United States Patent [19]

Pedersen et al.

[11] **3,790,301**

[45] **Feb. 5, 1974**

[54] **AXIAL FLOW FAN, THE IMPELLER VANES OF WHICH ARE ADJUSTABLE DURING OPERATION**

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[22] Filed: **July 1, 1971**

[21] Appl. No.: **158,794**

[30] **Foreign Application Priority Data**

July 10, 1970 Denmark 3610

[52] **U.S. Cl.** **416/108,** 416/114

[51] **Int. Cl.** **F04d 29/36**

[58] **Field of Search** 416/108, 109, 112–115, 416/167, 168

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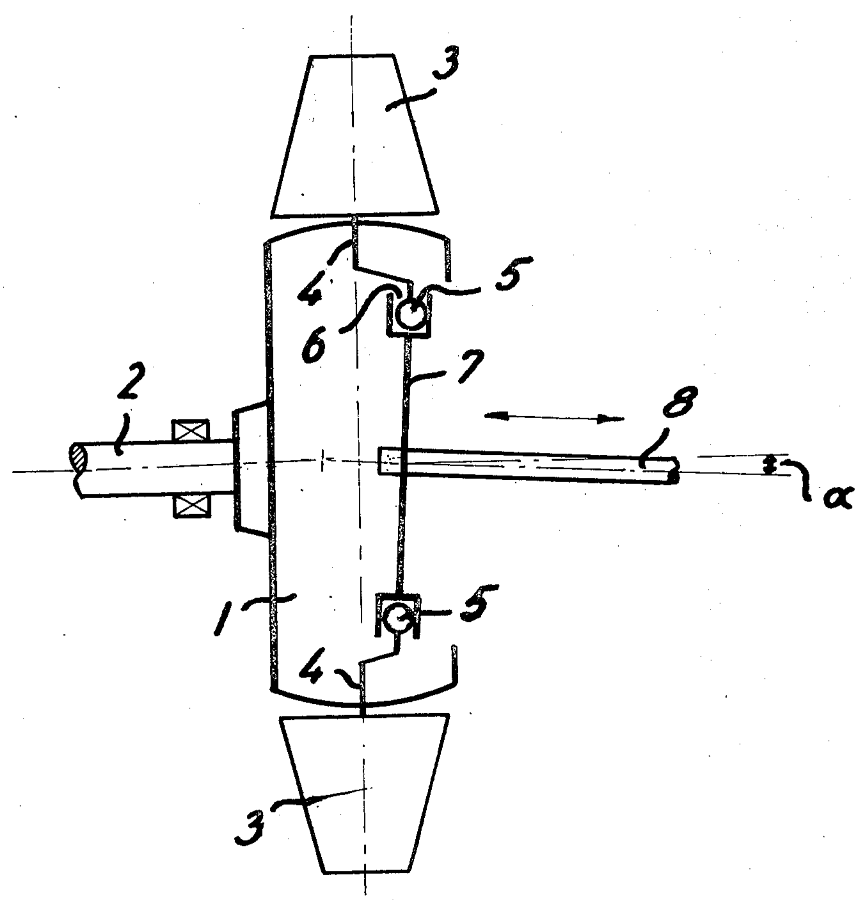
Attorney, Agent, or Firm—Richard C. Sughrue et al.

[57] **ABSTRACT**

An axial flow fan comprising a plurality of impeller vanes that are angularly adjustable during operation by means of an adjustment member operatively connected with the vanes and having an axis that either forms a small angle with the axis of rotation of the impeller or has a small transverse distance therefrom. The adjustment member may be axially or angularly displaceable for causing the desired vane adjustment in such a way that during each revolution of the impeller the vanes are successively acted upon by the adjustment member, i.e., at least one but not all of the vanes are adjusted at a time.

10 Claims, 15 Drawing Figures



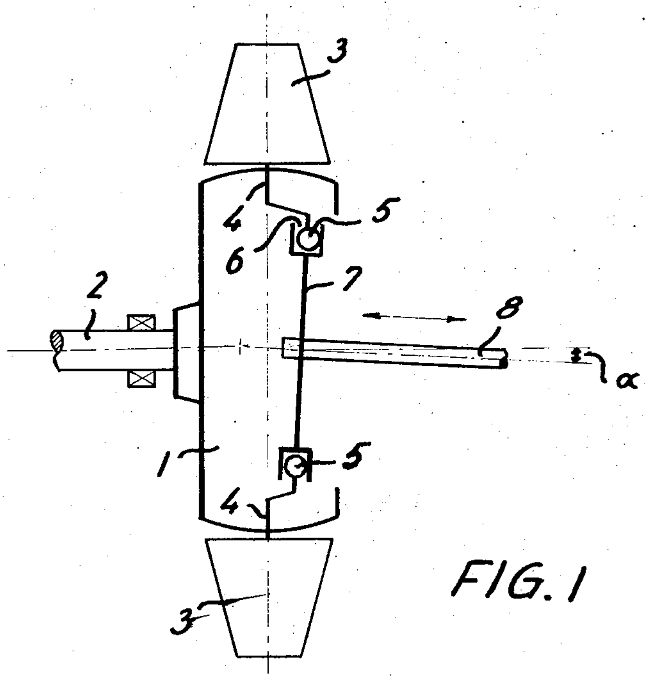


FIG. 1

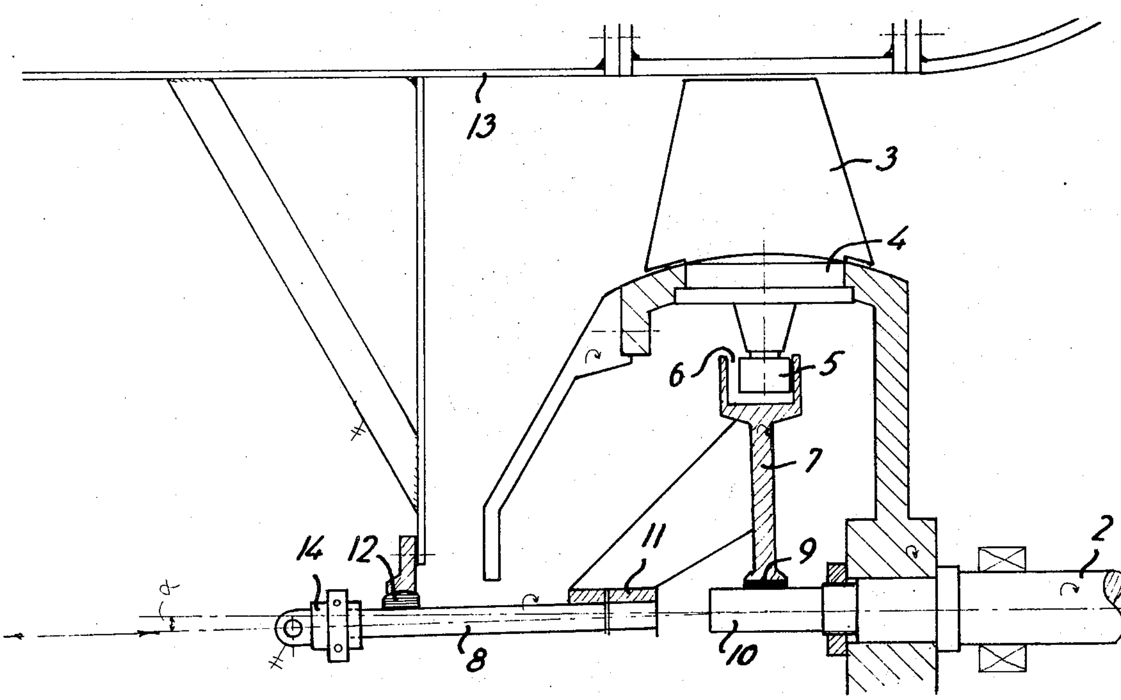


FIG. 2

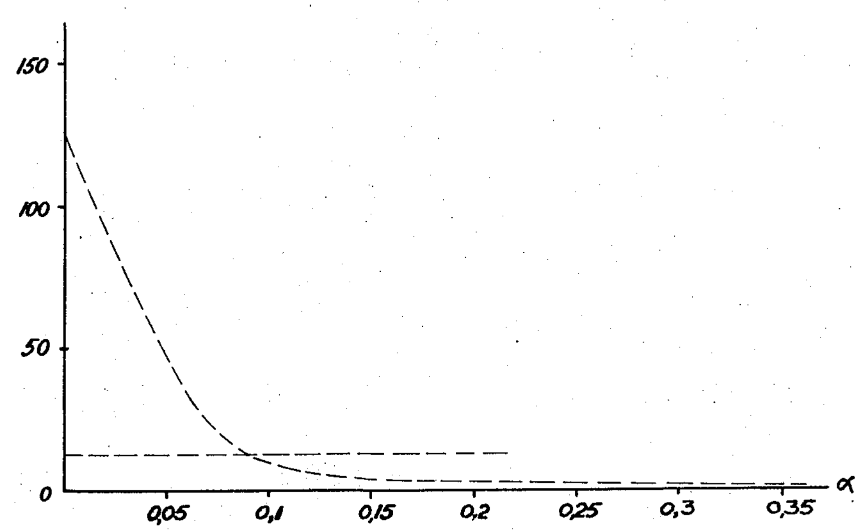


FIG. 3

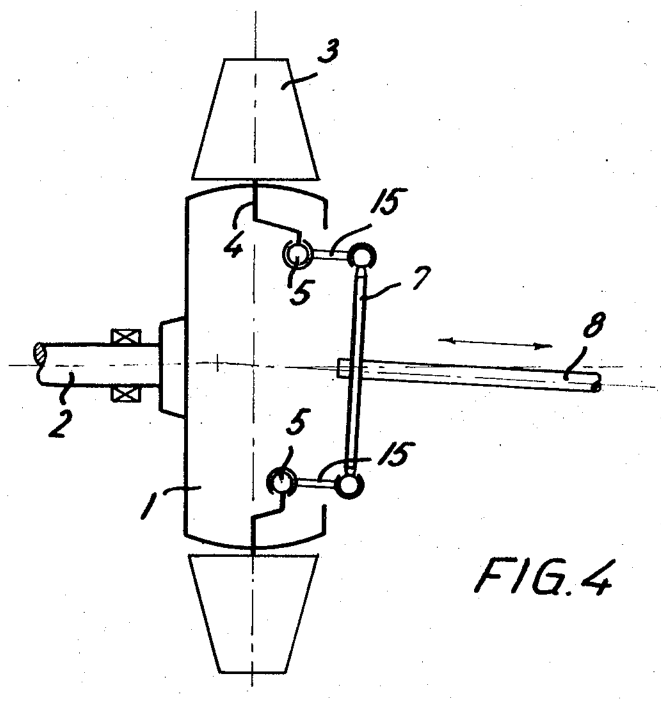


FIG. 4

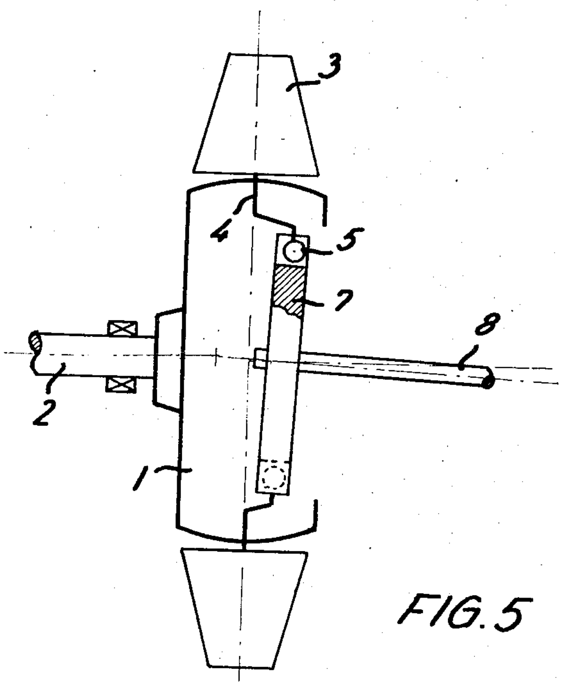


FIG. 5

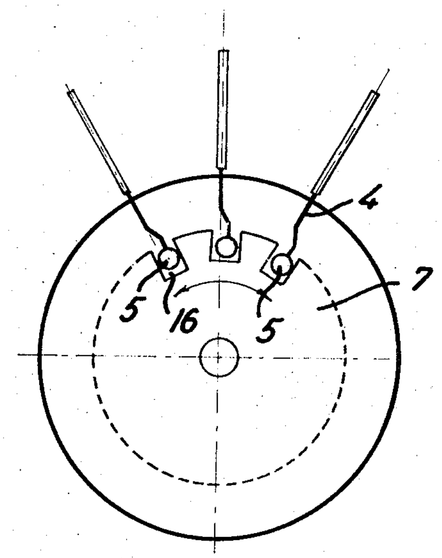


FIG. 6

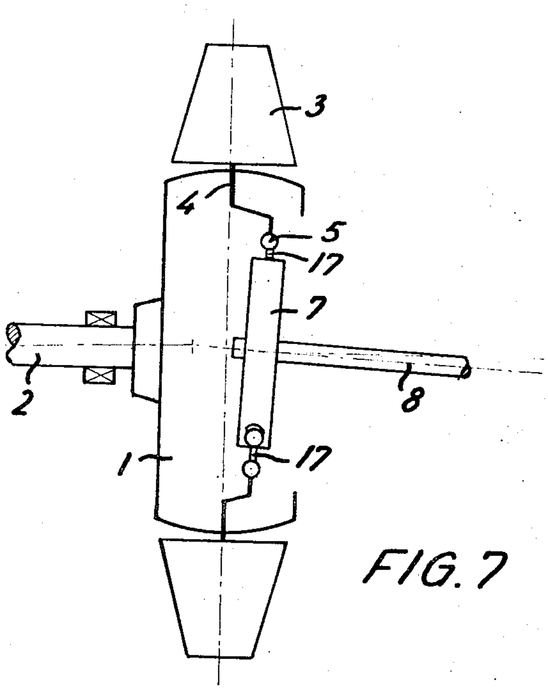


FIG. 7

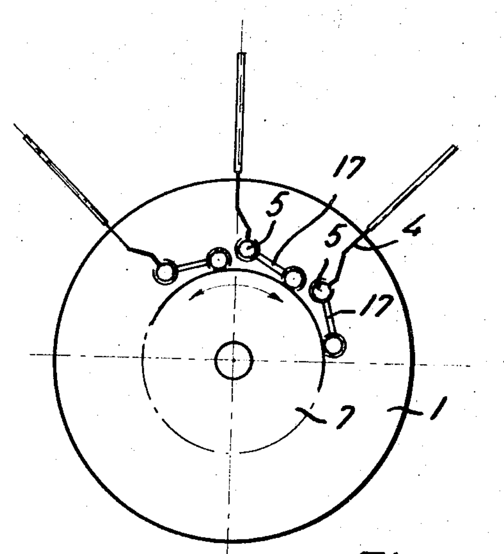


FIG. 8

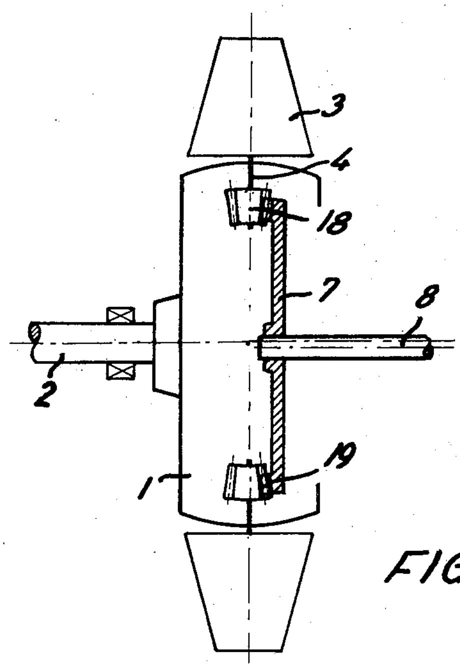


FIG. 9

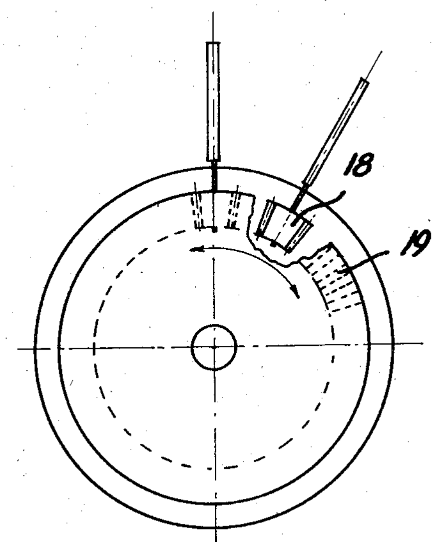


FIG. 10

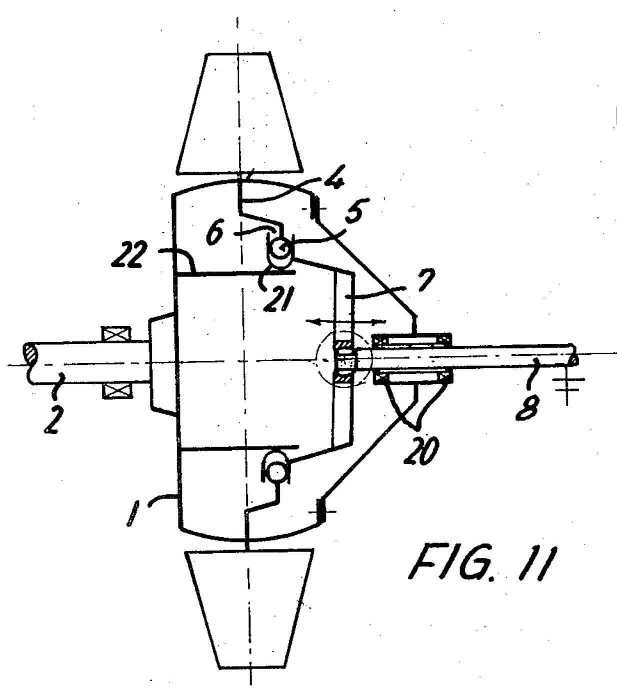


FIG. 11

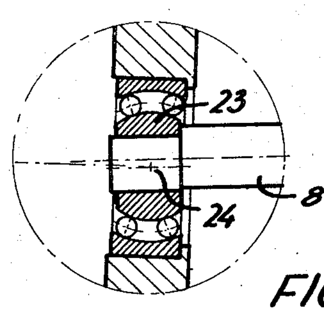


FIG. 12

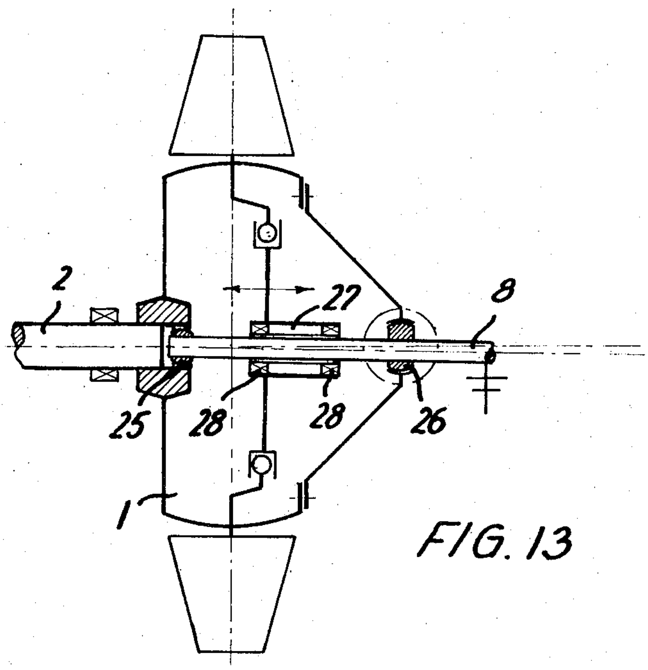


FIG. 13

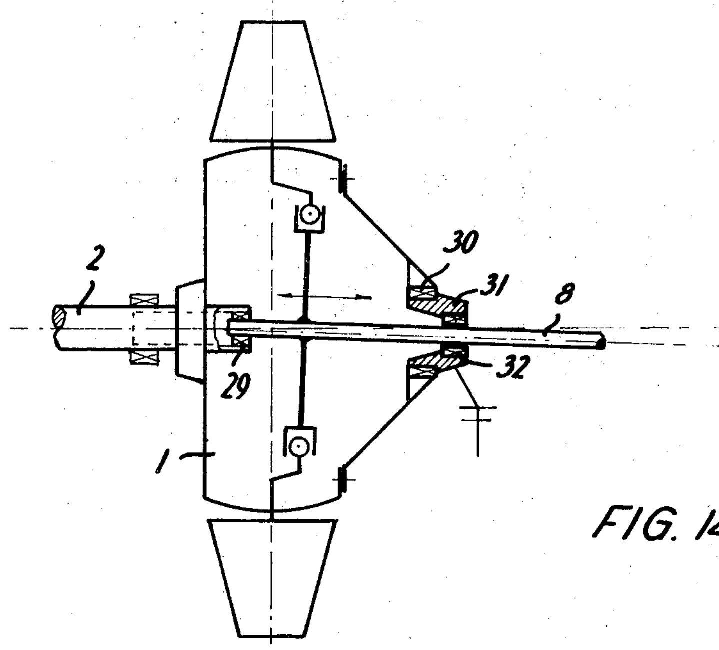


FIG. 14

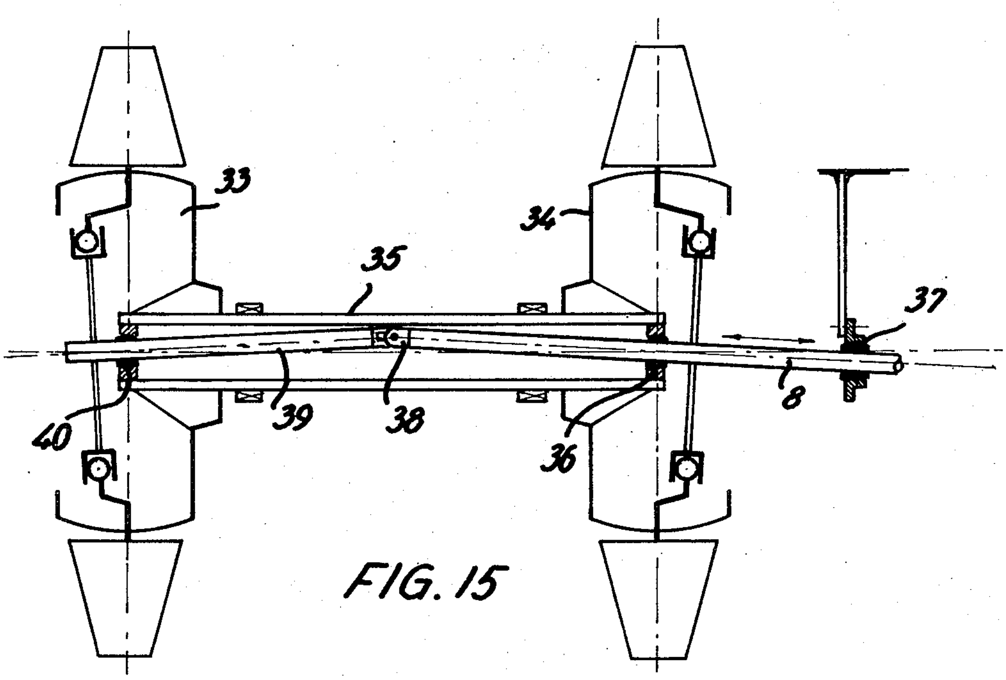


FIG. 15

AXIAL FLOW FAN, THE IMPELLER VANES OF WHICH ARE ADJUSTABLE DURING OPERATION

The invention relates to an axial flow fan, the impeller vanes of which are adjustable during operation, the vane shafts being journalled in the impeller hub and each provided with an eccentrically mounted element which is actuated by a common adjustment member.

With the prior art axial flow fans, very substantial forces are required for displacing the adjustment member in order to obtain a desired change in the vane position. This is connected with the fact that during operation there is very high friction in the bearing means of the vane shafts since each vane, during the operation of the fan, is subjected to a very high centrifugal force that has to be resisted by the axial bearing of the vane shaft. For constructional reasons, this shaft can only be of a rather small diameter, just as the eccentricity of the eccentrically mounted member is very limited so that the force to be transmitted from the adjustment member to the eccentrically mounted member actuates on a relatively short arm.

In actual practice, for example, the adjustment force required may amount to 25 kgs. which, in the case of an impeller having 10 vanes, means that the requisite adjustment force to be applied in the axial direction amounts to 250 kgs.

Especially when the adjustment is to be automatically controlled, it is difficult to apply such high forces and for this reason various gear systems of a more or less complicated construction have been developed so that by means of a suitable gearing ratio, the outer force that is to be applied in order to effect the adjustment is reduced.

However, this too gives rise to difficulties, because the adjustment must be effected inside the impeller while it is in rotation.

It has now been proven that these difficulties can be greatly reduced by means of an impeller according to the invention, which is characteristic in that the adjustment member is so designed and arranged that the said eccentrically mounted element of each vane, in the course of an adjustment movement, is influenced by the adjustment member in the direction of adjustment for a part only of each revolution of the impeller in such a way that, at the most, the total number of vanes less one, and, at least, one single vane at the a time are subjected to the adjustment action in the course of an adjustment movement.

The common adjustment member can, as known, be stationary in relation to the rotational movement of the impeller, or it can rotate in conjunction with the impeller. In both instances, the adjustment movement may be a displacement along the axis of the adjustment member or an angular displacement around this axis.

The adjustment member may have many different configurations; it may, for example, be a circular disc having a circumferential groove, in which eccentric pins on the vanes are engaged and which is displaced in the axial direction in order to adjust the vanes. This form of construction of the adjustment member is taken as the basis for the following explanation of the effect achieved by means of the invention.

When the invention is utilized in an axial flow fan, the adjustment member of which is formed by such a disc, the axis of the disc and the axis of rotation of the impeller must include a certain angle so that the plane of the disc forms an angle with a plane perpendicular to the axis of rotation of the impeller. As a result of this inclined position of the adjustment disc, when no adjustment occurs the distance of the contact surface of the disc from a specific pin in the course of one revolution of the disc varies according to a sinusoidal function and generally in such a way that the distance varies between 0 and a maximum value which is determined by the inclination. So far as different pins are concerned, the individual distance variations will include a phase displacement corresponding to the angular distance between the associated vanes. When an adjustment occurs by a displacement of the disc in its axial direction, the sinusoidal variation will be superimposed with a linear variation perpendicular to the abscissal axis of the sinusoidal curve, but this linear variation, i.e., the displacement movement, will generally be so slow in comparison with the rotational velocity that during a single revolution only a relatively minor displacement of the sinusoidal curves of the individual pins may occur, and these pins are subjected to a pressure one at a time, so that in this way only a reduced force is required for the adjustment as compared to the force necessary when all the pins are to be simultaneously subjected to equally high adjustment forces.

It will be apparent that it is possible thereby to reduce the requisite adjustment force to the force which is needed to adjust a single vane, which thus represents a substantial reduction, but tests have shown that even a more significant reduction can be achieved, for example, to less than half of this force.

It is rather difficult to present an exhaustive and exact theory that would explain these conditions.

Of course, the basic principle of the explanation will have to be that a part of the adjustment force originates from the fan motor as a consequence of the inclined position of the disc.

Just how it is possible for this to come about may be explained as detailed below.

First of all, it has to be imagined that the adjustment disc is immobile in the axial direction while the vanes assume a specific position. For the sake of clearness it may be presumed that there only are two diametrically opposed vanes with associated adjustment pins.

Looking at one pin and one side of the groove in the disc, the distance separating these will gradually decrease from a maximum value to the value 0 when contact takes place. This contact is presumed to take place before the point is reached at which the relative axial movement of the disc is reversed so that during a certain fraction of a revolution the contact force varies from 0 to certain maximum value and, subsequently, back to 0, while a correspondingly varying elastic deformation takes place.

If it is imagined that the maximum value of the force in question does not reach the force that is required for effecting a movement of the pin, no change in the vane position takes place during the rotation of the disc.

While one pin contacts one side of the groove, the diametrically opposed pin will be in contact with the other side of the groove and a force is thus exerted also on this opposed pin during the said fraction of a revolution, which force is insufficient for effecting an adjustment of the vane.

The forces acting on the two pins are oppositely directed, which is to say that no axial reaction is created in the disc shaft and, thus, no load on the external adjustment system. The work performed is converted exclusively into heat and the force originates solely from the drive motor of the fan.

If a certain force is now exerted in the axial direction of the disc via the external adjustment system, then one pin, at the moment when the two forces previously mentioned are at their maximum, will be subjected to a force in the adjustment direction which is composed of the said maximum value and the applied axial adjustment force, while the other pin is subjected to the difference between the two forces.

If it is imagined that a force of 25 kgs. is required to displace the pin and that the said maximum value of the force applied by the rotation of the disc is 24 kgs. an external adjustment force of slightly more than 1 kg. will result in an adjustment of the pin in question because the total of the two forces becomes more than 25 kgs. while no movement of the second pin takes place since the force applied to this pin is somewhat less than 23 kgs.

The displacement of the first pin takes place for so long as the total of the two forces exceeds 25 kgs. which will be the case during a relatively small fraction of half a revolution and, in the course of this half revolution, the second pin remains immobile.

The same repeats itself during the following half revolution but in such a way that the first pin will remain immobile while the second pin is moved.

When an axial force of 1 kg. is thus exerted on the disc shaft in a certain direction, a corresponding adjustment of the two vanes will take place in short jerks and in such a way that one of them always is immobile when the other one moves.

In this way, an adjustment is effected by means of an external adjustment force of 1 kg. because the 24 kgs. that are required to attain the 25 kgs. necessary for the adjustment are produced by the drive motor of the fan.

The aforesaid explanation will certainly have to be regarded as a very simplified one and is scarcely exhaustive. It may thus also be imagined that for a complete explanation it will be necessary to take into account those motive forces that occur each time contact is established between a pin and one side wall of the groove. Moreover, it very likely is of significance that the initial value of the friction which has to be overcome when effecting the adjustment is higher than that of the friction in the continued movement, so that due to the inclined position of the adjustment disc, when summing up the frictional forces, only the initial friction of one of the vanes and the lower friction of all the other simultaneously moved vanes have to be considered.

How much importance is to be attached to the individual factors dealt with as well as possibly others, is difficult to decide and, presumably, depends also to some extent on the design in each individual case.

As mentioned, the adjustment movement may either be an axial or an angular displacement and in both instances the invention can be carried into effect by the inclined position of the adjustment shaft in relation to the axis of rotation of the impeller. However, in the case where the adjustment movement is an angular displacement, the axis of the adjustment member may also be parallel to the axis of the impeller and be spaced therefrom by a small distance.

It will be apparent that the effect of this feature, when the adjustment movement is an angular displacement, will be fully analogous to the one explained in the foregoing in connection with an inclined disc having an axial adjustment movement.

The invention will now be more fully explained with reference to the accompanying drawings, in which

FIG. 1 shows an embodiment of the axial flow fan according to the invention in a purely diagrammatical fashion,

FIG. 2 a part of same in greater detail,

FIG. 3 a curve illustrating the effect of the invention,

FIGS. 4 and 5 two different embodiments,

FIG. 6 an axial view of the embodiment of FIG. 5,

FIG. 7 a further embodiment,

FIG. 8 an axial view of the embodiment of FIG. 7,

FIG. 9 yet another embodiment,

FIG. 10 an axial view of the same embodiment,

FIG. 11 a still further embodiment,

FIG. 12 a detail of same on a larger scale,

FIGS. 13 and 14 two further embodiments, and

FIG. 15 an embodiment comprising two fan wheels.

FIG. 1 shows an impeller with a hub **1** rigidly connected to a drive shaft **2** and a plurality of vanes **3** journalled in the hub **1** and each having a shaft **4** and an adjustment pin **5** mounted eccentrically thereon and engaging with an external groove **6** in an adjustment disc **7** that is rigidly mounted on an adjustment shaft **8**, which is supported displaceably in an inclined position so as to form a slight angle α in relation to the shaft **2**.

From FIG. 2 can be seen how the disc **7** is slidably supported by a bearing **9** on an extension **10** of the drive shaft **2**.

Disc **7** has a body **11** that is rigidly connected to the adjustment shaft **8**, which is supported in a bearing **12** that is rigidly connected to the fan housing **13**.

14 is a coupling element which is mounted for rotation on the end of adjustment shaft **8** but is held against axial displacement thereon, so that it is possible, by means of a pulling or pressing force, to displace the disc **7** in the axial direction, whereby it exerts an action on the eccentrically mounted adjustment pins **5** and, thereby, on the vanes **3**.

By the bearing **12** being mounted eccentrically in relation to drive shaft **2**, the desired inclined position of adjustment shaft **8** and, thereby, of disc **7**, is achieved. Disc **7** is perpendicular to the axis of adjustment shaft **8**.

The disc **7** is coupled to the impeller wheel by means of carriers (not shown) so as to rotates with the same angular speed as the wheel. Seen from a specific pin, the disc will perform oscillating motion.

For this reason, sufficient play shall be provided for in groove **6** so that the oscillating movement can take place without causing a rotation of the vanes so long as the adjustment shaft **8** is not displaced in the longitudinal direction, unless for special reasons such an oscillating movement of the vanes is desired.

For a given setting of adjusting disc **7**, the excentric pins **5** on all the vanes will be situated in a plane perpendicular to the axis of rotation of the impeller wheel while the circumferential groove **6** is located in an oblique plane intersecting the pane through the excentric pins along a line perpendicular to the impeller axis. This condition, which is diagrammatically illustrated in FIG. 1, exists as soon as one complete revolution has been performed after an axial adjusting displacement of disc **7**. It is also easily understood from FIG. 1 that, during rotation of the vanes **3** and of the adjusting disc **7** rotating together therewith through one complete revolution, each of the excentric pins **5** will perform a sinusoidal transverse movement in the groove **6** during which each pin will be engaged by opposite sides of the groove at points displaced 180° relative to each other.

FIG. **3** shows a curve, in which the traction force to be exerted in the axial direction of the adjustment shaft for adjusting the vanes is shown as a function of the angle α.

The curve shown is recorded with an impeller comprising 10 vanes and having a velocity of rotation of 1,480 rpm.

When the angle is zero, that is to say with a known design in which the adjustment shaft **8** is mounted coaxially in extension of the drive shaft **2**, an adjustment force of 125 kgs. is required and since there is a total of 10 vanes, this means that an adjustment force of 12.5 kgs. is needed for each vane.

When the angle increases from zero, a variation in the load on the individual pins occurs in the course of each revolution of the disc between a point at which there is full load on the pin in question and a point at which, due to the increasing size of the angle, there is a diminishing magnitude of the load so that in the case dealt with here, at a certain angle only 9 of the pins are loaded at a time and, subsequently, with a slightly bigger angle, only 8 of the pins are, and so on until, finally only a single pin is loaded at a time.

In conformity herewith the curve shows a decreasing force with an increasing angle and, in accordance with an obvious reflection, this decrease should continue until, at a certain angle, an adjustment force of 12.5 kgs. is required corresponding to only a single pin at a time being actuated in order to rotate the associated vane.

In FIG. **1**, this force is marked with a horizontal dotted line and it is seen from the curve that the requisite force is in actual fact reduced further as the curve substantially approaches the 0-line or, at any rate, a line lying very close to the 0-line. It is thus seen that the value of 12.5 kgs. is obtained by an angle of around 0.08°–0.09°, while at 0.35° the force has come down close to 1 kg.; that is to say, to less than 1/10 of what could immediately be expected, but in conformity with the theoretical explanation given in the foregoing.

In a qualitative respect, the curve will remain generally the same irrespective of the dimensions and particular design of the impeller and the adjustment system. The quantitative course, however, will vary considerably, depending in particular on the number of vanes and the radius of the circle through the points of action of the adjustment member.

The curve of FIG. **3** is for a fan having an external diameter of only 650 mm, but in the case of impellers having bigger diameters and presenting correspondingly more space for the adjustment means and for a greater number of vanes the first part of the curve will have a substantially steeper drop than the one shown in FIG. **3.**

FIG. **4** shows a design which, in general outline, corresponds to the one shown in FIG. **1**, with the difference that the connection between the adjustment member **7** and the adjustment pins **5** is formed by means of a link **15.** The play in the link connections corresponds in this case to the play in groove **6** in FIG. **1.**

While the embodiments previously described are of the type in which the adjustment movement is an axial movement, FIGS. **5** and **6** show an embodiment in which the adjustment movement is a rotational movement.

In this instance, the adjustment pins **5** engage with notches **16** in the circumference of the adjustment disc **7.**

The fact that the effect is the same as in FIG. **1**, can be seen in the following manner. When no adjustment movement takes place, the disc **7** rotates together with the wheel. Any point of the disc **7** follows a path which when projected on to a plane perpendicular to the shaft **2** of hub **1** is an ellipse. As the angular velocity is constant, the projected speed of the point will vary so that it has a minimum value when the point is situated on the short axis of said ellipse, and a maximum value when the point is located on its long axis. This means that a notch **16** on the disc **7**, for a certain part of a revolution rotates slightly faster and, for another part of the revolution, slightly slower than the impeller, which results in varying actions on the pins in a manner corresponding to the one in FIG. **1.**

FIGS. **7** and **8** show an embodiment corresponding to the one shown in FIGS. **5** and **6**, only with the difference that the pin **5** is connected to the adjustment member **7** by means of a link **17.**

In the embodiment shown in FIGS. **9** and **10**, the adjustment elements mounted eccentrically on the shafts **4** of the vanes **3** consist of teeth on pinions **18**, while the adjustment member **7** comprises a toothed rim **19** engaging with these pinions **18.**

In this case, the adjustment shaft **8** is parallel to drive shaft **2**, but their axes are slightly spaced from each other.

Since the axis of shaft **2** is an axis of rotation for the disc **7**, the distance of any point of this disc from the axis of rotation will, in the course of one revolution, vary from a minimum to a maximum value, and since the angular velocity is constant, a corresponding variation will occur in the velocity, resulting in a forward- and backward-directed rotation in relation to the impeller and in relation to the pinions **18.** Consequently, the effect is the same as in the embodiments described in the foregoing.

FIGS. **11** and **12** show an embodiment in which the adjustment movement is an axial movement.

In this embodiment, similar to FIG. **1**, the pins **5** engage with a groove on the disc **7**, but a difference is that the adjustment shaft **8** is only displaceable and not rotatable. Moreover, it is mounted on line with the drive shaft **2** and is supported by two roller bearings **20** in a sleeve which is able to slide on the shaft **8.**

The disc **7** is guided by a bearing **21** sliding on a guide cylinder **22** mounted coaxially and rigidly inside the impeller **1**, and the disc **7** is further supported on the end of the shaft **8** by a spherical bearing **23** the outer ring of which is mounted centrally in the disc while its inner ring is mounted on an eccentric terminal pin **24** on the shaft **8.**

On account of the eccentric pin **24**, the center of the disc **7** is held on a line that does not coincide with the geometrical axis of the drive shaft **2**, and due to the guiding by the bearing **21** sliding on the coaxial cylinder **22**, the disc will consequently assume an inclined position. The effect is thus identical with the one in FIG. **1.** The sliding bearing **21** is designed with a transversely curved inner surface so that it will only contact cylinder 22 in a narrow area and will, by an adjustment movement, perform a helical movement on the cylinder so that a significant reduction of the friction against the displacement movement is obtained.

The embodiment shown in FIG. 13 likewise reminds one of the embodiment shown in FIG. 1, but in FIG. 13 the inclined adjustment shaft 8 is stationary so that it cannot be rotated nor displaced.

One end portion of the adjustment shaft 8 is supported in a central bearing 25 in impeller 1 immediately at the end of the drive shaft 2. Besides, the shaft 8 is supported at its opposite end by the impeller in a bearing 26 which also is centrally mounted in the impeller, but fits on the shaft 8 with an eccentric bore or sleeve. The disc has a hub 27 provided with two roller bearings 28 on a sleeve that can be displaced on the adjustment shaft 8 with the aid of means, not shown, operated by pressure fluid admitted through a passage way in the adjustment shaft 8.

The embodiment shown in FIG. 14 fully corresponds to the one shown in FIG. 1, except for the support of the displaceable and rotating adjustment shaft 8.

The shaft is supported in a central slide bearing 29 fitted in a bore in the end of the drive shaft 2.

At the opposite side the impeller carries a centrally mounted bearing 30 permitting the impeller to rotate on a rigidly mounted sleeve 31, in which the shaft 8 is supported in a bearing 32 which is fitted in an eccentric bore in this sleeve 31.

It is remarked that the inclined position in most of the figures has been drawn in a somewhat exaggerated fashion, particularly when it is a question of impellers that have a large diameter, since generally the vanes should not, unless desired for special reasons, perform any movement in the course of a revolution when no adjustment movement is applied, that is to say when no adjustment is made the play in the connections and the elastic compression of the materials should be sufficient to allow the relative movement between the adjustment means and the impeller during the rotation so as to compensate for the inclined position.

Moreover, it will be apparent from FIG. 1 as well as from other figures that the same result can be achieved by securing disc 7 and permitting it to perform only an axial movement when the adjustment is to take place. If this is the case, the pins 5 circulate through the groove 6 and, consequently, provision should be made for the necessary reduction of the friction, e.g., the employment of rollers.

FIG. 15 shows an embodiment of a two stage fan comprising two impellers 33 and 34 secured to a common short and hollow shaft 35.

A drive shaft (not shown) is connected to the impeller 33 which, via a shaft 35, drives the other impeller 34 in a manner that is commonly known.

In its principal features the adjustment mechanism corresponds to the one shown in FIG. 1. The displaceable and rotatable adjustment shaft 8 is centrally journalled in impeller 34 in a bearing 36 mounted in the front end of the hollow shaft 35, and is further journalled in a stationary bearing 37 mounted outside the impeller and off-set from the geometrical axis of the shaft 35 so that the shaft 8 is inclined.

The shaft 8 extends half the length of the hollow shaft 35 where, by means of a universal 38, it is connected to an additional shaft 39 which, symmetrically with

shaft 8, is journalled in a bearing 40 in the opposite end of the hollow shaft 35, so that the shaft 39 forms the same angle to the axis of the shaft 35 as does the shaft 8, but is oppositely inclined. The shafts 8 and 39, together with the connection via the universal 38, thus constitute a common adjustment shaft for the two impellers.

From the foregoing detailed explanation of the function it will be understood that for obtaining the best possible result the size of the angle between the drive shaft and the adjustment shaft, or of their transverse distance in the case of an embodiment analogous to that of FIG. 9 should depend not only on the size of the impeller and the number of vanes, but also on the elastic deformation due to the action between the adjustment member and the eccentrically mounted elements. This deformation is very slight and can consequently vary substantially due to incidental dimensional variations within the production tolerances normally employed. For this reason it will be expedient for the angle or the transverse distance to be adjustable which may be effected in any suitable manner. For example, in an embodiment using an eccentric sleeve, this sleeve may comprise two eccentric sleeves which can be rotated relatively to each other in order to provide for the adjustment.

What we claim is:

1. An axial flow fan having impeller vanes which are adjustable during operation, said fan comprising:
- an impeller hub, rotatable about a first axis;
- a plurality of vanes supported in rotational symmetry by said hub, said vane having a central vane shaft which is journalled rotatably in the hub to adjust the vane to different pitch settings and a pitch setting member mounted excentrically relative to the vane shaft on the end of the vane adjacent to the hub, and
- a common adjustment member coupled to said hub for rotation together therewith, said adjustment member being rotatable about a second axis displaced relative to said first axis and being provided with means adapted to continuously cyclically engage and disengage said eccentric pitch setting members on all vanes so that the adjustment member bears against each individual pitch setting member during a part only of each revolution of the hub and the adjustment member, said adjustment member being furthermore mounted to be adjustable relative to the pitch setting members in opposite adjustment directions so that the adjustment member during an adjustment movement exerts a sufficient force on each pitch setting member to cause the vane associated therewith to turn relative to the hub during a fraction only of the part of said revolution during which the adjustment member bears against said pitch setting member,

whereby at a given instant during an adjustment movement a number of vanes between one and the total number of vanes minus one are subjected to the adjustment action.

2. An axial flow as claimed in claim 1 wherein said second axis is inclined at a small angle relative to said first axis.

3. An axial flow fan as claimed in claim 1, wherein the engaging means of said adjustment member have contact points for contacting opposite sides of the pitch setting members and the adjustment member is

mounted in such a way that in case of no adjustment movement the distance between one contact point and the side of its associated pitch setting member contacted by said contact point when adjusting in one direction is at any instant during rotation equal to the distance between a diametrically opposed contact point and the side of the diametrically opposed pitch setting member contacted by said latter contact point when adjusting in the opposite direction.

4. An axial flow fan as claimed in claim **3,** wherein said contact points are situated on two substantially parallel contact surfaces of the adjustment member separated such a distance from each other that for each revolution two diametrically opposed contact points situated on one and the other contact surface come into contact with their associated pitch setting members even in case of no adjustment movement.

5. An axial flow fan as claimed in claim **1** further comprising means for adjusting the relative positions of the adjustment member and the pitch setting members to compensate for incidental dimensional variations within normally employed production tolerances.

6. An axial flow fan as claimed in claim **1,** in which the adjustment movement of said adjustment member is an angular displacement around its axis, characterized in that said second axis is parallel to said first axis and is located a short transverse distance from it.

7. An axial flow fan as claimed in claim **1,** in which the adjustment member is displaceable and rotates with the impeller, characterized in that the adjustment member is rotatably and displaceably supported on a stationary shaft that is guided in two bearings mounted in the hub, at least one of these bearings being mounted eccentrically in relation to the drive shaft of the impeller.

8. An axial flow fan as claimed in claim **1,** in which the adjustment member rotates with the impeller and is connected to a displaceable adjustment shaft, characterized in that the adjustment shaft is supported in a slide bearing at the side of the hub remote from the drive shaft and on line with the drive shaft, and is secured against rotating, and that its end inside the hub comprises an eccentric pin on which the adjustment member is journalled by a spherical bearing, the adjustment member being further journalled on a sleeve rigidly connected to the impeller hub and coaxially mounted by means of a slide bearing having a curved internal bearing surface, the diameter of which decreases from both ends of the bearing and in an inward direction.

9. An axial flow fan as claimed in claim **1,** in which the adjustment member rotates with the impeller and is displaceable by means of an adjustment shaft, characterized in that the adjustment member is rigidly connected to the adjustment shaft which is supported in two bearings in the hub, of which one is mounted centrally in relation to the drive shaft while the other is mounted opposite to the drive shaft and comprises an inner and an outer bearing, of which the outer bearing is mounted centrally in the hub and is connected to the inner bearing by means of a stationary sleeve having an eccentric bore.

10. An axial flow fan as claimed in claim **1,** comprising two impellers mounted on a common hollow shaft, the adjustment member for both impellers being rigidly mounted on an adjustment shaft, characterized in that the adjustment shaft comprises two portions connected by means of a universal located between the two impellers and inside the hollow shaft, and that each impeller comprises a centrally mounted bearing, one portion of the shaft being journalled in one of these bearings only, while the other portion of the shaft is journalled in the other bearing as well as in an eccentrically mounted bearing.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,790,301 Dated February 5, 1974

Inventor(s) Henry Valdemar Pedersen et al.

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

In the drawings, cancel sheet 1 containing Figs. 1 and 2, and substitute the attached sheet.

Signed and sealed this 10th day of September 1974.

(SEAL)
Attest:

McCOY M. GIBSON, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents

FORM PO-1050 (10-69)

USCOMM-DC 60376-P69
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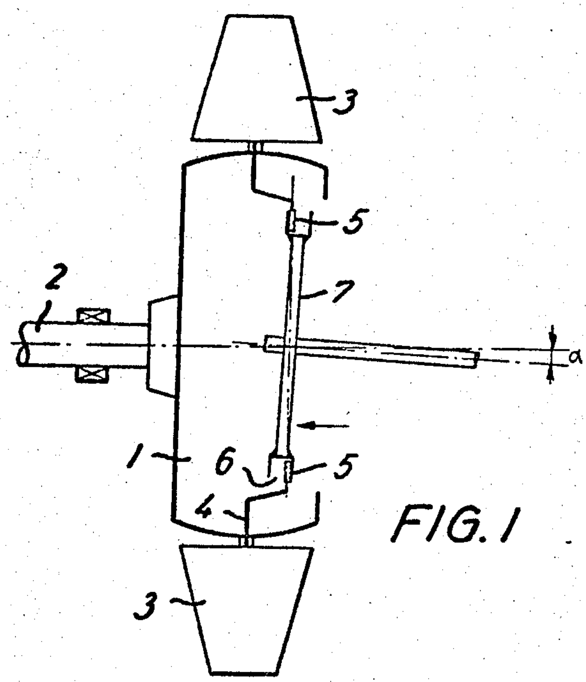

FIG. 1

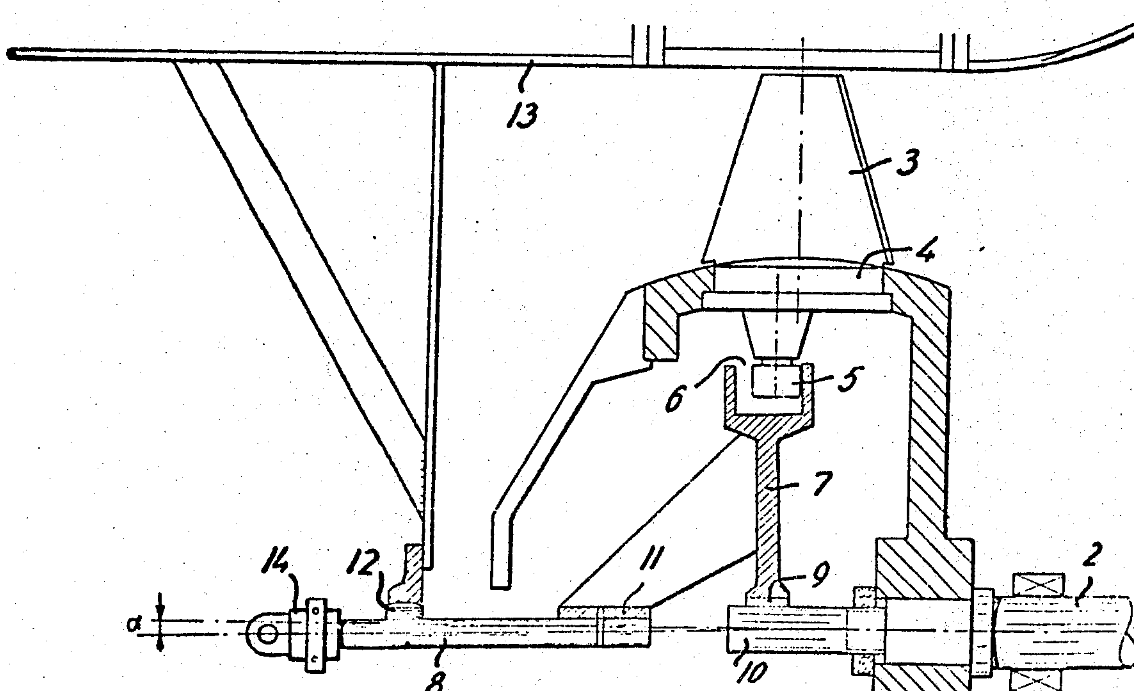

FIG. 2