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Ribaud

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[54] **SUPERSONIC CENTRIFUGAL COMPRESSORS**

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[51] **Int. Cl.**[2] **F04D 21/00;** F04D 29/44

[58] **Field of Search** .... 415/181, 143, 213, 211, 415/219 A, 163; 416/183, 188

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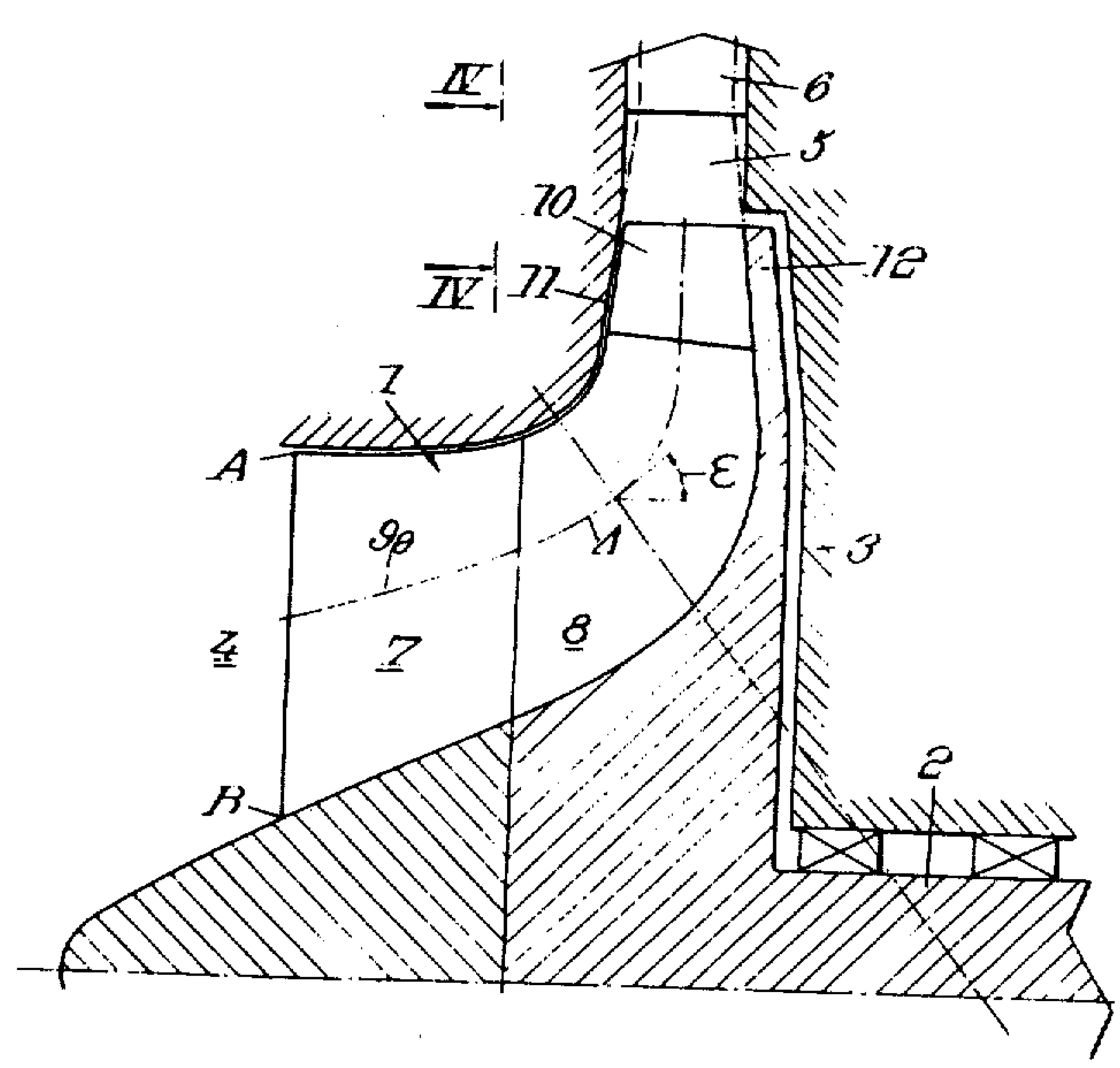
[57] **ABSTRACT**

A supersonic centrifugal compressor comprises a rotor located in a housing having a fluid intake eye. The fluid (air for instance) successively travels through an intake region wherein the rotor has a small number of blades which deflect the fluid tangentially by a small amount only, then through a compression region wherein the rotor has a higher number of blades producing tangential and meridian flow deflection. Last, the fluid flows substantially radially with respect to the rotor into a stationary diffuser.

11 Claims, 4 Drawing Figures



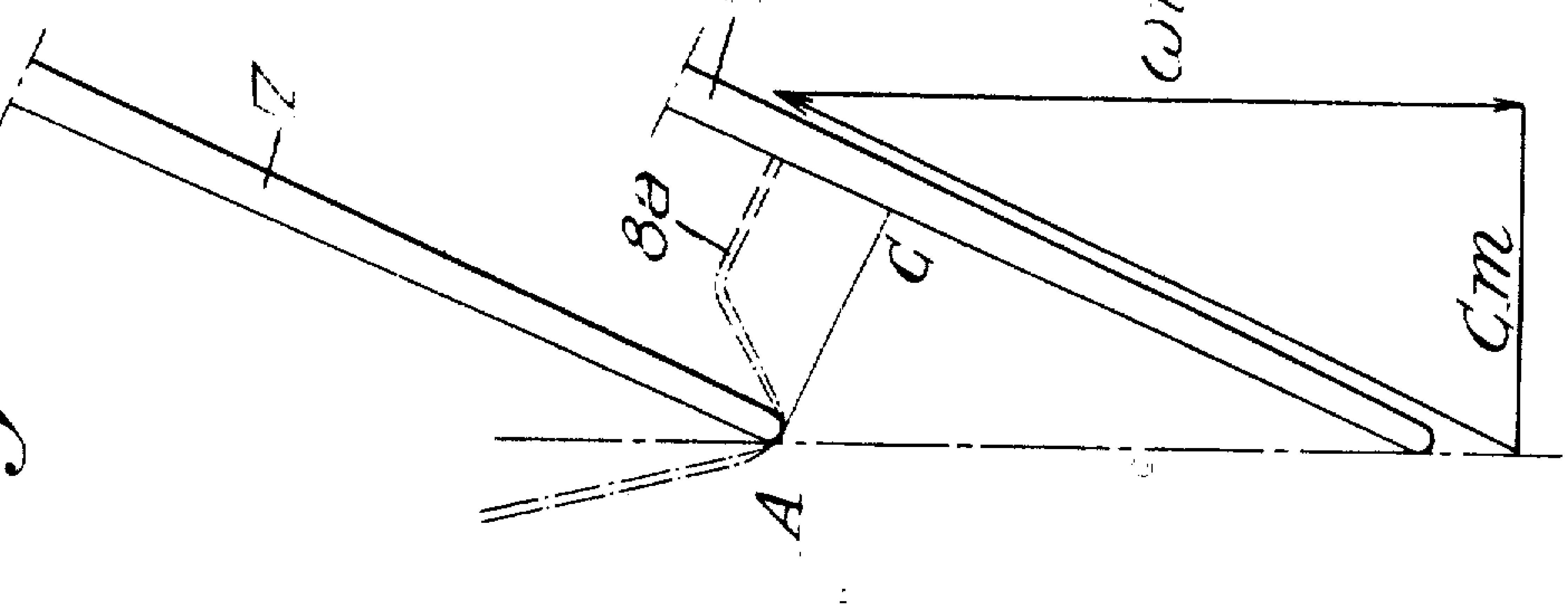


Fig. 3.

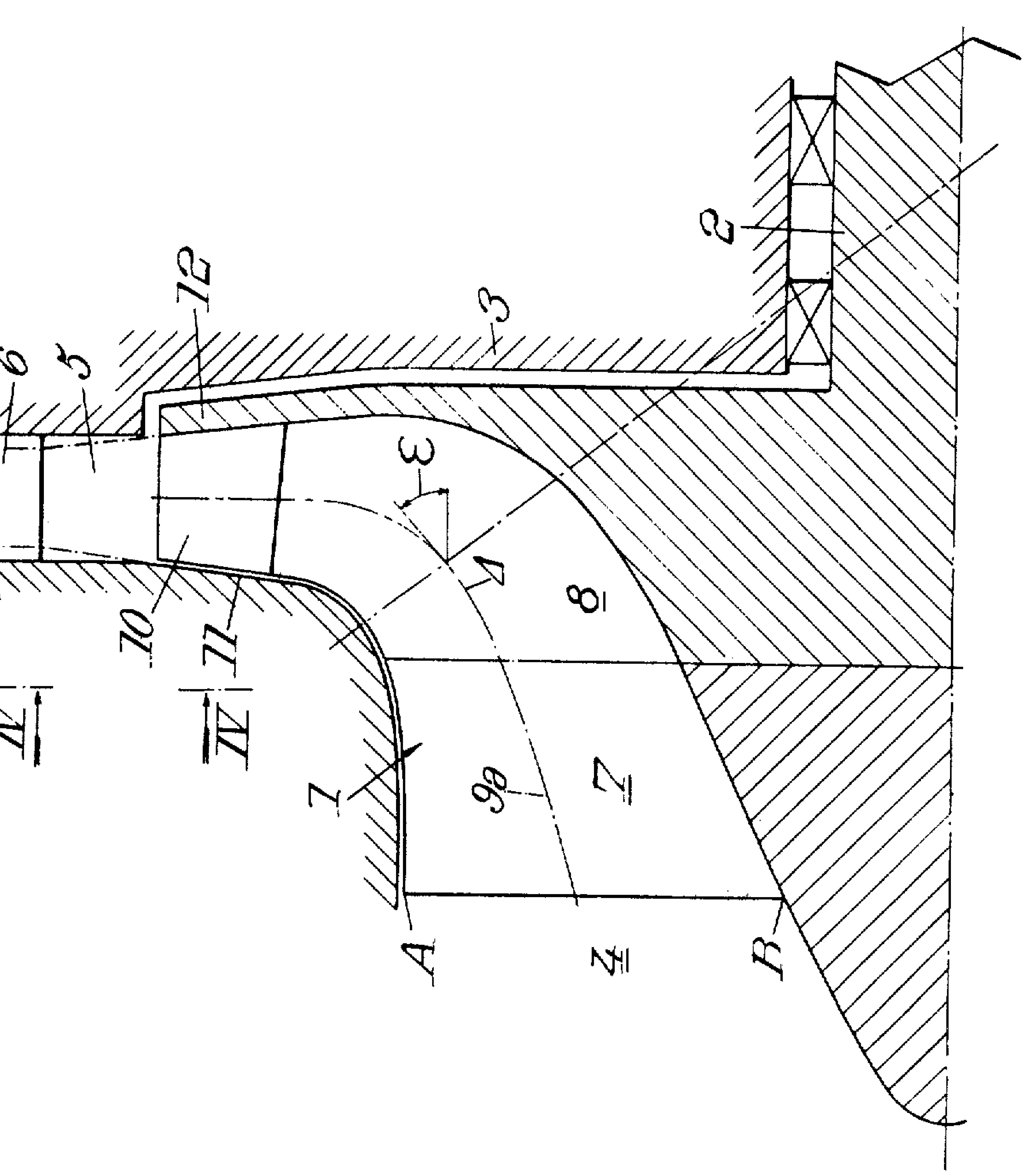


Fig. 1.

Fig. 2.

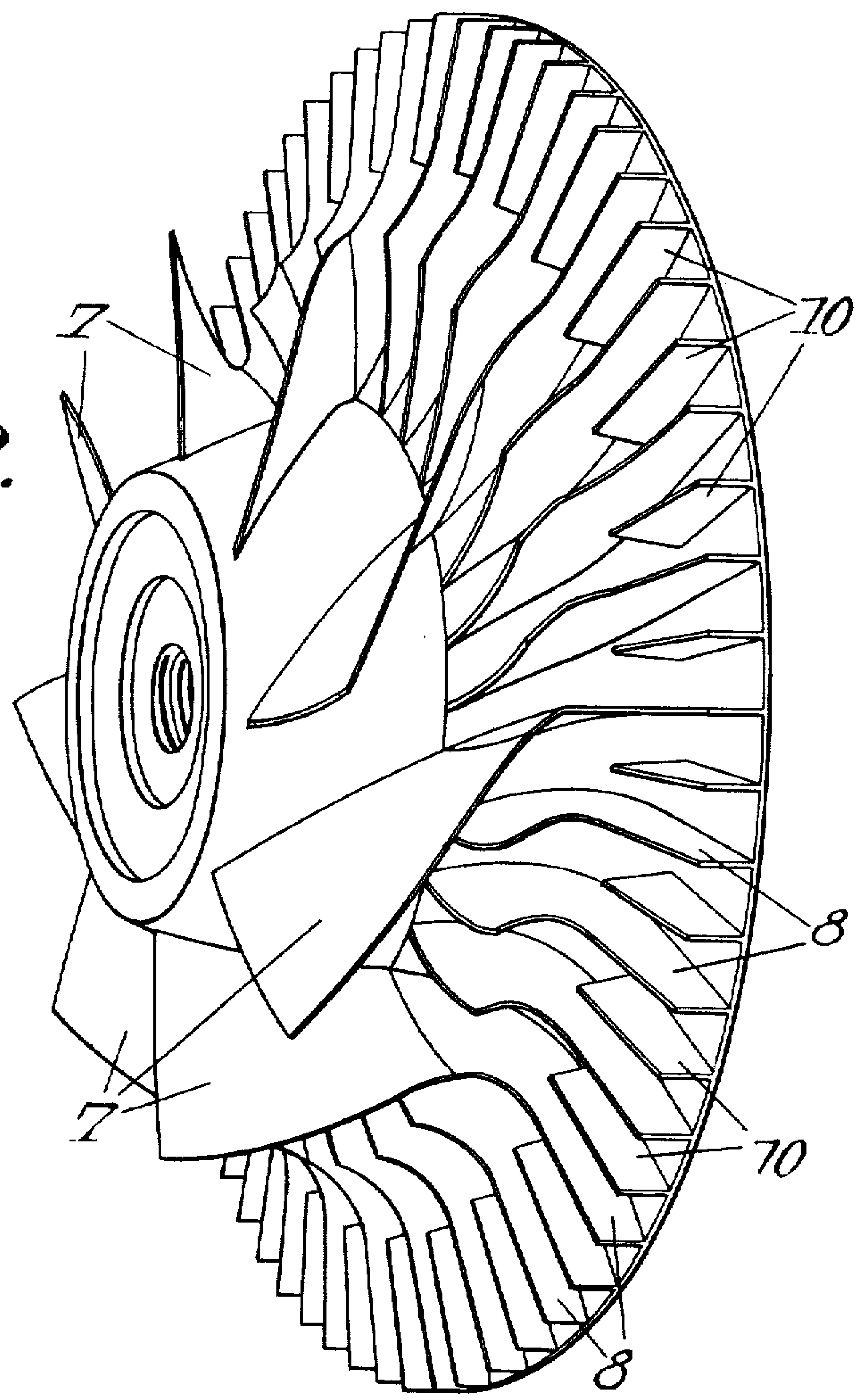
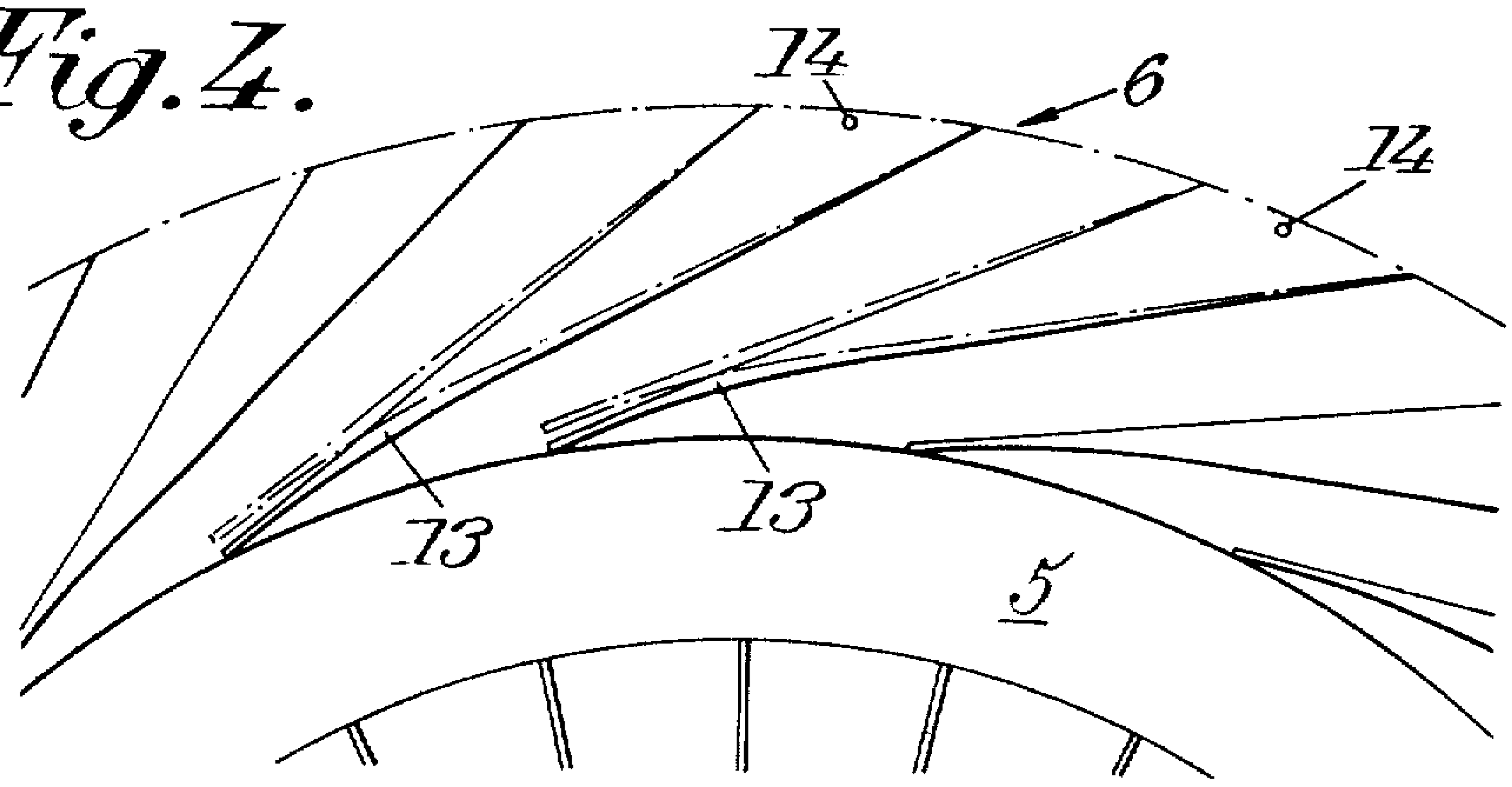


Fig. 4.



SUPERSONIC CENTRIFUGAL COMPRESSORS

BACKGROUND AND SUMMARY OF THE INVENTION

The invention relates to supersonic centrifugal compressors and more particularly to compressors which are adapted to provide a large flow rate which is large as compared with the front dimensions of the rotor.

A conventional centrifugal compressor comprises a rotor consisting of a disc secured to a shaft and provided with blades and mounted in a housing comprising a diffuser. Fluid enters through an axial aperture in the housing (or two apertures in the case of a double-flow compressor) and is accelerated by the blades, after which its pressure increases owing to slowing down with respect to the blades and owing to centrifugation. On leaving the rotor, the fluid has considerable kinetic energy which is recovered in the form of pressure in the diffuser. There are numerous known kinds of diffusion systems, the most widely-used being the vaneless diffuser and the vaned diffuser when the intake flow is supersonic, and the scroll.

If a centrifugal compressor is to provide a high compression rate, the rotor peripheral speed must be high, typically about 600 m/s for compression rate of about 10 if the fluid is air. Under these conditions, the gas flow leaving the rotor has a supersonic absolute velocity. In the case, as before, of a centrifugal compressor providing a highly specific flow rate (the ratio of the mass flow rate to the frontal cross-section of the rotor) and a high compression rate (e.g. 10 as hereinbefore mentioned) the absolute velocity at the blade tip at the rotor intake will usually also be supersonic. When the ratio between the peripheral radius and the intake radius of the rotor is 1.5, the relative velocity at the blade tip at the rotor intake is considerably greater than that of sound, typically of about MACH 1.3. Since the flow is supersonic both in the rotor intake regions near the blade tips and in the diffuser, recompression shock waves will necessarily occur. The compressor efficiency is closely dependent on the manner in which the flow is organized in the shock wave regions and on the stabilities thereof. Furthermore, in most prior-art compressors, the tangential and axial velocities of the fluid at the rotor outlet are not uniform, thus reducing the efficiency of a vaneless diffuser surrounding the rotor so that it is necessary, in many cases, to provide the rotor with guide means, such as described in French Pat. No. 7,219,200 of the assignee of the present application, while such guide means are satisfactory, they have the disadvantage of making the compressor more complicated.

Other known centrifugal compressors (French Pat. application No. 2,023,770) comprise an upstream wheel forming an axial compressor followed by a wheel keyed to the same shaft and having blades separated from those of the outer wheel by an axial gap for flow stabilization. The latter approach increases the axial length and weight of the compressor, which is objectionable in aeronautics, and renders it necessary to machine two sets of blades haaving different characteristics, thereby rendering machining more intricate and costly.

It is an object of the invention to provide a supersonic centrifugal compressor which is improved with respect to prior-art supersonic compressors. It is a more specific object to provide a compressor whose efficiency,

more particularly that of the rotor, is improved, and in which a relatively uniform tangential and axial flow is achieved at the rotor outlet, i.e. in the axial direction of the rotor.

To this end, in a supersonic centrifugal compressor comprising a bladed rotor and a stationary diffuser surrounding the rotor and borne by a housing having at least one axial fluid intake, the fluid to be compressed successively travels through an intake region wherein the rotor has a small number of blades producing a slight tangential deflection in the fluid and a slight divergence in order that the recompression shock be stabilized near the leading edge on the compressing surface of the blades and breakdowns in the flow and a compression region. In the compression region, the rotor has a higher number of blades simultaneously producing tangential deflection of the fluid and deflection in the meridian plane. Last, the fluid flows through a region where the flow is substantially radial with respect to the rotor.

The tangential deflection between two successive points along a same flow line is the variation in the stag between the two points. The stag is the angle formed by the tangent to the flow line at a given point and the meridian line of the plane which is tangent to the flow surface at the same point.

The intake region is also designed to stabilize the recompression shock wave. In that same intake region, the deflection along the upper surface of the blades and the divergence may advantageously be of about 7°. Usually, the number of blades in the intake region is between six and twelve. In the compression region, between **24** and **32** blades can be provided, the number being higher if the front size of the rotor is greater. As an example, intermediate blades, e.g. 3 or 4 intermediate blades per main blade, can be provided between the blades extending all the way along the rotor, starting from the intake. In the outlet region, additional short substantially radial blades are provided and adapted to prevent breakdowns or stall in the flow on the upper surface. In practice, at least **32** blades are usually required in the outlet region. In many cases, it may be sufficient to provide an additional guide blade in the outlet region in the middle of each duct bounded by two longer blades.

The compressor according to the invention is suitable for use in a number of technical fields. More particularly, it may be used in aeronautics as a turbo-jet intake component. It can also be used in industry, inter alia when heavy gases have to be compressed and when it is important to obtain high efficiency and/or a high specific flow rate, e.g. for compressing heavy gases, e.g. uranium hexafluoride used in isotopic enrichment plants.

The invention will be more clearly understood from the following description of a particular embodiment of a supersonic compressor given by way of example.

SHORT DESCRIPTION OF THE DRAWINGS

FIG. **1** is a diagrammatic half sectional view of a supersonic compressor, showing the essential components thereof, along an axial plane;

FIG. **2** is a perspective view of the compressor rotor;

FIG. **3** is a simplified graph showing part of the intake of the rotor blades and the corresponding velocities diagram; and

FIG. 4 is a simplified view along IV—IV of FIG. 1, showing the rotor outlet and part of the diffuser.

DESCRIPTION OF A PREFERRED EMBODIMENT

Referring to FIG. 1, there is shown a centrifugal compressor for use where maximum reduction of the front size of the rotor is not a primary requirement. Such compressors may be used mainly in stationary installations. Since the rotor diameter is relatively large, the total number of vanes can be substantially higher than in the case of a compressor for use in aeronautics, when the radial dimensions have to be smaller.

The compressor comprises a single flow rotor **1** carried by a driving shaft **2** and provided with blades. A vaneless diffuser **5** and a vaned diffuser **6** are provided around the rotor in a housing **3** which surrounds the rotor and limits an intake **4**. The rotor **1**, which is provided with blades, draws the fluid to be compressed (e.g. a heavy gas) through intake **4** and forces it into the vaneless diffuser, whence it travels into the vaned diffuser **6** and finally into an annular scroll (not shown) surrounding the vaned diffuser.

During its travel from intake **4** to diffuser **5**, it can be considered that the fluid travels through three successive regions.

The intake region begins at the leading edge of blades **7** (FIGS **1**, **2** and **3**). As already stated, the flow will be considerably above sonic speed at the blade tip at point A (e.g. the relative speed of the fluid with respect to the blades will be MACH 1.3) and will be only slightly below sonic speed at the root of the blade at point B (e.g. MACH 0.9) Consequently, under started flow conditions, e.g. during normal operating conditions, a recompression shock wave **8***a* will form and will be connected to an oblique shock wave coming from the leading edge. The shock wave **8***a* must be in a region where the fluid is only very slightly deflected by the blades and where the flow is only very slightly divergent in order to prevent breakdown or stall therein.

In practice, in spite of the increased thickness of the blades from the leading edge onwards, there is a divergence owing to the blade curvature, and the latter must be moderate (more particularly on the upper surface of the blade). More particularly, the diffuser throat cross-section must be selected so that the shock wave **8***a* is kept in the immediate neighbourhood of the leading edge at the blade tip, on the compress-surface side (i.e. near the line AC). It is known that, if the throat section is decreased, the shock wave tends to move towards the leading edge. Even if the shock wave is substantially level with AC, there is no appreciable risk of surge, since the relative flow speed remains subsonic at the root of the blade.

In order to obtain satisfactory efficiency, the recompression shock wave **8***a* must be in a region in which the fluid is only very slightly deflected by the blades, since this minimises overspeeds along the upper surface, i.e. the intensity of shocks, and prevents considerable breakdown in the flow or stall. In practice, 7° may be considered as near the optimum value for the angle of deflection between the leading edge A of the blade and the point C where the blade intersects the perpendicular from the leading edge A of the next blade. A divergence of 7° is likewise acceptable.

To obtain a large mass flow rate, only a small number (about 8) of blades **7** must be provided in the intake region, so that the total thickness of the blades does not excessively reduce the most restricted cross-area left for fluid flow. Advantageously, the blades are narrow and have a high hub ratio (ratio between the radii of the blade tip and the blade root) in the intake cross-section.

That ratio may be from 2 to 2.2. The thickness of the blade, in an embodiment which will again be referred to later, is e.g. 1.5 mm at the blade root and 0.75 mm at the blade tip. The stag angle of the leading edge of the blade tips is e.g. 60° to 65° with respect to the axis.

Since there is a small number of blades, the limit flow rate which can be provided by the rotor can be determined relatively accurately from the triangle of velocities given in FIG. **3**. The rate of flow velocity (or meridian velocity C_m) can be calculated by constructing the triangle of velocities, since the rotating speed ω_r and the direction of the fluid flow relative to the blades (substantially in the direction of the blades in the intake region AC) are known.

On leaving the intake region, the fluid travels into the deflection region, in which the blades azimutally (or tangentially) deflect the flow to an angle which is usually of about 58° if the total deflection at the blade tip is of the order of 65° and if, as already stated, the deflection in the intake region does not exceed 7°.

The following steps may be taken in order to obtain a large deflection together with satisfactory uniformity of flow at the rotor outlet, where the blades are substantially radial.

First, the blades in the compression region are shaped so that they deflect the flow simultaneously in the tangential direction and in the meridian direction. The center line **9***a* (FIG. **1**) is given a radius of curvature defined by the following conditions: the radius of curvature of the central flow meridian (or centroid) line **9***a* is such that the pressure gradient is zero along a direction at right angles to the centroid line. Starting from this line, the blade skeleton is defined by a straight line which bears on the rotor axis and the central current line **9***a* while remaining perpendicular to the centroid current surface (which is a surface of revolution, while the meridian plane has an angular position which changes along the blade).

The corresponding equations can be written as follows:

$$-\frac{1}{\rho}\frac{\delta p}{\delta n} = -F_n + \frac{C_m^2}{R} - v^2\frac{\cos\epsilon}{r} = 0 \qquad (1)$$

$$F_n = 0 \qquad (2)$$

$$R = \frac{C_m^2\, r}{v^2 \cos\epsilon} = \frac{ds}{d\epsilon} \qquad (3)$$

In addition to these equations, the following conditions in respect of the central meridian or centroid line (the line whose distance at each point from the axis is such that it divides the flow into two parts having the same cross-section) should be fulfilled:

— the rate of variation in the meridian velocity, in dependance on the curvilinear abscissa along the line should be substantially linear or proportional,

— the rate of variation of the tangential deflection in dependence on the curvilinear abscissa s along the meridian line is s^α, with α between 1.5 and 2.

Equation (1) defines the average flow line, in conjunction with the above-mentioned conditions. In the equations, F_n denotes the force component at right angles to the blade, i.e. in the direction of the line Δ in FIG. 1; the term

$$\frac{C_m^2}{R}$$

corresponds to the force produced by the curvature of the flow surface; and the third term corresponds to the centrifugal force, r denoting the radius at the point in question and ϵ denoting the angle between the axis and the tangent to the meridian flow line. The third equation, in which C_m denotes the meridian velocity and v denotes the absolute tangential velocity, defines the radius of curvature R. The radius of curvature R of the central meridian line is infinite at the intake at the outlet, ϵ being equal to 90° at the outlet and v (the tangential velocity) being substantially zero at the intake.

For a same curvilinear abscissa, the deflections of the blade tip and the blade root as determined by the previously-defined method of generation are different from the deflection along the central meridian line. The coefficient α should be made low enough to ensure that, starting from constructive data (e.g. the hub ratio and the intake angle), a tip deflection is computed which does not detrimentally affect the mechanical stresses at the blade root to an excessive extent. If the peripheral velocity (at the blade tip) is too high, the skeleton may have to be generated from straight lines which are not perpendicular to the central meridian line any longer.

Another step is to increase the number of blades, in order to ensure satisfactory efficiency and uniform velocities in the compression region. The blades **7** extending all along the rotor are supplemented with blades **8** such that the total number of blades in the compression region is between 24 and 32. To this end, in the example shown, three additional blades **8** are inserted between each two blades **7**. The blades **8** have a shape which is identical with that of that part of blades **7** in the same annular portion. The thickness of the leading edge will be greater than that along A-B, since the speed is sub-sonic. In the example considered hereinbefore, the thickness and diameter of the leading edge may be 1.5 mm at the tip and 3 mm at the root of blades **8**.

In order to simplify manufacture, the rotor can be made in two components. A first component, comprising the upstream wheel, bears those portions of the blades **7** which are upstream of the leading edge of blades **8**. A rear part, forming the wheel proper, bears the rest of the blades. The rotor can be machined after the wheel and the outer wheel have been assembled.

Last, after substantially complete tangential deflection, the fluid flows through a region of substantially radial flow which comprises that part of the blades **7** and **8** which extends until the rotor outlet and supplemental short blades **10** adapted to guide the flow and prevent breakdown from occuring therein on the upper surface of the blades.

The inner and outer walls of the rotor illustrated by way of example in FIG. **1** are designed so that the meeridian velocity in each cross-section taken at right angles to the flow surface corresponding to $9a$ increases in direct proportion of the curvilinear abscissa along the central meridian line; this result is achieved by providing the radial flow region with convergent surfaces **11** and **12**. If, for example, a convergent angle of 12° is selected, the surfaces of the radial portion will be inclined at 6° to the radial direction. Of course, an asymmetrical arrangement is possible or even necessary in a double flow compressor.

The flow speed at the rotor outlet is uniform in both the axial and the tangential direction. Since the flow is axially uniform, a vaneless diffuser **5** may be used which is known to have a low efficiency if there is marked non-uniformity. A vaneless diffuser has the advantage of reducing the absolute supersonic velocity of the flow at the rotor outlet, without producing a shock.

An industrial compressor according to the invention can include a large vaneless diffuser **5**; the ratio between its outer and inner diameter can be up to 1.30. Space requirements frequently render a lower ratio — at most 1.15— advisable. In an aeronautical compressor, having a compression rate of 5 to 6 and for M ~ 1.5 at the intake, the ratio is frequently limited to 1.05 in order to reduce bulk.

A vaned diffuser **6** is disposed downstream of the vaneless diffuser **5** and has an axial depth equal to that of the wheel (i.e. corresponding to the axial length of the trailing edge of the blades). Advantageously, the number of ducts is large, so that the bulk can be reduced. The leading angle can be low, e.g. between 6° and 12 °, with respect to the tangential direction, but increasing the diffusion length for a given radial size. The diameter of the leading edge of each blade of diffuser **13** is typically about 5 to 10% of the throat width. The inlet of each actual duct forms a throat having a length which is substantially equal to half its depth, and adapted to stabilize the recompression shock waves. Downstream, the blades define a duct having an angle of divergence of approximately 6° to 7°. The uncovered part of the upper surface of each blade, i.e. the intake region from the leading edge which does not bound a duct, is preferably in the form of a spiral. The ratio between the outlet cross-section of the diffuser and the intake cross-section perpendicular to the flow may be e.g. between 3 and 3.5 for a divergence of 6°.

The blades can be angularly adjustable around a shaft **14** so that the throat width can be adjusted by moving the blades from the broken-line position to the continuous-line position in FIG. **4**, in order to determine the position for optimum efficiency.

The vaneless diffuser **5** having parallel surfaces could be replaced by a convergent diffuser extending the rotor ducts, as illustrated in chain-dotted lines in FIG. **1**.

In the example given, the diffuser had **23** channels, and this has given satisfactory results. A centrifugal compressor of that type can be constructed which provides a per stage efficiency greater than 80% for a compression rate between 6 and 8, if the fluid is air.

I claim:

1. A supersonic centrifugal compressor comprising a rotor provided with at least first blades and second blades and with a stationary diffuser surrounding the rotor and carried by a housing having at least one axial fluid inlet, wherein said rotor and housing limit an intake region, a compression region and a radial flow region through which the fluid flows successively and wherein said blades each have a tip portion and a root portion and extend continuously from a leading edge to a trailing edge, the trailing edges of all said blades being located in said radial flow region,

said intake region extending from the leading edges of said first blades to the leading edges of said second blades, said second blades being shorter than the first blades, said first blades having a stag variation in the intake region which is such as to produce a tangential fluid deflection which is small as compared with the tangential flow deflection in the compression region, and having a substantial stag at their leading edges,

said first and second blades being so shaped in the compression region as to produce the major portion of the tangential flow deflection and azimuthal flow deflection,

and said first and second blades having trailing end portions directed radially with respect to the rotor in said radial flow region.

2. A compressor according to claim **1**, wherein the deflection along the upper surfaces of the blades between the intake and the orthogonal projection of the leading edge of the next blade is about 7° at the tip of the blade, and the divergence between the tips of adjacent blades is about 7°.

3. A compressor according to claim **1**, wherein the diffuser comprises a vaneless diffuser surrounded by a vaned diffuser and the ratio of the outer diameter of the vaneless diffuser to the rotor outlet diameter is between 1.15 and 1.30.

4. A compressor according to claim **1**, wherein in the radial flow region, the rotor has third blades whose leading edges are located at the outlet of the compression region.

5. A compressor according to claim **4**, wherein the thickness of the first blades increases from their leading edges up to a cross-section level with the leading edges of the second blades.

6. A compressor according to claim **4**, wherein the rotor has from six to 12 first blades, from 24 to 32 blades in the compression region, and at least 36 blades in the radial flow region.

7. A compressor according to claim **1**, wherein the absolute outlet angle of the fluid from the rotor is between 6° and 12° with respect to the tangential direction.

8. A compressor according to claim **7**, wherein the diffuser has vanes and the angular position of the diffuser vanes can be adjusted so as to alter the cross-section of throats of said diffuser.

9. A compressor according to claim **1**, wherein the hub ratio in the compressor intake cross-section is of the order of 2 to 2.2.

10. A compressor according to claim **1**, wherein the stag angle of the leading edges of the rotor blades at the tip of the blades is from 60° to 65°.

11. A compressor according to claim **4**, wherein the rotor consists of an upstream wheel bearing a front fraction of the first blades, and a wheel which carries the balance of the blades.

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