one part of the energy in the rotor remains constant or even decreases, while the other part then must increase so much more. Thus it happens, for instance, on blowers with guide vanes mounted ahead of the rotor that $c_3 < c_2$. On the contrary, it never happens that $p_3 \leqq p_2$ with the customary axial blowers.

If the guide vanes follow the rotor, the circumferential component c_{3u} cancels, hence $c_{4u} = 0$ and $c_4 = c_{4m}$. The pressure of the delivered gas usually rises in these guide vanes, $(p_4 > p_3)$. In the adjoining diffuser, the speed c_4 is largely transformed in pressure; hence also a pressure rise.

By degree of reaction R of a blower is meant the ratio of pressure difference achieved in the rotor to the total delivery head of the blower. Disregarding the losses, it is

$$R = \frac{p_3 - p_2}{\gamma} : H.$$

The greater this ratio R is, the greater the pressure directly produced in the rotor, the smaller at the same time the portion of the pressure to be produced by conversion of the speed in the adjoining guide vanes and in the diffuser. The smaller the ratio R the higher the load to be imposed on the diffuser.

Since the conversion of speed into pressure in a diffuser is invariably accompanied by fairly great losses, there has been a tendency until quite recently toward a very high degree of reaction, in spite of being aware of the increasing difficulties created in the rotor (reference 1).

According to the subsequent pressure rise, equation
(4), a pressure difference $p_3 - p_2$ in an axial rotor can
occur only if the relative speed w of the delivered gas
decreases while passing through the rotor. The relative
speed w_2 at entry into the rotor must exceed the speed
w_3 at the exit. In consequence, the rotor blades operate
in a simultaneously deflected and retarded flow, which is
known to have a special tendency to break away at the rotor
blades. To forestall this break-away of flow for reasons
of efficiency, high blade loading must be avoided, i.e.,
no unusually high delivery head should be demanded from
the blades for a specified circumferential speed. The
rotors of modern axial blowers are usually computed with
the help of the two-dimensional airfoil theory, for which
a great number of lift and drag test data are available.
The findings of such tests are plotted in polar diagrams
which give the relation between lift coefficient c_a and
drag coefficient c_w (a in fig. 4). c_a and c_w are
directly proportional to the lift and drag and are depend-
ent on the angle of attack δ. The test data for the same
profile in retarded flow are also shown (b in fig. 4).
It discloses the important fact that both curves are ap-
proximately in agreement within the range of small δ. As
δ increases, the break-away of flow on the airfoil back
occurs as a result of unduly strong retardation on the air-
foil in the retarded flow. The lift increase has practi-
cally ceased, but the drag increases so much that this an-
gle δ becomes useless for practical operating conditions.

The break-away on the same airfoil in normal, i.e.,
not retarded flow, does not occur before considerably
higher δ and c_a. Since c_a is practically proportion-
al to the delivery head of the blower, it indicates that
the rotor blade of an axial blower can produce a substan-
tially greater delivery head in normal flow than in re-
tarded flow.

Of course, the delivery head is also proportional to
the blade number z, which might suggest the compensation
of the lower lift coefficient c_a in the rotor as a re-
sult of retarded flow by a greater number of blades. But
every blade added increases the frictional surface of the
rotor. Besides, the flow is also retarded on the rotor
hub and on the blower wall. The break-away resulting
therefrom sets an insuperable limit for the pressure rise
in the rotor. In any attempt at further increase of de-
livery head, the creation of pressure in the rotor must be
abandoned and the entire pressure production relayed in the

adjoining guide vanes and diffuser. The result is then
what is termed a constant-pressure blower, or, named after
its inventor: Schicht blower.

In itself the idea of using constant pressure for
pumps or blowers is not new. For instance, the celebrated
French engineer, Rateau, points to this possibility and
stresses the great importance of the diffuser (reference
2), although he himself never built constant-pressure
blowers. Around 1890 he experimented on blowers with
rotors having axial inlet, radial discharge and 45^0 for-
ward curved blades (reference 3). These blowers had a
vaneless circular chamber with adjoined spiral as diffuser.
They were to produce 28 percent of the pressure head in
the rotor and the rest in the diffuser. But the experi-
mental results were so unsatisfactory that Rateau abandoned
this type of design and warned against it.

The cause for this failure is to be found in the fact
that the application of a comparatively low degree of re-
action in radial wheels resulted in very great losses be-
cause of the particularly difficult pressure conversion in
the diffuser. In his later axial blowers and pumps, Rateau
did not follow the concept of low degree of reaction, but
built these machines so as to produce a large portion of
the pressure energy in the rotor. Subsequently, the appli-
cation of constant pressure in blower design was then gen-
erally viewed as being impossible.

The new invention has disproved this prejudice and at
the same time given the means for designing constant-pres-
sure blowers of high efficiency. These means are, first,
the application of the axial blower with its particularly
favorable diffuser and secondly, the reference to the fact
that equal pressure must be maintained before and behind
the rotor and that the pressure must be actually kept con-
stant for every point of the rotor passages. These claims
are incorporated in the basic patent (German Patent No.
633,155). Experiments have proved that failure to comply
with the second requirement impairs the efficiency in un-
supportable manner.

The described ideas can be expanded in two ways.
First, the fact of the case is that the best efficiency is
not always precisely attained just when $R = 0$. The best
value for R may be greater or even less than zero. While
with $R > 0$ the known blower versions are approached, the
application of $R < 0$ (low-pressure blower) is entirely

new territory. Here the experiments are, on the other
hand, not without prospects, since the delivery head ob-
tainable at low pressure will probably grow. The limit
of application of low pressure in the rotor lies in the
steadily increasing diffuser losses.

MATHEMATICAL PRINCIPLES

According to Euler's fundamental equation of the
power transfer in turbines, the delivery head H_{La} pro-
duced in the rotor is:

$$H_{La} = \frac{c_3^2 - c_2^2 + u_3^2 - u_2^2 + w_2^2 - w_3^2}{2 g} \tag{1}$$

The rotor lift H_{La} is greater than the effective
blower lift H, which has to cover the flow losses V_{1-5}
in the guide vanes, the rotor, and the diffuser.

As concerns the rotor itself, one part of the trans-
mitted delivery head must cover the rotor losses V_{2-3};
a second part is measurable as speed increase and the rest
as pressure increase in the rotor:

$$H_{La} = V_{2-3} + \frac{c_3^2 - c_2^2}{2 g} + \frac{p_3 - p_2}{\gamma} \tag{2}$$

The comparison of equations (1) and (2) gives the
pressure rise in the rotor

$$\frac{p_3 - p_2}{\gamma} = \frac{u_3^2 - u_2^2 + w_2^2 - w_3^2}{2 g} - V_{2-3} \tag{3}$$

Discounting the minor discrepancies in the circumferential
speed and the rotor losses, leaves

$$\frac{p_3 - p_2}{\gamma} = \frac{w_2^2 - w_3^2}{2 g} \tag{4}$$

In the axial blower, a pressure rise in the rotor is con-
tingent upon retardation of the relative speed in the
rotor.

In the constant-pressure blower, retardations in the rotor are to be avoided, hence the speed remains the same. This requirement is extended to include a specified constant degree of reaction in the blade duct which may be prescribed arbitrarily. This demand leads to a condition for the sectional variation in the rotor passages which is to be based upon the stream filament rather than airfoil theory. The rotor blades are assumed to be so close together as to reflect blade passages (fig. 5).

The delivery head produced up to any point P is:

$$H_P = \frac{1}{g}\,(u\,c_u - u_2\,c_{2u})$$

$$= \frac{1}{g}\,[u(u - w\cos\beta) - u_2(u_2 - w_2\cos\beta_2)] \qquad (5)$$

while the pressure difference produced up to point P is, according to equation (3):

$$\frac{p - p_2}{\gamma} = \frac{u^2 - u_2{}^2 + w_2{}^2 - w^2}{2\,g} - V_{2-P} \qquad (6)$$

The degree of reaction at this point is:

$$R = \frac{p - p_2}{\gamma} : H_P$$

The values of equations (5) and (6) when entered into this formula give a squared equation for w; the result is:

$$w = \frac{Q}{z\,F} = R\,u\cos\beta$$

$$+ \sqrt{R^2\,u^2\cos^2\beta + (u^2 - u_2{}^2)(1 - 2R) + w_2{}^2 - 2Ru_2w_2\cos\beta_2 - 2gV_{2-P}} \qquad (7)$$

Q is the gas volume (in m³/s) delivered by the rotor; z, the number of blades; and F, the inside section of a blade passage (in m²), measured at right angle to w. Equation (7) represents a relation between w or F and the blade angle β, which must be preserved if the required degree of reaction is to prevail at every blade point. If R = 0, $V_{2-P} = 0$, and $u_2 = u$ (loss-free axial constant-pressure blower), we are back to the foregoing condition

of $w = w_2$. $R < 0$ written in equation (7) affords the cross-sectional change for a low-pressure blower.

The condition $w_2 = w = w_3 = const$ leads directly to the rotor form typical for the constant-pressure blower. Figure 6 illustrates tht speed triangles at rotor inlet and outlet for a constant-pressure blower with preceding guide vanes, and figure 7 with following guide vanes. Both manifest a strong increase of the meridian component c_m of the speed in the rotor. The same holds in slightly smaller degree, for the guide vanes. In the conventional axial blowers c_m is usually considered constant. Figure 7 shows the discharge triangle for such blower as dash-dotted lines. Such blowers have in consequence constant blade length, wheras that in the constant-pressure type decreases considerably in flow direction (fig. 8).

Owing to the very small blade length of the constant-pressure blower at the rotor outlet, its hub is usually much thicker than in a normal blower, D_i/D_a to 0.9. Hence the circumferential speed at that point is comparatively high. But, as already pointed out, the circumferential speed at the hub at the rotor outlet decides the attainable delivery head in the rotor. Hence, the very thick hub of the constant-pressure blower also increases the delivery head over that of a normal blower.

On passing through the blower, the delivered gas contains a changing helix, i.e., a rotatory motion, which, because of the centrifugal force, induces an outwardly increasing pressure. Fundamentally, the demand for constant pressure is restricted to only one stream filament. On all others, the pressure must change at passing, increasing or decreasing depending on the assumptions in the design. Every blower having constant pressure in the median stream filament has at the same time in the other parts of the blades a slightly positive as well as negative degree of reaction. These facts must be taken into consideration in a detailed analysis.

Changes in rotor-blade length produce differences in the circumferential speed, at the same time as c_m is then no longer purely axial, but rather contains a radial component. In an accurate analysis, the blades must then be divided by streamlines in parts of narrow width, as is customary in water-turbine design, however, with the added condition of continuous compliance with equation (7). This is especially conducive to experimentation. And it

affords, as in water turbines, spatially curved blades of known method of presentiation. But the extent to which such blades are feasible from the point of view of centrifugal force is, of course, another question. All these difficulties disappear when the blades can be computed sufficiently approximately with disregarded radial extension, as is usually the case.

The frequently quite important discrepancies in rotor blade length at the inlet and outlet sides impose a fairly great axial depth t, (fig. 8). The result is a longer blade passage which would justify the application of the stream filament theory for the calculation. Nevertheless, the blowers are generally analyzed by airfoil theory if test data on profiles resembling the required blade form are available.

Of particular interest in this connection is the use of offset rotor blades obtained from a greatly curved blade conformable to figures 9 and 10 by cutting it up in radial planes and arranging the separate pieces over the rotor circumference. It affords slightly curved blade profiles, each of which deflects the flow a little. The individual blade pieces themselves are always in a sound flow, because the offset has placed them out of the boundary-layer zone of the preceding piece. Each of these blade pieces is calculable by airfoil theory so that breakaway phenomena are safely avoided. At the same time, the blading can be laid out so that as a whole equation (7) is approximately satisfied. Complete satisfaction is not possible, because the premises of stream filament and airfoil theory are not the same. Rotors with offset blades are shown in figures 11 and 12. The upper limit of delivery head obtainable in one stage lies in the critical speed. But whether this limit is reached sooner by the constant-pressure blower than by the normal blower, is possible but not certain. Comparative tests are necessary.

DIFFUSER

The diffuser is the most vital part of the constant-pressure blower, and the efficiency of the whole blower is chiefly dependent upon the losses in it. It also forms the limit for the application of the negative degrees of reaction since through them the gradient to be converted in the diffuser is increased. However, it is of great

importance that the efficiencies of carefully designed diffusers are considerably higher than most engineers assume. Judged on the basis of all the data gathered from tests on Kaplan turbines and constant-pressure blowers it may be stated that the diffuser efficiency may certainly exceed 90 percent. Far more difficult is the control of the susceptibility of the diffuser to inlet disturbances.

Diffuser design should proceed from the following viewpoints: The cross-sectional enlargement should not be too abrupt; a small residuary helix in the gas leaving the rotor or guide vanes reduces the break-away tendency on the diffuser wall; it is useless to try to achieve the cross-sectional enlargement through a pointed hub, where almost always break-away occurs. The hub on the constant-pressure blower is therefore as a rule cylindric or slightly tapered behind the rotor and cut off at the diffuser end (fig. 3).

The diffuser of the constant-pressure blower naturally requires a great structural length, and herein lies a serious drawback of this blower. It imposes the single-stage type, which in turn is a profound limitation of its range of application, since high-pressure blowers must have more than one stage. On the other hand, it has been found that the diffuser length in many cases does not interfere, since passages for the delivered gas must be joined any way (figs. 13 and 14).

DESIGN

One substantial advantage of the constant-pressure blower lies in its cheap manufacture, since, rotor included, it can be welded almost wholly from sheet metal. The rotor blades are shaped by jig and welded to the welded metal hub. The same, of course, holds for guide vanes and diffuser. The Schicht blower produces a specified pressure at lower circumferential speed than any other axial blower. Its diameter and weight can therefore be kept smaller for a given speed, and the centrifugal forces themselves are not excessive. It is not necessary to have the rotor slot on the circumference particularly small, because the slot losses are very low on account of the constant pressure. The blower drive may, if necessary, be housed in the hub.

The use of sheet-metal blades in place of the usual faired rotor blades involves no perceptibe loss in efficiency for the following reason. The flow resistance of the sheet-metal blades of itself need not be greater than on the faired blades. The pure rotor blade loss in any blower is but a fraction of the total loss, so that this slightly greater loss is almost imperceptible. The use of airfoil sections in the other blowers heightens the danger of break-away in the retarded flow. For the same reason, the surface condition of the rotor blades so important otherwise, is secondary on the constant-pressure blower.

EXPERIMENTAL DATA

Several constant-pressure-blower designs are shown in figures 11 to 14; they exemplify the typical decline in radial blade depth toward the rotor discharge, the large diffuser, and the very thick hub. The blower is practical wherever the required delivery pressure can be obtained in one stage at, of course, very high pressure ratio. Higher temperature or dust content in the delivered gas does not restrict its use.

Naturally, the development of the blowers proceeded on the basis of a great number of tests. The writer himself investigated two constant-pressure blowers according to the specifications for the power and acceptance tests on compressors (German Standard Specifications 1945); one intended for a mine, the other for an exhausting plant (figs. 14, 15, and 16). In the induced draught blower it was impossible to record the pressure before and behind the blower directly onto the blower itself. The flue gases are deflected by two rectangular elbows, containing deflectors (fig. 14). The pressure loss at these points could not be measured separately, hence the recorded values of pressure difference, pressure factor, and efficiency (plotted in figure 16) are lower than in reality, amounting perhaps to several percent.

The results indicate that efficiencies in excess of 80 percent can be reached with the constant-pressure blower. Such values are equally obtainable with the best blowers of other types. But the pressure of considerably more than 0.8 reached with the constant-pressure blower is not likely to have been equaled so far. On top of that, it should be remembered that in both blowers the

guide vanes are fitted aft, whereat the pressure ratio
in itself is a little lower than when the guide vanes
are fitted before the rotor. It is certain that pres-
sure ratios of more than 1 are obtainable by optimum effi-
ciency. The inventor himself reached a value ψ = 1.4,
whereby, to be sure, the efficiency dropped to around
60 percent.

Other experiments of the inventor dealt with the
effect of the speed and pressure variation in the rotor
on the efficiency. If the rotor is so computed that the
section at inlet and outlet complies with the constant-
pressure condition (equation (7)), where R = 0, while
the sections within the rotor do not comply with it, the
efficiency declines immediately. This is associated with
the fact that then the delay at isolated points in the
rotor passages is such as to induced break-away of flow.

The effect of blade offset on the efficiency was also
analyzed. This effect is greater than at first assumed
and may cause material differences in efficiency and in
the characteristic curves.

It is difficult to make any general statement about
the noise in constant-pressure blowers, because definite
conclusions must be based on a comparison with a blower
of normal design under identical installation and operat-
ing conditions. At any rate, it may be stated on the
basis of many past experiences that the constant-pressure
blower, is less noisy than the usual axial blower, a fact
which is no more than expected when bearing in mind its
much lower circumferential speed than other axial blowers
of equal delivery head.

APPLICATION OF CONSTANT PRESSURE IN PUMP DESIGN

In principle, the constant-pressure method is as ap-
plicable to the delivery of liquids as of gases. In this
manner, it is possible to obtain particularly high lifts
with axial pumps. But so far no experiments have been
made in this direction. The reason for this was, first,
that the range of the blower, because of its great cir-
cumference, required all energy force, and secondly, be-
cause there is apparently no great demand for axial pumps
of high lift. In any case, the constant pressure process
promises here also considerable increase in design possi-
bilities.

In experiments with constant-pressure axial pumps the cavitation hazard is paramount, since the constant-pressure blading creates the entire delivery head in the rotor as speed energy, and hence particularly high speeds in the rotor without rise in hydraulic pressure. Although this does not necessarily entail cavitation, general experience teaches that cavitation hazard is imminent. The reason for this is that speed increases occur in every flow if the flow direction is changed or if the flow is around a body (rotor blades). This increase of speed is always proportional to the mean speed. Hence the resulting low pressures are proportional to the square of the high mean speed, hence they themselves must be very high. For that reason the suction capacity of a constant-pressure pump will probably not be as high as that of a pump of conventional design. Experiments in this direction should produce some very interesting results.

Translation by J. Vanier,
National Advisory Committee
for Aeronautics.

REFERENCES

1. Keller, B. C.: Axialgebläse, Zürich, 1934, and

   Eck, B.: Ventilatoren, Berlin, 1937.

2. Rateau, A., Eydoux, D., and Gariel, M.: Turbines Hydrauliques, Paris, 1926, p. 408.

3. Bull. Soc. Ind. Minér. Saint-Etienne, vol. 6, 1892, p. 144.
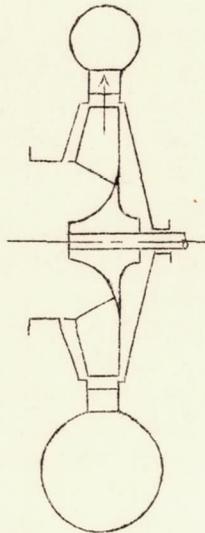
Figure 1.- High-pressure radial blower.
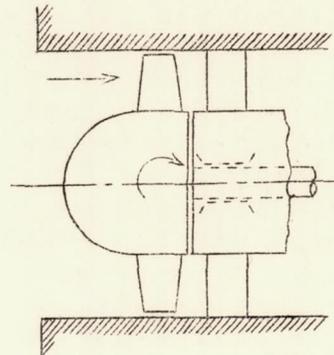
Figure 2.- High-pressure axial blower.
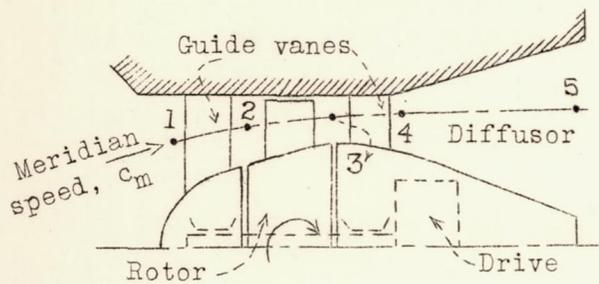
Figure 3.- Axial blower with two guide vanes.
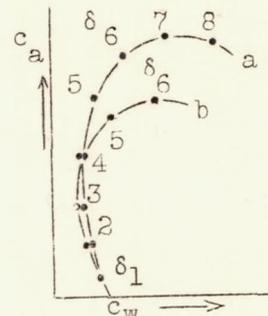
Figure 4.- Polar diagram. a = normal flow b = retarded flow
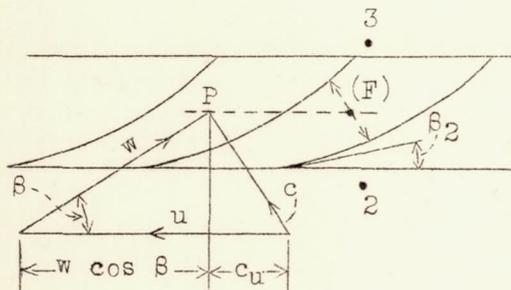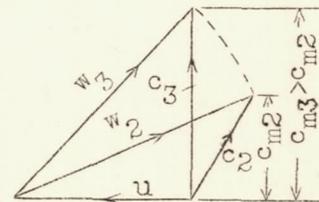

Figure 5.- Speed triangle in blade passage.

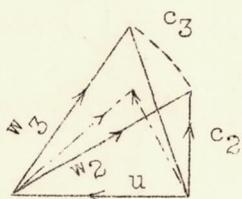Figure 6.- Constant-pressure blower with preceding guide vanes.

Figure 7.- Constant pressure
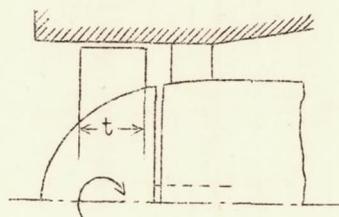   blower with adjoin-
ing guide vanes.

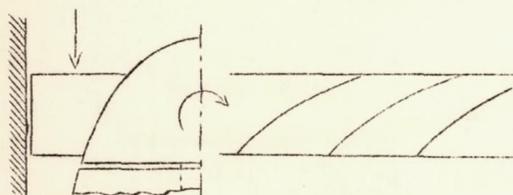Figure 8.- Constant pressure
        blower.
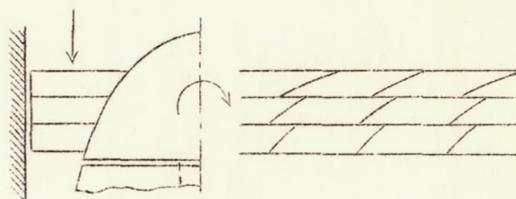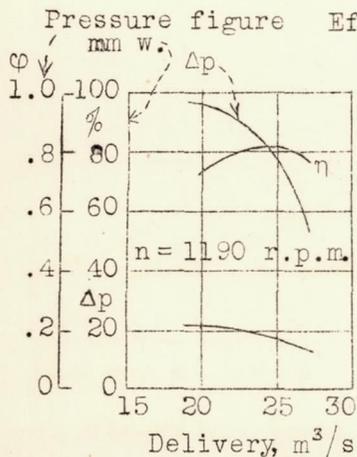   t = blade width

Figure 9.- Rotor with single
        row blades.

Figure 10.- Rotor with offset
        blades.

Pressure figure   Efficiency, Pressure difference, Δp

n = 1190 r.p.m.

n = 200 r.p.m.

Delivery, m³/s

Delivery, m³/s
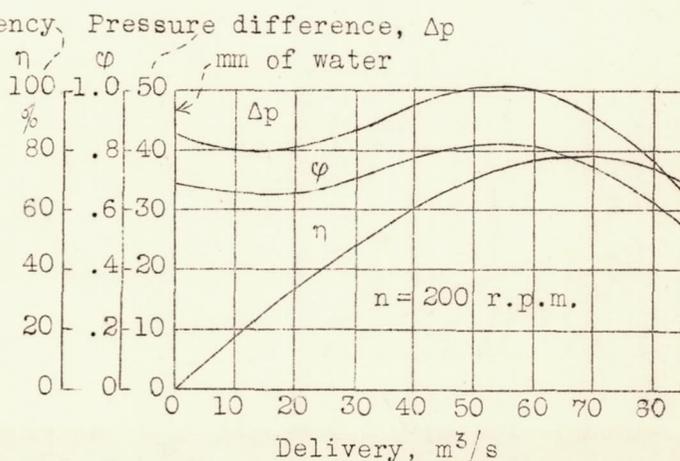
Figures 15,16.- Operation characteristic curves of two constant-press-
        ure blowers with adjoining guide vanes.
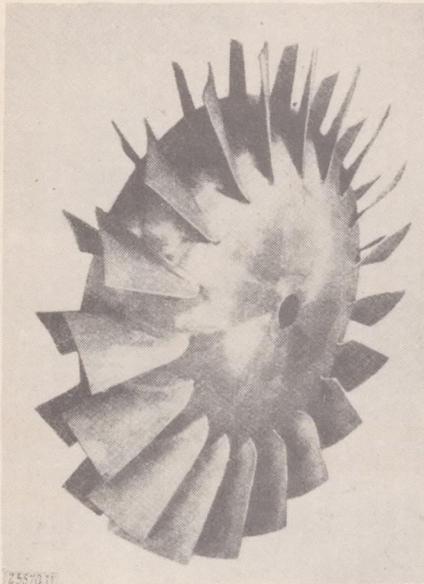
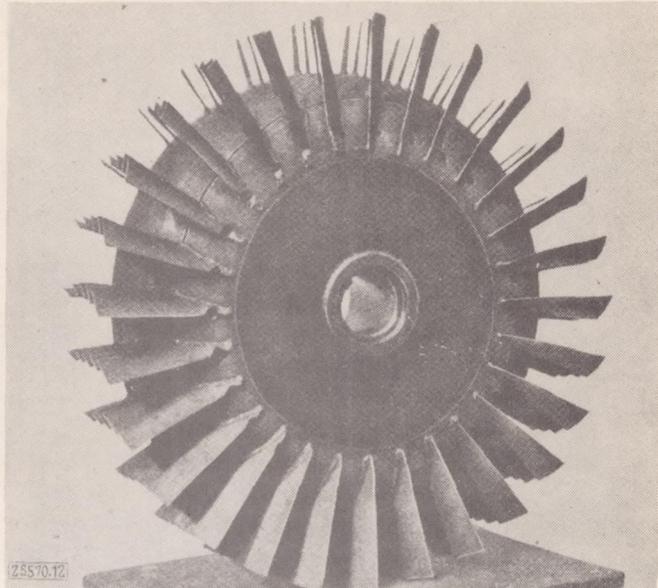Figure 11.- Two-row rotor of a constant-pressure blower.

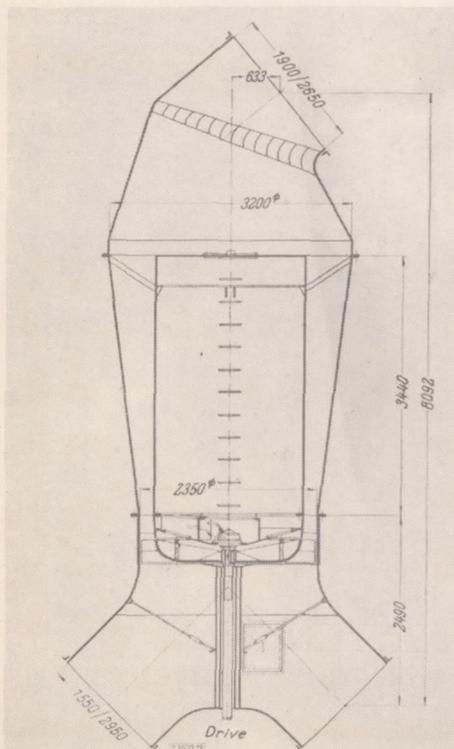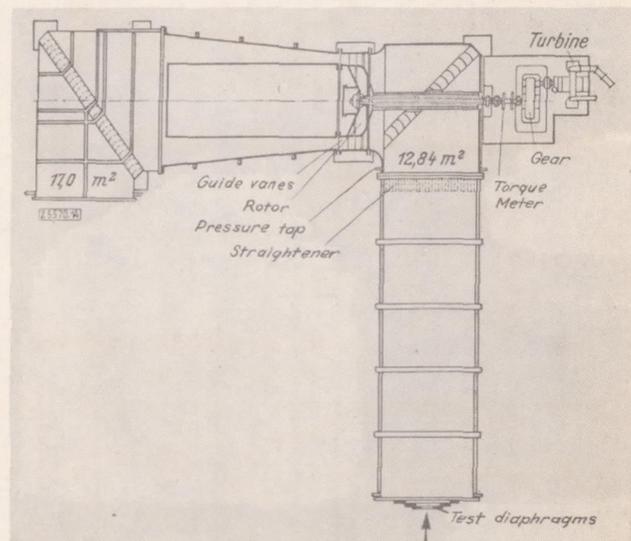Figure 12.- Five-row rotor of a constant-pressure blower.

Figure 14.- Induced draught constant-pressure blower.

Figure 13.- Constant-pressure blower for steam boiler (fuel gas 70 m /s, delivery head 120 mm , r.p.m. 585, power input 106 kw).