Fan Engineering 1840-1930

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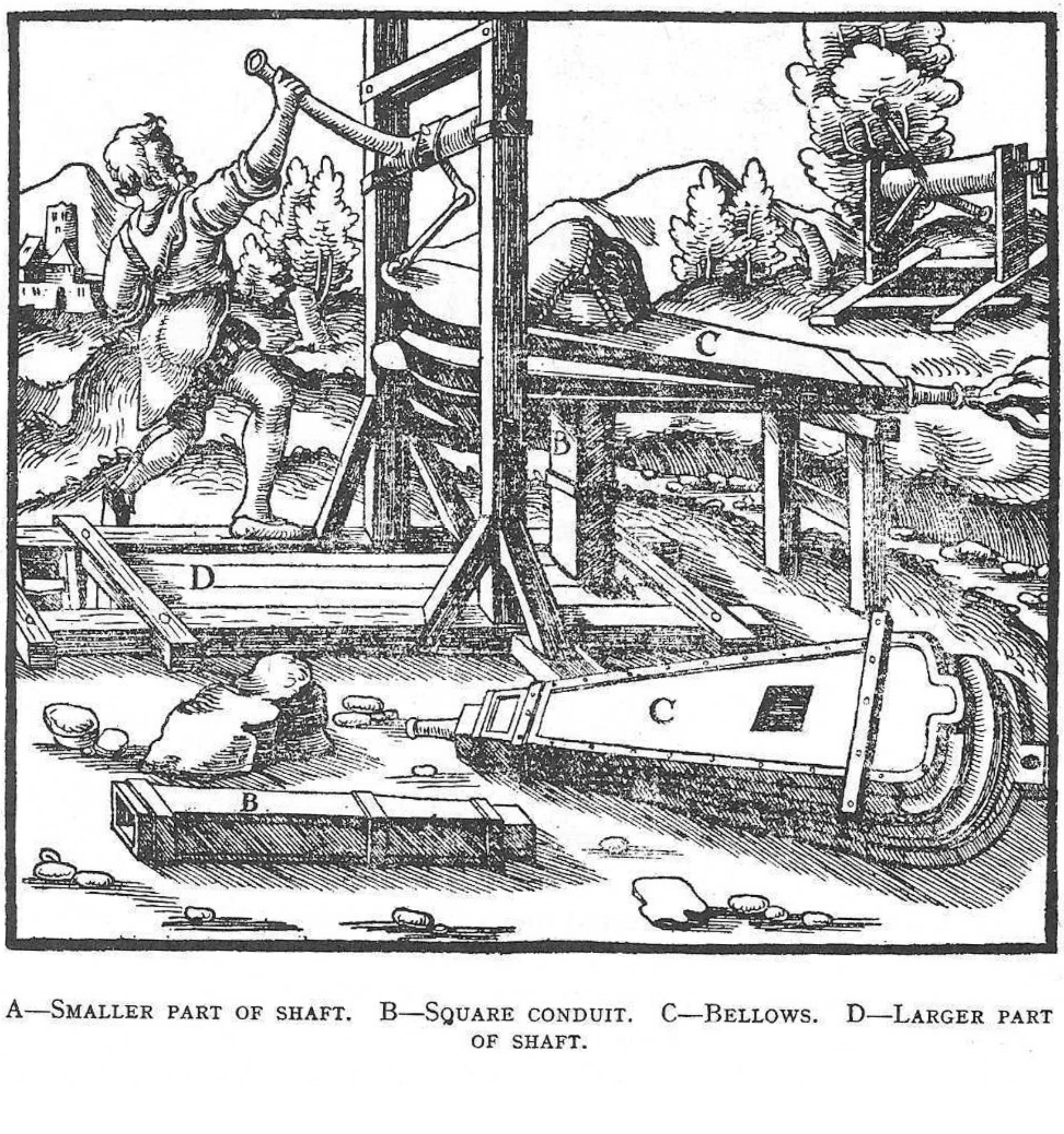
Introduction

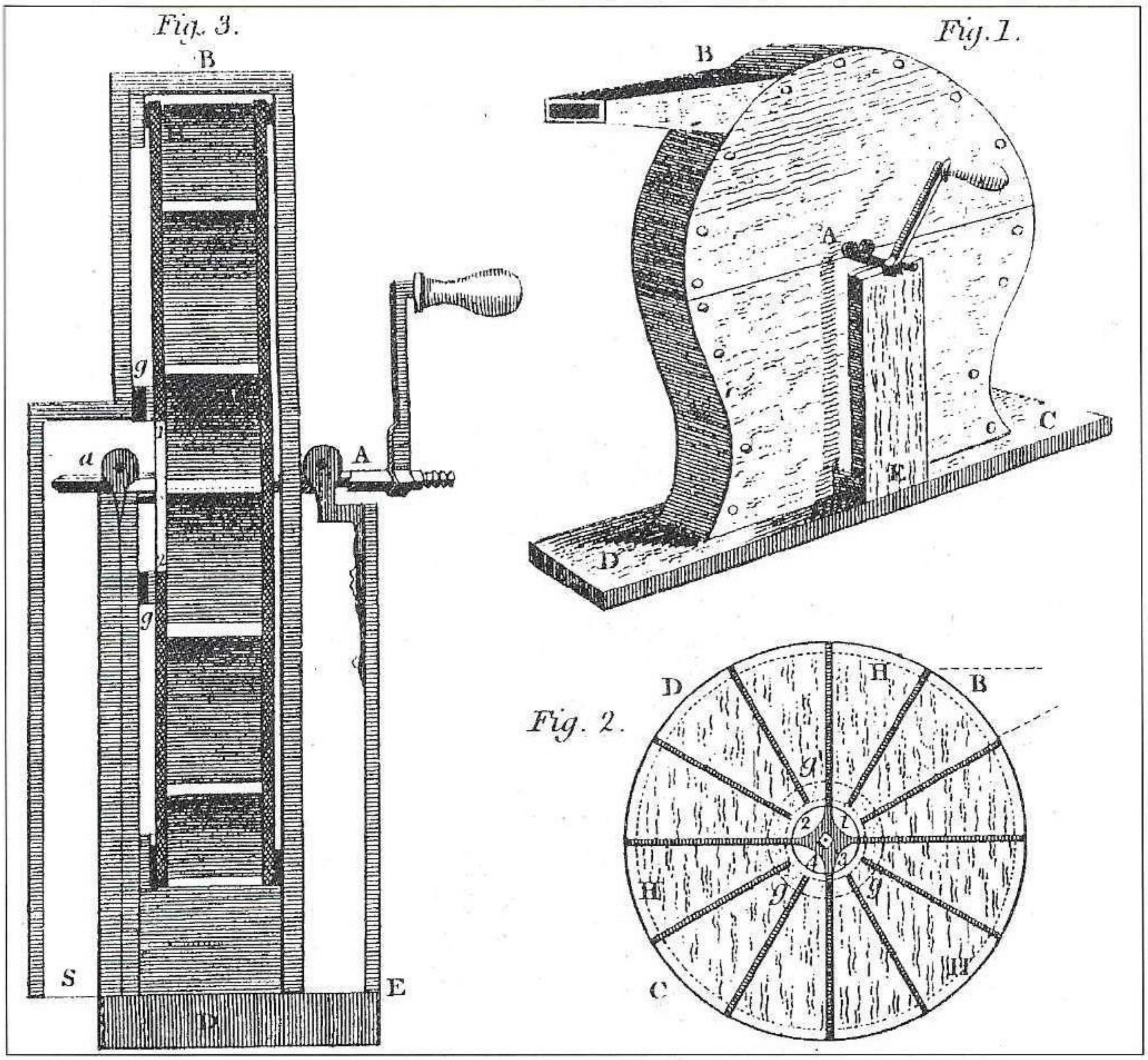
The CIBSE Heritage Group Archive, compiled over some forty-five years, holds a modest, but important, collection of textbooks and information on Fans and Fan Engineering. The earliest use of fans and ventilation equipment seems to have been in the mines of Saxony (Germany) in the mid-16th century. This was described by Georgius Agricola in his book De Re Metallica (1556), and illustrated by a series of intricate woodcut illustrations (see Section-2 of this ebook). These early fans were wind-powered or operated by hand. In addition, use was made of large bellows worked by hand and levers and even by horses.



A—Smaller part of shaft. B—Square conduit. C—Bellows. D—Larger part of shaft.

A—Machine first described. B—This workman, treading with his feet, is compressing the bellows. C—Bellows without nozzles. D—Hole by which heavy vapours or blasts are blown out. E—Conduits. F—Tunnel. G—Second machine described. H—Wooden wheel. I—Its steps. K—Bars. L—Hole in same wheel. M—Pole. N—Third machine described. O—Upright axle. P—Its toothed drum. Q—Horizontal axle. R—Its drum which is made of rundles.

Of particular historical interest is the Fanning Wheel of Dr Desaguliers used in the House of Commons in 1735.



46. Fanning Wheel of Dr Desaguliers, 1735. Used in the House of Commons, where the term ventilator was used to describe the man employed to turn the crank handle which operated the fan. Philosophical Transactions (abridged), VIII, 1735, plate II.

The date of the introduction of curved fan blades is uncertain, but is probably about 1840. However, Walker preferred the screw form of fan impeller.

Ventilation

Real ventilation is so uncommon that . . . the architect usually thinks the object has been attained if some of the windows can be opened.

Ventilating and Warming, Prof. Ernest Jacob, 1894.

Fig. 1.

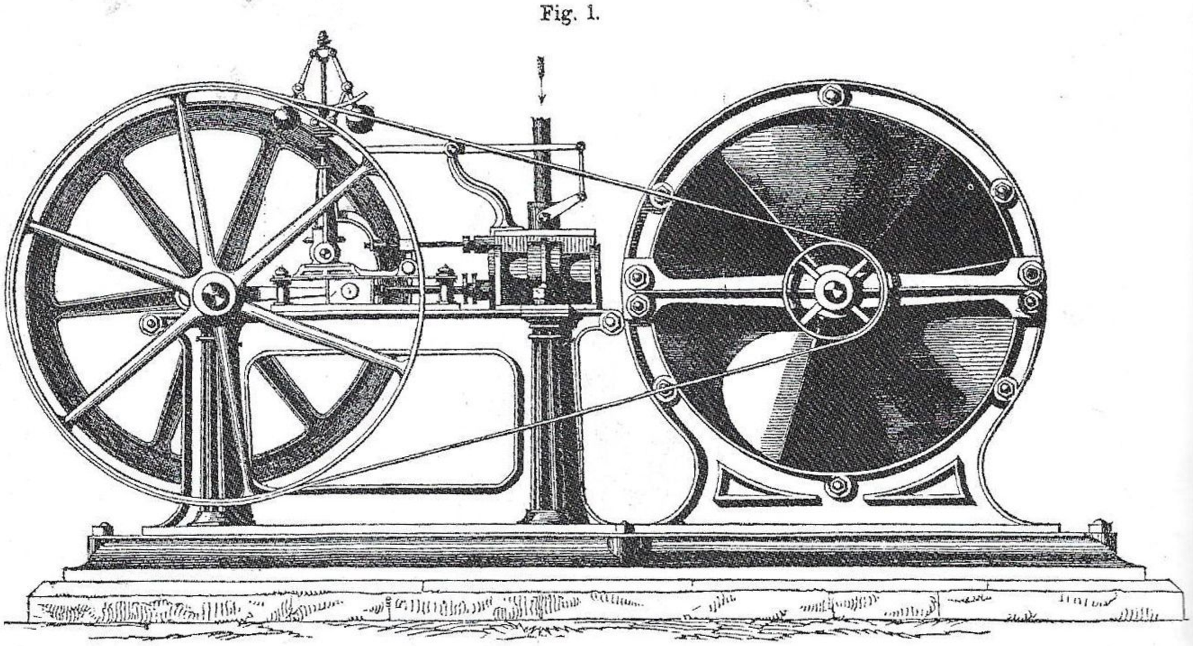
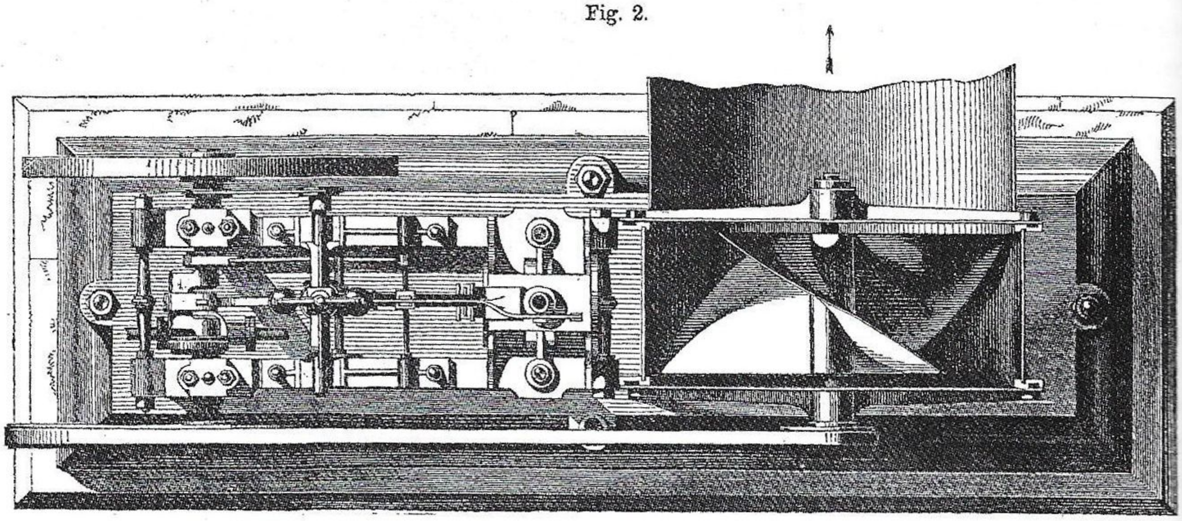


Fig. 2.



45. *Early Screw Fan (strap-driven from a steam engine). The quantity of air delivered in one minute by a screw 4 feet diameter, performing 500 revolutions per minute, was tested by the writer, by a delicate anemometer, and found to exceed 5000 cubic feet. Useful Hints on Ventilation, W Walker, 1850, figs. 1 & 2.*

The History of Fan development is given in Part-3 and Part-4 of this ebook. This includes the backward-curved fan of Buckle (1847), the first true multiblade fan by the American Bennett Hotchkiss (1863), the similar form of multiblade fan by Charles Barlow of London (1878), leading to the successful *Sirocco* fan of Samuel Davidson (1898). Meanwhile in France, fan designs included that of Fournier & Cornu (1896). These latter designs are all centrifugal fans and are the main story featured in this ebook, though mention is made also of axial and cross-flow types.

The 19th century also saw the establishment of major fan manufacturers. Benjamin Franklin Sturtevant (a shoemaker) of Boston, Massachusetts made a simple fan in about 1850, some three hundred years after the use of fans by the Saxony miners. By 1872 the Sturtevant Blower Company was in full operation. By 1884, the Buffalo Forge Company of Buffalo, NY, was making ventilating equipment and by 1896 was a major manufacturer of centrifugal fans (and in 1901 gave Willis Carrier *The Father of Air Conditioning* his first job). Other important fan-makers around the turn of the century included the American Blower Company of Detroit, Michigan, Matthews & Yates of Swinton in Manchester, Sutcliffe also of Manchester and Davidson of Belfast.

As the design and manufacture of fans progressed, the understanding of the basic fan laws became fairly widely known, i.e. how fans perform in a system and the effect of speed changes on air volume delivery, pressure development and power requirements. This is explained in most ventilation and air conditioning textbooks, but the design of fan impellers is almost totally ignored.*

However, this was not the case in Germany where Dr.Ing Bruno Eck published details of fan research and the mathematical treatment of fan design. Copies of his *Preface to the Second German Edition* (1952) and of his *Preface to the First English Edition* (1973) follow. His book *Fans* was translated and edited by Dr Rams Azad & Dr David R Scott. A copy of their *Foreword* follows, noting that in this field "Great Britain is a comparatively backward industry" and that Germany and Russia have developed important performance research techniques.

FANS

DESIGN AND OPERATION OF CENTRIFUGAL, AXIAL-FLOW AND CROSS-FLOW FANS

BY

Dr.-Ing. BRUNO ECK

First English Edition

Translated and Edited by

Dr. RAM S. AZAD

and

Dr. DAVID R. SCOTT

PERGAMON PRESS

OXFORD NEW YORK TORONTO

SYDNEY BRAUNSCHWEIG

*In 1954, this subject was covered by London's National College for Heating, Ventilating & Fan Engineering which ran a specialist Diploma in Fan Engineering (though in 1955 this became an optional main subject choice in the standard H&V Diploma.)

AUTHOR'S PREFACE
TO THE FIRST ENGLISH EDITION

It is a pleasant task to write a preface to the English translation of my book on fans, particularly as this field has taken a new turn over the last decade. Before this time the radial-flow fan was one of the least efficient fluid-moving machines. Nowadays the reverse is true, and the change has been brought about by scientific research. In the past aerofoil theory had been successful when applied to axial-flow fans but not to radial-flow fans. The situation has changed owing to the development of boundary-layer theory and its application to radial-flow fans.

At the same time as fresh progress in the design of the radial-flow fans was taking place, the long-forgotten cross-flow fan was newly discovered. Of especial importance in the new development of this particular fan was the decisive effect of a free whirl unknown up to that time.

Thus theoretical advances have opened up very interesting possibilities for the development of new fans, and both engineers and research men should devote their full attention to this branch of turbo-machines.

Bruno Eck

FOREWORD TO THE FIRST ENGLISH EDITION

DR. BRUNO ECK has said in the preface to the second edition of his well-known textbook on fan engineering that "The fan has been the stepchild of technology". By this he means that firms have been left to develop their own designs without assistance from research institutions and the Government and, in consequence, little real information has been published.

In England the situation is such that no authoritative textbook on fan engineering exists.

The editors recognised this and, as a result, the present translation of Dr. Eck's book into English has just been completed. The expert advisers of the publishers, after a survey of the international literature, unanimously endorsed our selection of Eck's book.

The book gives the fundamental fluid dynamics of turbo-machinery and applies this to fans. It continues with experimental investigations of fans and includes a comprehensive account of all work done in this field. Both theoretical and empirical results are then applied to problems of design. All applications of fans are dealt with in considerable detail. Some sixty pages of the present edition are devoted to the important subject of noise generation and its control in fans.

The book reveals many gaps in existing knowledge and presents a challenge to both engineers and research workers in the field.

It is agreed by leading experts that the days of "cut-and-try" in fan design are over and that fundamental investigations involving the development of advanced measurement techniques are essential.

Our own survey of the literature shows that two institutes, one in Germany and one in Russia, have developed such techniques, which are adapted to the direct measurement of fluid flow between the blades of the rotating impeller.

Thus all components of the velocity triangle may be measured directly without relying on dubious assumptions.

The validity of existing theories of fan design may now be tested experimentally for the first time.

It appears that no work of a comparable nature is being done in this country. From the point of view of developing what in Great Britain is a comparatively backward industry, it seems to us essential that effort and finance be devoted to this field. It is noteworthy that, on the Continent, where fan technology is considerably in advance of our own, many universities have a chair of Turbo-machinery.

The book should be of interest to students, teachers, practising engineers, designers, and research workers, and indeed anyone concerned with the design or application of fans.

CHAPTER IX

THE MAIN TYPES OF CENTRIFUGAL FAN

50. HIGH-PERFORMANCE FANS

The characteristic feature of fan engineering up to the Second World War was the development of the axial-flow fan. Efficiencies of over 80% could be obtained with this type of fan. In one important application, e.g. for the induced draught, the centrifugal fan was completely replaced by the axial-flow fan. Andritzky[(1)] in 1943 reviewed the types of fan employed as induced-draught fans. This report was confined almost entirely to axial-flow fans with efficiencies of around 80%. For the ventilation of mines the radial-flow fan had long been in use. From the experience obtained with centrifugal pumps, particularly with the earlier developments of Rateau, a newer type of impeller with a small diameter ratio appeared and with this an efficiency of about 80% was achieved. The dimensions, and therefore the price of centrifugal fans, was prohibitive in comparison with the new axial-flow fans, and in consequence of this the former were unacceptable. Figure 93

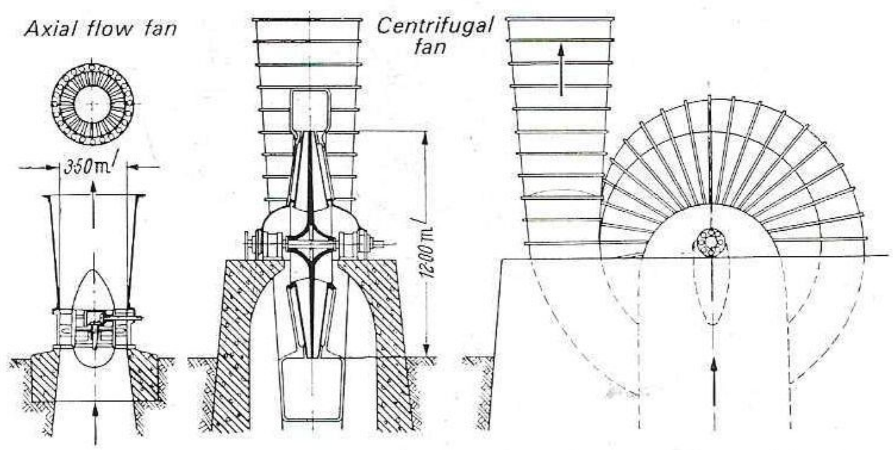


FIG. 93. An axial mine fan and a radial mine fan of an old type compared in size for identical duties.

shows an axial-flow fan[(2)] and a centrifugal fan of the older type being employed as mine ventilators. The unimaginative engineer might well conclude that centrifugal fans were obsolete.

Why were axial-flow fans developed and why were they better than centrifugal fans? It is not possible to give a definite answer to this, but it is related to the success of the aerofoil

[1] Andritzky, M., Axial-flow fans for induced draught plant, *Braunkohle*, 1943, p. 497.

[2] Linsel (Mine ventilation, *VDI-Z*, 1953, p. 429) assessed the best efficiency of German mine axial-flow fans at 83%.

theory immediately after the First World War. This success was due in part to the assumption of a frictionless fluid, and aerofoil theory achieved a remarkable correlation with actual results. In 1922 Bauersfeld(3) already showed a way in which aerofoil theory could be applied to turbines. In this case also the theory was applied with great success. The same method was applied to centrifugal fans. Kucharski(4) had already opened this line of inquiry. Subsequently considerable progress was made in solving purely mathematical problems. The theory of frictionless flow is, unlike the theories of aerofoil and axial-flow machines, divorced from reality. Although the user did not benefit directly from the theoretical analysis, it was instructive. At that time research was confined almost entirely to the types of impeller commonly used in centrifugal pumps and compressors, i.e. impellers with narrow rims and small diameter ratios. The construction of fans received little attention. The centrifugal fan, therefore, was neglected. But a few individual engineers and companies were working on the problems of centrifugal fans, and thus slowly brought about a change.

From the point of view of purely scientific research, fundamental problems arose owing to the difficulties of the mathematical analysis of the boundary layer. This necessitated an additional term to the existing differential equation describing the boundary layer.

About 1951 and 1952 a significant event occurred in the development of centrifugal fans, i.e. the appearance of new high-performance fans (Fig. 96). The way in which the improvements were achieved will be understood when we consider the impellers. At one time investigations were carried out to find out how the design of centrifugal pumps could be applied to the design of centrifugal fans. Figure 94 illustrates a typical impeller which

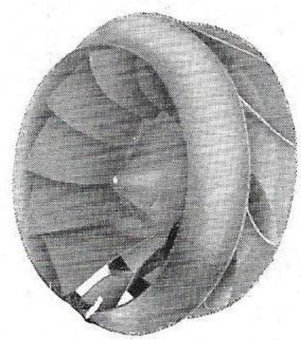


FIG. 94. Impeller with blades brought deep into the suction chamber.

resulted from this type of investigation. It is interesting to note that the blades go deep into the intake chamber, a procedure usually adopted in pumps to offset the effects of cavitation. The efficiency of this design is in the order of 80%, the speed coefficient σ is in the region of 0·4. On the other hand, the designer has the opportunity of taking the advantage of aerofoil blades. The Babcock–Stork blower, for example, was designed with aerofoil blades (Fig. 95). An improvement in the efficiency curve was obtained with a speed coefficient σ of approximately 0·43. A completely new method was then used in the design of high performance fa ns (Fig. 96). In contrast to previous designs, *the discharge width was increased*

[3] Bauersfeld, The basis for calculation of high speed impellers, *Z VDI*, 1922, p. 41.

[4] Kucharski, W., *Strömungen einer reibungsfreien Flüssigkeit bei Rotation fester Körper*, Munich, Oldenbourg, 1918.

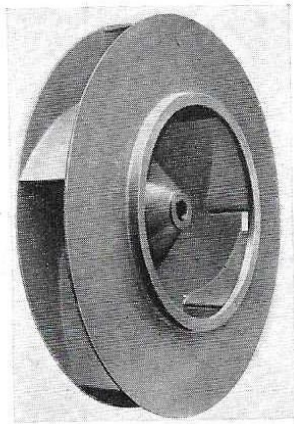


FIG. 95. Impeller with aerofoil blades. Babcock Stork.

FIG. 96. High-performance impeller.

considerably; it was, in fact, almost doubled. The blades had a pronounced backward curvature and the entry was well rounded off. There was no aerofoil profiling or projection protruding into the suction chamber. With this type of impeller a speed coefficient σ of 0·66 was obtained, and it opened up new prospects for the centrifugal fan. For the very first time the efficiency could be raised to nearly 90%. Figure 97 shows the first design of a large mine fan, which at the acceptance trial gave an efficiency of 90%. This impressive result raised the centrifugal fan from the position of being the worst machine to that of the best. A speed coefficient $\sigma = 0{\cdot}66$ gave the designer, *for the first time, the opportunity of obtaining smaller dimensions.* Impellers with the highest efficiencies and smallest dimensions were obtained in the new fields of application. Both features are of vital importance to fans in all applications.

A partial example showing how the mathematical approach to centrifugal fan impeller design has evolved (from Eck)

39. OPTIMUM INLET DIAMETER AND BLADE ANGLE AT ENTRY[1]

In considering the shape of an impeller there are still questions to be answered, e.g. what are the deciding factors to be considered when selecting the diameter ratio d_1/d_2? It is to be expected that the impeller losses are dependent upon the ratio d_1/d_2 as previous calculations have already shown.

Because the maximum velocity—which generally appears in the impeller—is the entry velocity w_1, we must pay due attention to its magnitude.

The simplest assumption to be made is that the smallest possible entry velocity w_1 is required. The losses in the impeller may be expressed as $\Delta p_{\text{imp}} = c\,(\varrho/2)\,w_2'$. The problem is to find a minimum value for w_1.

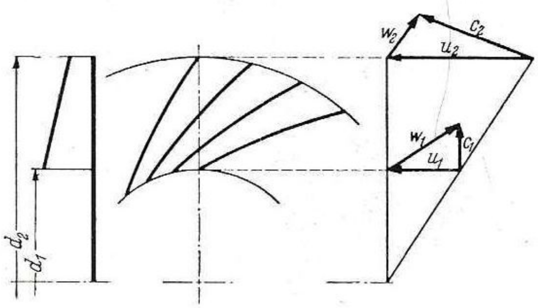


FIG. 67. Impeller with velocity triangles.

From Fig. 67 we get $w_1^2 = u_1^2 + c_{1m}^2$. c_{1m} is obtained from the output V per second, taking into consideration the narrowing of the blades, where

$$c_{1m} = \frac{V}{\pi d_1\, b_1\, [(t-\sigma)/t]} = \frac{c_{0m}}{(t-\sigma)/t}$$

(c_{0m} is immediately before the blading); b_1 is derived from the previous condition of acceleration at the entrance and possible reduction in the area because of the impeller hub,

$$b_1 = \frac{1}{4}\,\frac{d_1}{\xi}\,[1 - \bar{\nu}^2].$$

Substituting for $u_1 = \omega\,(d_1/2)$,

$$w_1^2 = c_{1m}^2 + u_1^2 = \frac{16\xi^2\, V^2}{\pi^2 d_1^4\, [(t-\sigma)/t]^2\, [1-\bar{\nu}^2]^2} + \omega^2\,\frac{d_1^2}{4}.$$

For a given volume V and angular velocity ω, i.e. the rotational speed n, with a fixed value for d_1, a minimum velocity w_1^2 will be obtained. Figure 68 shows how the velocity w_1 varies as d_1 changes from its optimum value.

[1] The following calculation, which first appeared in the first edition, 1937, was completed later in B. Eck, New calculation method for axial and centrifugal fans, *Schweiz. Bauztg.*, 1939, No. 4.

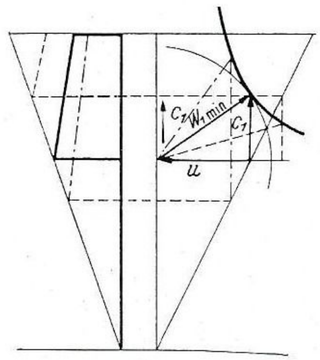


FIG. 68. Variation of relative entry velocity with the inlet diameter, where V = constant, n = constant. The diagram shows the minimum relative velocity.

From the inlet velocity triangle,

$$w_1 = \frac{u_1}{\cos\beta_1} \quad \text{and} \quad w_1^2 = \frac{u_1^2}{\cos^2\beta_1}.$$

Substituting for

$$u_1 = \frac{\pi d_1 n}{60},$$

we get

$$w_1^2 = \frac{\pi^2 d_1^2 n^2}{60^2 \cos^2\beta}.$$

To establish d_1 we shall calculate the volume of flow according to

$$V = c_{0m}\,\pi d_1\, b_1 = c_{1m}\,\frac{t-\sigma}{t}\,\frac{\pi}{4}\,d_1^2\,[1-\bar{\nu}^2]\,\frac{1}{\xi},$$

from which

$$d_1^2 = \frac{4V\xi\,[t/(t-\sigma)]}{[1-\bar{\nu}^2]\,c_{1m}\pi}.$$

Inserting this into the equation for w_1^2,

$$w_1^2 = \frac{\pi^2 n^2}{60^2\cos^2\beta_1}\,\frac{4V\xi\,[t/(t-\sigma)]}{[1-\bar{\nu}^2]\,c_{1m}\pi},$$

and multiplying both sides by w_1,

$$w_1^3 = \frac{4\pi n^2\, V\xi\,[t/(t-\sigma)]}{60^2[1-\bar{\nu}^2]\cos^2\beta_1\,(c_{1m}/w_1)} = \frac{4\pi n^2\, V\xi\,[t/(t-\sigma)]}{60^2\,[1-\bar{\nu}^2]\cos^2\beta_1\,\sin\beta_1}.$$

The minimum value of w_1 is a function of the angle β_1 and is obtained by equating $(\mathrm{d}w_1^3/\mathrm{d}\beta_1) = 0$. Calculation yields the simple result

$$\tan\beta_1 = \frac{1}{\sqrt{2}}. \tag{107}$$

To obtain the corresponding diameter of the entry d_1 we take the above equation for d_1^2 into consideration and substitute it into

$$c_{1m} = u_1 \tan\beta_1 = \frac{\pi\, d_1\, n}{60} \tan\beta_1,$$

from which

$$d_1 = \sqrt[3]{\frac{240\xi V\,[t/(t-\sigma)]}{[1-\bar{\nu}^2]\,\pi^2\, n \tan\beta_1}}. \tag{108}$$

For the case where

$$\xi \approx 1; \quad \frac{t}{t-\sigma} \approx 1; \quad \bar{\nu}^2 \approx 0,$$

$$d_1 = \sqrt[3]{\frac{240 V}{\pi^2\, n \tan\beta_1}}$$

or

$$V = \frac{d_1^3\, \pi^2\, n \tan\beta_1}{240}. \tag{109}$$

To obtain a dimensionless form we shall consider

$$V = \varphi\, u_2 \frac{\pi\, d_2^2}{4} \quad \text{and} \quad u_2 = d_2 \frac{\pi\, n}{60}.$$

Thus we derive the important ratio d_1/d_2:

$$\frac{d_1}{d_2} = \sqrt[3]{\frac{\xi\varphi\,[t/(t-\sigma)]}{[1-\bar{\nu}^2]\tan\beta_1}}, \tag{110}$$

from which the simple formula for the volume coefficient φ is obtained, i.e.

$$\varphi = \tan\beta_1 \left(\frac{d_1}{d_2}\right)^3 \frac{1-\bar{\nu}^2}{\xi}\, \frac{t-\sigma}{t}.$$

This can be simplified because in most cases

$$\bar{\nu}^2 \approx 0, \quad \xi \approx 1 \quad \text{and} \quad \frac{t-\sigma}{t} \approx 1,$$

so that

$$\varphi = \tan\beta_1 \left(\frac{d_1}{d_2}\right)^3. \tag{111}$$

Figure 69 graphically illustrates this equation.

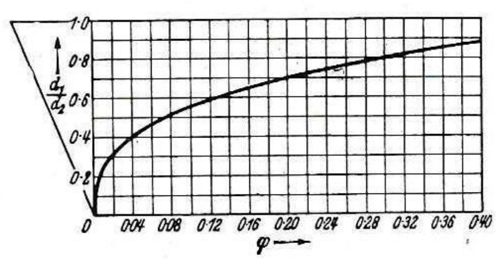


FIG. 69. Optimum value of d_1/d_2 as a function of φ.