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(12) # EUROPEAN PATENT APPLICATION

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(54) **A turbine blade.**

(57) A turbine blade (10) having such a profile that (A) a straight line (H) is drawn such that (a) said straight line passes a point of intersection (J) between an extension of a first straight line (F) which, together with a second straight line in parallel with the axis of a circular blade array, defines an inlet angle (α_1) and an extension of a third straight line which, together with a fourth straight line in parallel with said axis, defines an outlet angle (α_2), (b) said straight line (H) is in parallel with said axis, and (c) said straight line (H) is spaced apart from the outlet end of the blade (10) by a distance greater than one half of the chord length (C) thereof; and (B) that at the point of intersection (P) between said straight line (H) and the center line (A) of the flow passage between adjacent blades, said point being the flow direction changing point, the smallest width (S_P) of the flow passage is less than about 0.4 times the width (t) of said flow passage at the inlet thereof.

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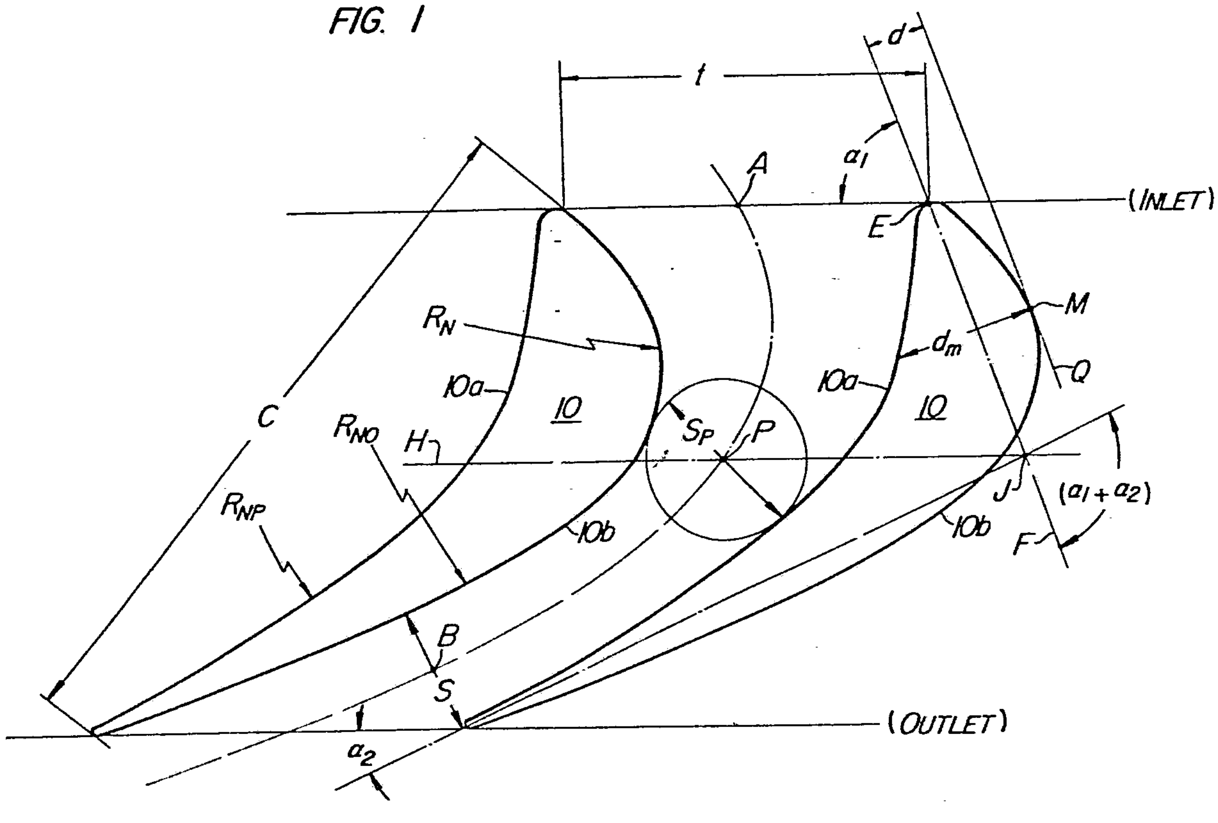
FIG. 1



t
d
a_1
A
(INLET)
E
M
C
R_N
d_m
Q
10a
10
10a
10
R_{NO}
H
S_P
P
R_{NP}
10b
J
(a_1+a_2)
10b
F
B
S
a_2
(OUTLET)

A TURBINE BLADE

The present invention relates to generally a velocity enhancing blade array of axial flow fluid machines and more particularly a turbine blade.

Various blade profiles which constitute the blade arrays of axial flow fluid machines, such as turbines have been designed and demonstrated. For instance, a turbine blade profile consists of a plurality of successively merging circular arcs whose radii of curvature are gradually decreased from the leading edge to the trailing edge. Blade profiles are in general designed to obtain a desired inlet angle, a desired outlet angle and a desired blade width or chord length, but hydrodynamical conditions in the flow passage between the adjacent blades are not taken into consideration. In addition, understanding of the performance of the blade profiles which can be used in practice is not sufficient. As a result, it has been very difficult to obtain a turbine blade profile which ensures high performance of an axial flow fluid machine. More specifically, the boundary layers are formed over the blade surfaces due to the viscosity of the fluid and flow past the outlet of the flow passage, resulting in the lack of velocity of the fluid at the downstream of the outlet. The degree of the lack of the velocity of the fluid at the downstream of the outlet determin the performance of the blade profile. The most important factor which must be taken into consideration in design of turbine blade profiles is the thickness of the boundary layer at the outlet of the flow passage between the adjacent blades. In general, the thinner the boundary layer at the outlet, the higher the performance becomes. It has been clarified that the development of the thickness of the boundary layer is closely correlated with the variations in velocity of the fluid passing through the flow passage. However so far the variations in velocity have not been taken into consideration in the design of a flow passage between the blades. As a result, no attempt has been made to suppress the formation of the boundary layer so that the separation of the boundary layer results, causing very serious adverse effects on the performance. Thus it has been difficult to obtain the turbine blade profiles which ensure the high performance.

One of the objects of the present invention is therefore to provide a turbine blade which can stabilize the boundary layers thereon, thus ensuring high performance.

Another object of the present invention is to provide a high-performance turbine blade which enables the fluid to flow through the flow passage defined between the adjacent turbine blades in such a way that the acceleration of the fluid is almost completed before the fluid reaches the flow direction changing point in the flow passage, whereby the boundary layers can be stabilized and high performance can be ensured.

To the above and other ends, briefly stated, the present invention provides a turbine blade having such a blade profile that (A) a straight line is drawn in such a way that (a) said straight line passes a point of intersection between an extension of a first straight line which, together with a second straight line in parallel with the axis of a circular turbine blade array, defines an inlet angle and an extension of a third straight line which, together with a fourth straight line in parallel with said axis, defines an outlet angle, (b) said straight line is in parallel with said axis and (c) said straight line is spaced apart from the outlet or discharge end of the turbine blade by a distance greater than one half of the chord length thereof; and (B) that at the point of intersection between said straight line thus drawn and the center line of the flow passage defined between the adjacent turbine blades, said point being the flow direction changing point, the smallest width of the flow passage is less than about 0.4 times the width of said flow passage at the inlet thereof, whereby the acceleration of the fluid flowing through the flow passage is almost completed prior to said flow direction changing point and consequently the boundary layers on the blades are stabilized to such a higher degree as unattainable by any prior art turbine blade profile.

The above and other objects, features and effects of the present invention will become more apparent from the following description of a preferred embodiment thereof taken in conjunction with the accompanying drawings, in which:-

Fig. 1 is a diagram of a turbine blade profile in accordance with the present invention;

Fig. 2 shows the development of the flow passage between the adjacent blades shown in Fig. 1;

Fig. 3 shows the pressure distributions on the surfaces of the turbine blade in accordance with the present invention;

Fig. 4 is a view used for the explanation of the behaviors of the boundary layer on the back surface of the turbine blade in accordance with the present invention; and

Fig. 5 shows the relationship between the deflection angle and the blade profile loss coefficient of the turbine blade in accordance with the present invention.

Referring first to Fig. 1, the features of a blade profile in accordance with the present invention will be described. A line H is first drawn which is in parallel with the axis of blades 10 (that is, the direction in which the blades 10 are mounted in a circular array) and which passes the point of intersection J between a first line F inclined with respect to a second line, which is in parallel with the axis of a circular turbine blade array, at an inlet angle $\underline{\alpha}_1$ and a third line inclined with respect to a fourth line in parallel with the above-mentioned axis at an outlet angle $\underline{\alpha}_2$. The position of this line H corresponds to the point at which the fluid flow is deflected in direction within the passage between the back surface 10b of the turbine blade 10 and the front surface 10a of the adjacent blade 10. As shown in Fig. 2, the inlet width of this passage is denoted by $\underline{t}$ while the outlet width, by $\underline{s}$. The passage width S_P is the width at the point P at which the center line A of the flow passage intersects the line H. The distance between the straight line H which passes the flow direction changing point P and the outlet of the blade is so selected as to be greater than one half of the chord length $\underline{C}$ of the blade 10. The portion of the blade profile above the straight line H is referred to as "the upstream portion" while the portion below the straight line H, "the downstream portion". The radius of curvature R_N of the upstream portion of the back surface 10b is smaller than that of the prior art blade profile while the radius of curvature R_{NO} of the downstream portion is greater than that of the prior art blade profile. In addition, the radius of curvature R_{NP} of the downstream portion of the front surface 10a is greater than that of the prior art blade profile.

Fig. 2 shows the development of the flow passage between the adjacent blades along the center line APB shown in Fig. 1. It is seen that the width of the flow passage is drastically reduced at the upstream portion from the inlet to the flow direction changing point P (from A to P in Fig. 1) while the decrease in width is gradual in the downstream portion (from P to B in Fig. 1).

In brief, according to the present invention, the radius of curvature R_N of the upstream portion of the back surface 10b (from the inlet to the straight line H in Fig. 1) is made smaller than that of the prior art blade profile. That is, $R_N/C < 0.15$ in mathematical terms. The radius of curvature R_{NO} of the downstream portion of the back surface 10b (from the straight line H to the outlet in Fig. 1 is expressed by $R_{NO}/C > 5.0$. The radius of curvature R_{NP} of the downstream portion of the front surface 10a is expressed by $R_{NP}/C > 1.3$. These conditions are summarized in TABLE 1 below.

TABLE 1

	upstream portion of back surface	downstream portion of back surface	downstream portion of front surface
radius of curvature	$\frac{R_N}{C} < 0.15$	$\infty > \frac{R_{NO}}{C} > 5.0$	$\infty > \frac{R_{NP}}{C} > 1.3$

TABLE 2 shows the relationship between the passage width S_P at the flow direction changing point P, the width S at the outlet and the width t at the inlet.

TABLE 2

Ratio	S/S_P	S_P/t
	< 1.0 > 0.9	< 0.4

Since the flow passage width at the flow direction changing point P is $S_P/t < 0.4$, the above width is smaller than that of the prior art blade profile at the upstream of the point P. On the other hand, since the flow passage width at the point P is $0.9 < S/S_P < 1.0$, the above width is greater than that of the prior art blade profile at the downstream of the point P. In summary, according to the present invention, as compared with the prior art blades, the curvature of the back surface above the straight line H, which passes through the flow direction changing point P, is made greater while the curvatures of the downstream portions of the front and back surfaces are made smaller or made substantially zero. Opposed to the prior art blade profiles consisting of successive merging circular arcs, according to the present invention, a flow passage profile can be defined in which an optimum acceleration of flow can be ensured. As a result, the acceleration of the fluid flowing through the flow passage between the blades can be substantially completed before the fluid reaches the flow direction changing point P.

Next the thickness of the blade profile in accordance with the present invention will be described with further reference to Fig. 1. The thickness of the upstream portion of the blade is very noticeably different from that of the prior art blade. The distance d between the straight line F passing the tip E of the blade and the point J and the straight line Q which is in parallel with the straight line F and tangential to the back surface 10b is 1.5 to 2.0 times as compared with the prior art blade. The increase in thickness results from the fact that the redius of curvature R_N of the upstream portion of the back surface 10b is reduced so that the upstream portion of the blade is increased in thickness. As a result, the acceleration of the fluid can be substantially completed before the fluid reaches the flow direction changing point P without changing the inlet angle α_1. In addition, the acceleration stabilizes the boundary layers and decreases their thickness. The fluid flow is deflected along the front and back surfaces 10a and 10b, which are concave and convex, respectively, so that satisfactory boundary layers are formed even after passing the flow direction changing point P. As a consequence, a uniform velocity distribution can be attained in the flow at the downstream of the outlet.

In summary, according to the present invention, the thickness d_m of the blade is given by the following dimensionless expression or parameter:

$$d_m/C > 0.23$$

where d_m is the distance from the point M, at which the straight line Q is tangent to the back surface 10b, to the point at which a straight line constructed at the point M at right angles to the straight line Q intersects the outline profile of the front surface 10a of the blade. It will be apparent that, as compared with the prior art blade in which d_m/C is 0.16, the upper portion of the blade is increased in thickness.

The features of the present invention will be more clearly understood from Fig. 3 which shows the flow in the passage between the blades is expressed in terms of the pressure acting on the blade surfaces. The pressure acting on the back surface of the blade has a high pressure drop ΔPs in the upstream portion of the flow passage from the inlet to the point P at which the flow is deflected. Since the pressure drop ΔPs approaches ΔP which is a pressure drop in the overall portion of the flow passage, the stabilized boundary layers can be formed. At the throat (indicated by S in Fig. 1), a very gentle increase in pressure is observed while a sudden pressure rise is observed in the case of the prior art blade. A sudden pressure rise (or the decrease in velocity) facilitates the formation of the boundary layers. That is, the pressure rise determines the conditions of the boundary layers formed and consequently the performance of the blade.

Shown in Fig. 4 are the velocity distribution V, displacement thickness $\underline{\delta}$ and momentum thickness $\underline{\theta}$ on the back surface 10b of the blade. The thicknesses $\underline{\delta}$ and $\underline{\theta}$ are the measures in determining the thickness of the boundary layer and are calculated (according to "TN D-5681", published by NASA, May 1970) based upon the pressure distributions shown in Fig. 3. As described above, according to the present invention, the acceleration is almost completed before the fluid reaches the flow direction changing point P so that both the displacement thickness $\underline{\delta}$ and the momentum thickness $\underline{\theta}$ can be decreased at the outlet of the blade ($l_x/L = 1.0$), whereby a high performance blade profile can be obtained.

From the data shown in Fig. 4, the blade profile loss coefficient $\underline{e}$ is obtained by the following equation.

$$e = \frac{1.74\,\theta}{1 - \delta}$$

where $\underline{e}$ is the blade profile loss coefficient;

$\underline{\delta}$ is the displacement thickness; and

$\underline{\theta}$ is the momentum thickness.

As compared with the prior art blade profile, the blade profile loss coefficient e of the blade profile in accordance with the present invention is reduced by about 30%.

In Fig. 5 is shown the relationship between the blade profile loss coefficient e and the inlet and outlet angles α_1 and α_2. The blade profile loss coefficient e is plotted along the ordinate while the deflection angle $[180° - (\alpha_1 + \alpha_2)]$, along the abscissa. It is seen that when the deflection angle is close to 100°, the blade profile loss coefficient can be made as little as about 0.02 as compared with the prior art blade having a blade profile loss coefficient of higher than 0.025. Thus the present invention provides a blade profile with a minimum loss and a higher degree of performance.

In summary, according to the present invention, the acceleration is almost completed before the flow direction changing point so that the boundary layers can be highly stabilized and consequently the velocity enhancing and high performance blade profile can be provided.

1. A turbine blade characterized by having a blade profile in which drawn is a straight line (H) which passes a point of intersection (J) between an extension of a first straight line (F) which defines an inlet angle (α_1) with a second straight line in parallel with the axis of a circular turbine blade array and an extension of a third straight line which defines an outlet angle (α_2) with a fourth straight line in parallel with said axis of a circular turbine blade array, said straight line (H) being in parallel with said axis of a circular turbine blade array and being spaced apart from the outlet or discharge end of said blade (10) by a distance greater than one half of the chord length (C) of said blade (10); and the smallest width (S_p) of the flow passage between the adjacent blades at the point of intersection (P) between said straight line (H) and the center line (A) of said flow passage, said point (P) being the flow direction changing point, is selected to be less than about 0.4 times as small as the width (t) of the inlet of said flow passage, whereby the acceleration of the fluid flowing through said flow passage is almost accomplished before said flow direction changing point (P) and thereby the boundary layers are stabilized.

2. A turbine blade as set forth in claim 1 further characterized in that said smallest width (S_p) of said flow passage at said flow direction changing point (P) is about 0.9-1.0 times the smallest width at the outlet of said flow passage.

3. A turbine blade as set forth in claim 1 further characterized in that the radius of curvature (R_N) of the portion of the back surface of the blade (10) at the upstream of said flow direction changing point (P) is less than 0.15 times the chord length (C) of said blade (10).

4. A turbine blade as set forth in claim 1 further characterized in that the radius of curvature (R_{NO}) of the portion of the back surface of the blade (10) at the downstream of said flow direction changing point (P) is greater than 5 times the chord length (C) of said blade (10).

5. A turbine blade as set forth in claim 4 further characterized in that the radius of curvature (R_{NP}) of the portion of the front surface of the blade (10) at the downstream of said flow direction changing point (P) is greater than 1.3 times the chord length (C) of the blade (10).

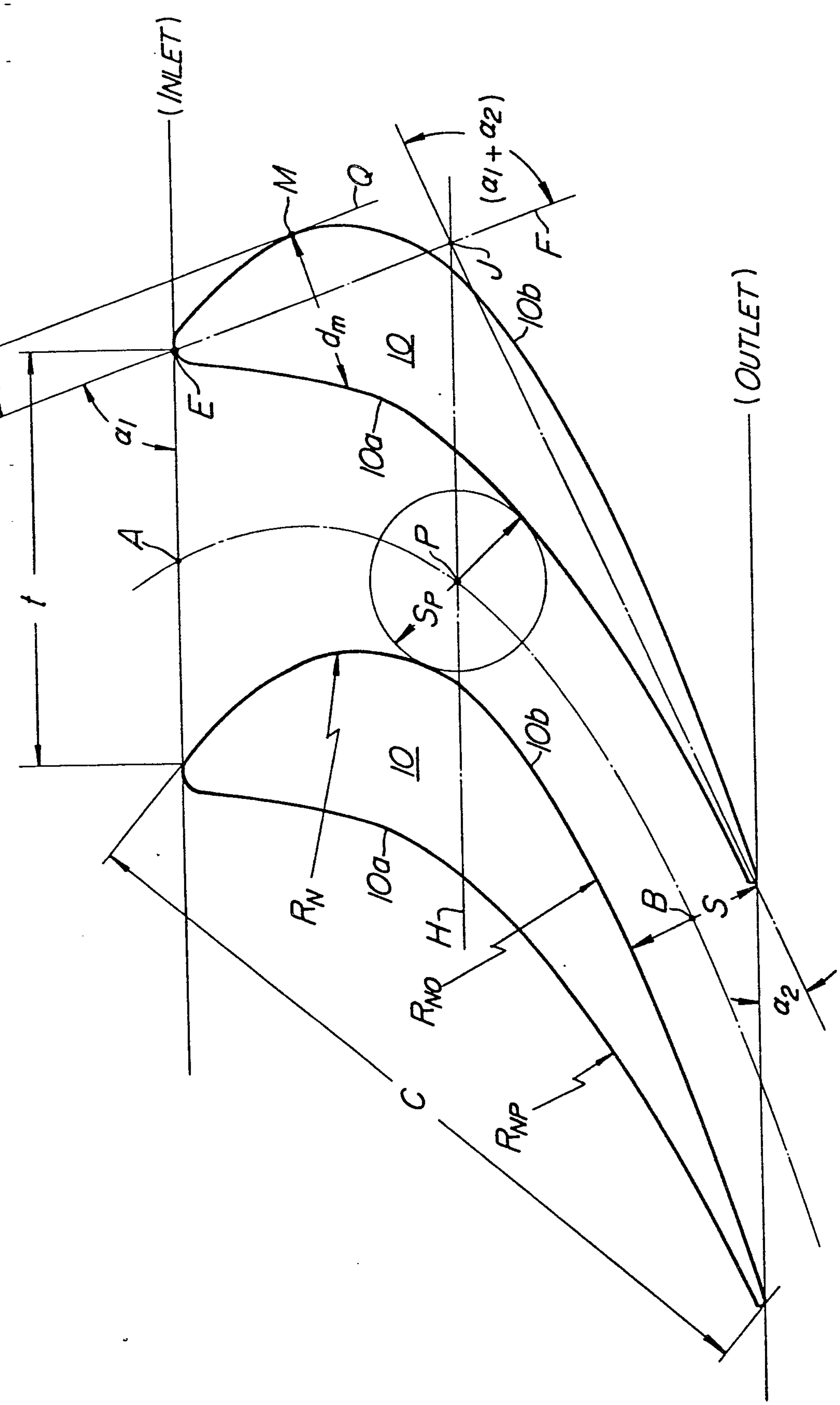

(INLET)
(OUTLET)
M
Q
$(a_1 + a_2)$
F
J
10b
d_m
10
d
E
a_1
10a
A
P
S_P
t
R_N
10a
H
R_{NO}
B
S
a_2
C
R_{NP}

FIG. 1

FIG. 2

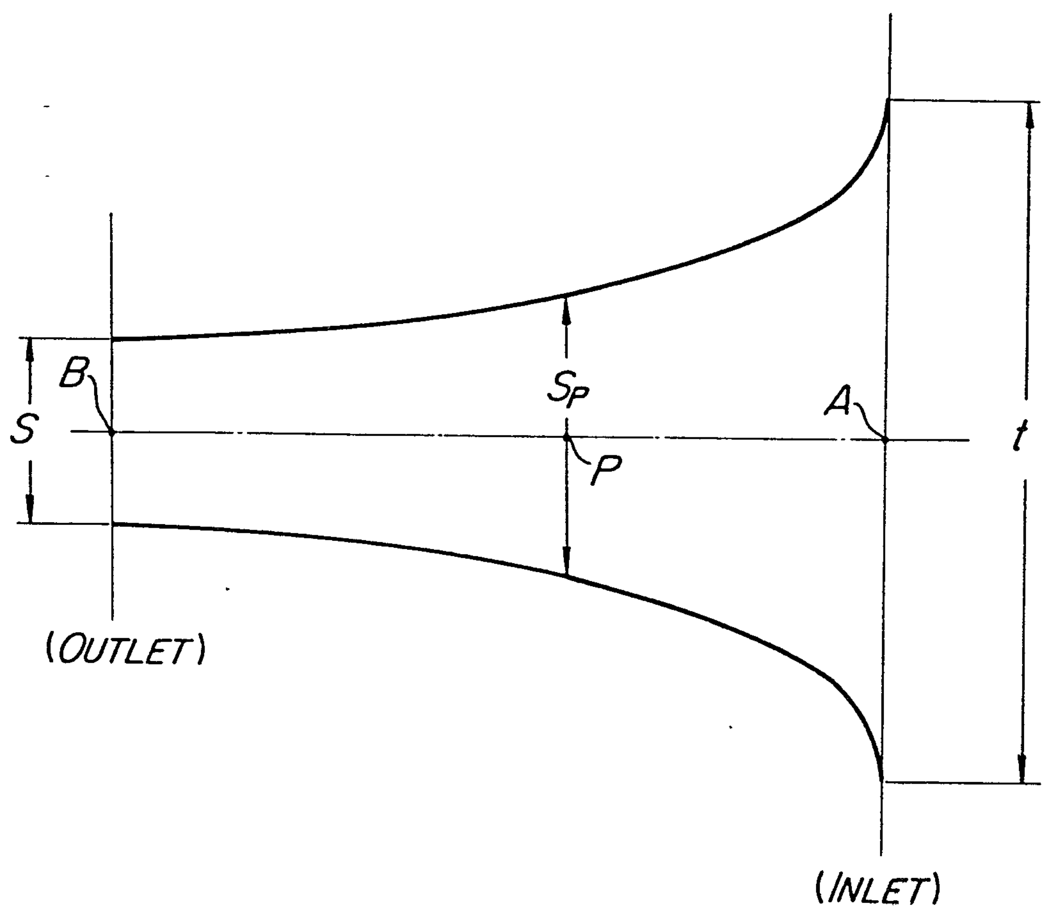


FIG. 3

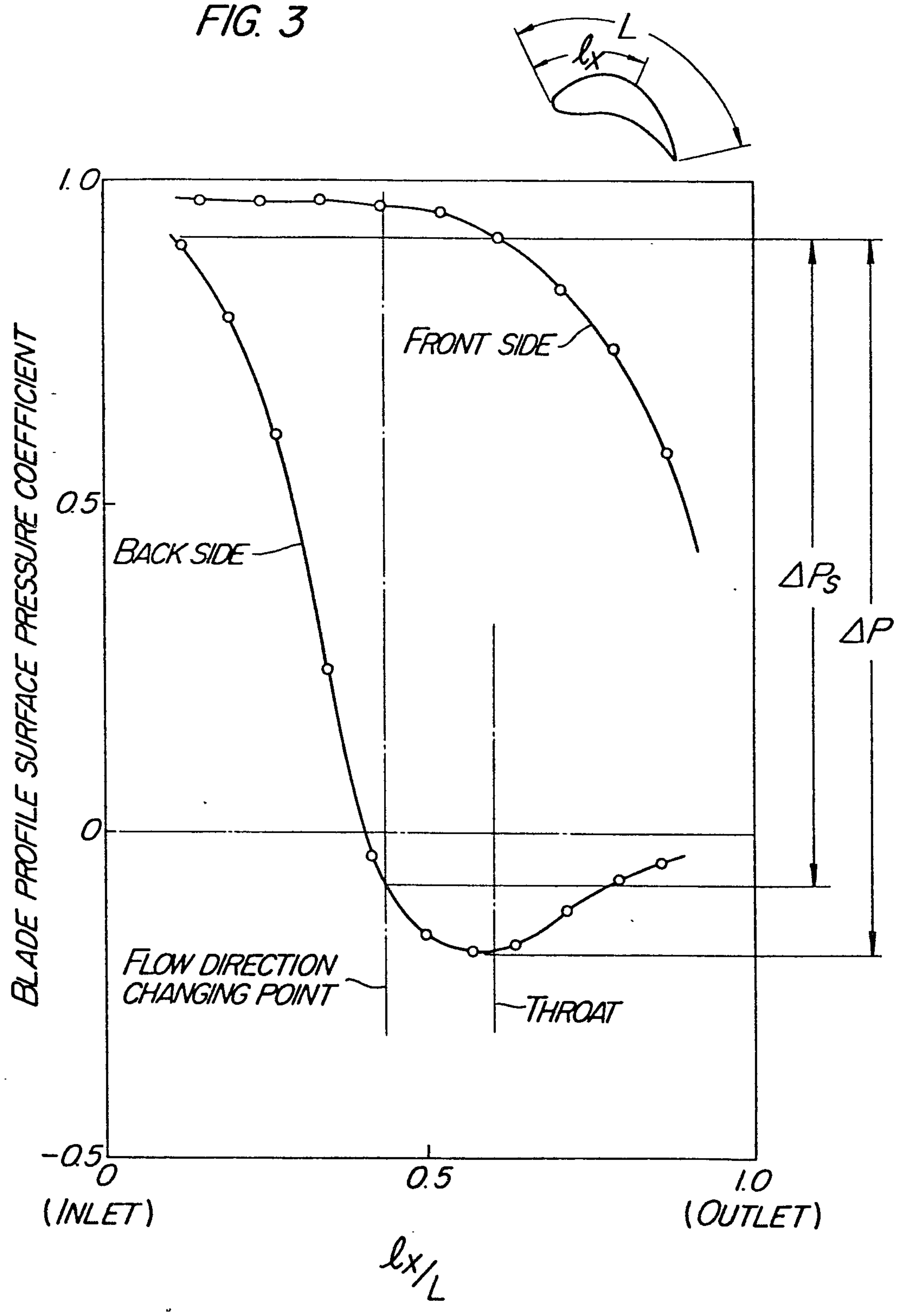

L
ℓ_x
1.0
0.5
0
-0.5
FRONT SIDE
BACK SIDE
FLOW DIRECTION
CHANGING POINT
THROAT
ΔP_S
ΔP
BLADE PROFILE SURFACE PRESSURE COEFFICIENT
0
(INLET)
0.5
1.0
(OUTLET)
$\ell x/L$

FIG. 4

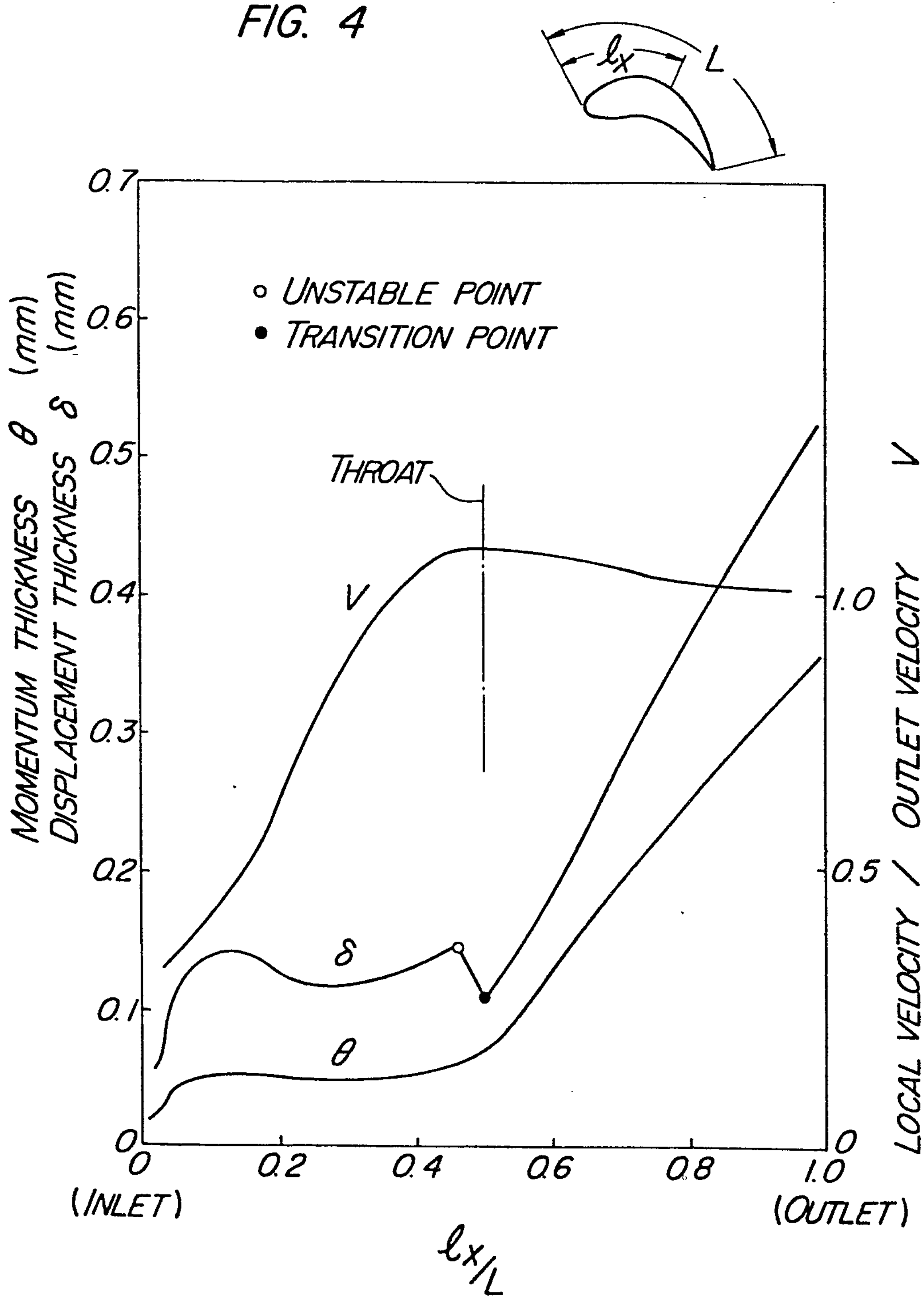
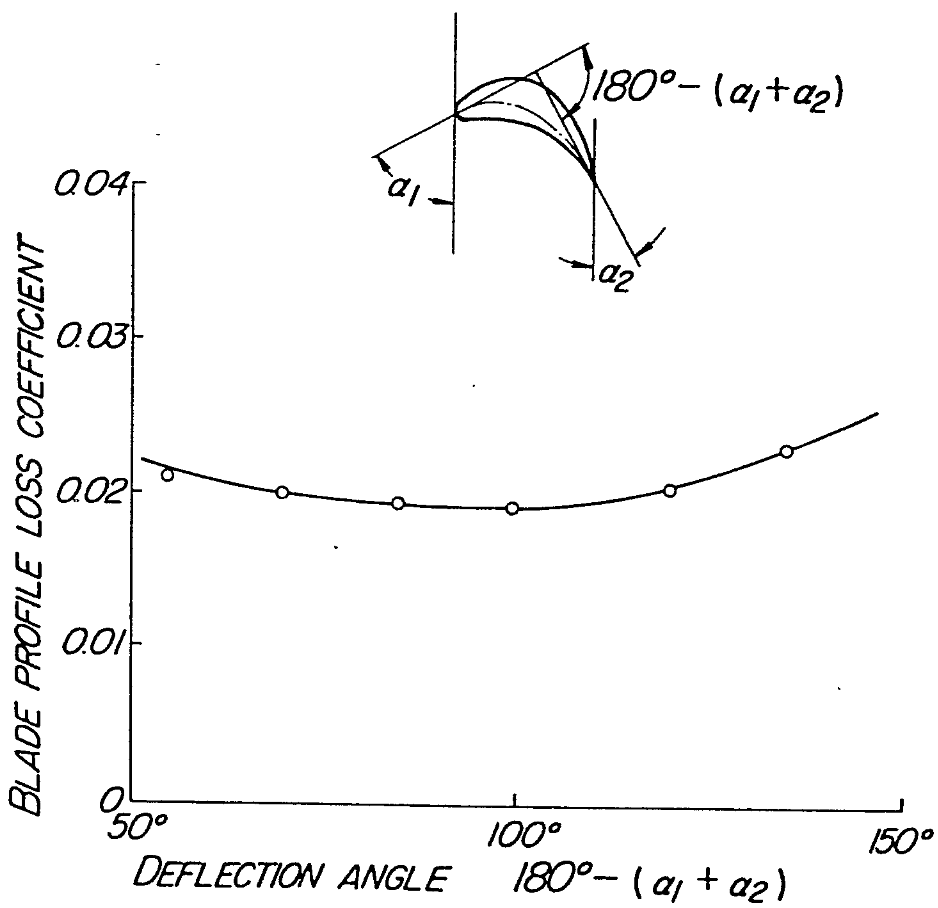

ℓx
L
o UNSTABLE POINT
• TRANSITION POINT
THROAT
V
δ
θ
MOMENTUM THICKNESS θ (mm)
DISPLACEMENT THICKNESS δ (mm)
LOCAL VELOCITY / OUTLET VELOCITY V
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1.0
0.5
0
0.2
0.4
0.6
0.8
1.0
(INLET)
(OUTLET)
ℓx/L

FIG. 5


180° − (a1+a2)
a1
a2
0.04
0.03
0.02
0.01
0
BLADE PROFILE LOSS COEFFICIENT
50°
100°
150°
DEFLECTION ANGLE 180° − (a1 + a2)

0023025

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EUROPEAN SEARCH REPORT

Application number
EP 80 10 4153

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim
	DE - B - 1 272 305 (SIEMENS) * In its entirety * --	1
	US - A - 3 475 108 (SIEMENS) * Column 1, line 67 - column 2, line 67; figure 3 * --	1
	FR - E - 616 250 (BROWN, BOVERI) * Page 2, left-hand column, line 50 - page 3, right-hand column, line 67 * --	1
	GB - A - 550 393 (WIBERG) * Page 2, right-hand column, line 70 - page 3, left-hand column, line 35 * --	1,3-5
A	VDI-ZEITSCHRIFT, vol. 93, no. 27, 21st September 1951, pages 872-873 Düsseldorf, DE. A. FRERICHS: "Über Gestaltung und Systematik neuerer Schaufelprofile für Dampf- und Gasturbinen". --	1-5
A	BROWN BOVERI MITTEILUNGEN, vol. 51, no. 12, December 1964, pages 752-761 Baden, CH. H.E. IMBACH: "Berechnung der kompressiblen, reibungsfreien Unterschallströmung durch ebene Schaufelgitter" -- ./.	1-5

CLASSIFICATION OF THE APPLICATION (Int Cl.[3])

F 01 D 5/14

TECHNICAL FIELDS SEARCHED (Int. Cl.[3])

F 01 D
B 23 P

CATEGORY OF CITED DOCUMENTS

X: particularly relevant
A: technological background
O: non-written disclosure
P: intermediate document
T: theory or principle underlying the invention
E: conflicting application
D: document cited in the application
L: citation for other reasons

&: member of the same patent family, corresponding document

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
The Hague	16-10-1980	BONVIN

EPO Form 1503.1 06.78

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EUROPEAN SEARCH REPORT

0023025

Application number

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-2-

DOCUMENTS CONSIDERED TO BE RELEVANT			CLASSIFICATION OF THE APPLICATION (Int. Cl. 3)
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	
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A	SIEMENS ZEITSCHRIFT, vol. 41, 1967, Beiheft "Dampfturbinen grosser Leistung", pages 113-119 Erlangen, DE. VON OTTO-ADALBERT VON SCHWERDTNER: "Strömungsuntersuchungen an Turbinenschaufeln" --	1-5	
A	SIEMENS ZEITSCHRIFT, vol. 8, August 1959, pages 516-520 Erlangen, DE. VON WALDEMAR ZICKUHR: "Ein vereinfachtes Verfahren zur angenäherten Bestimmung der Dicke der Schaufelprofile von Turbomaschinen" ----		