Oct. 18, 1960 G. R. TOTHILL 2,956,739

VARIABLE PITCH IMPELLERS AND CLOSURE SEALS THEREFOR

Filed Feb. 13, 1957 5 Sheets-Sheet 1

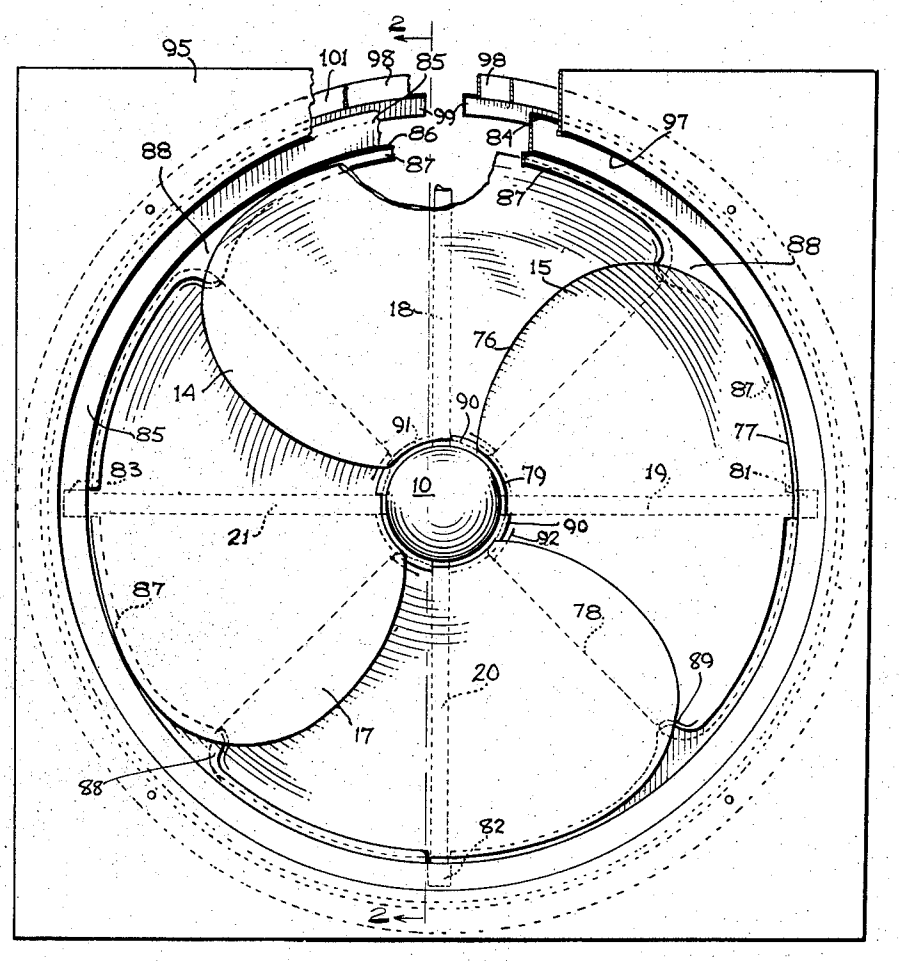


Fig. 1.

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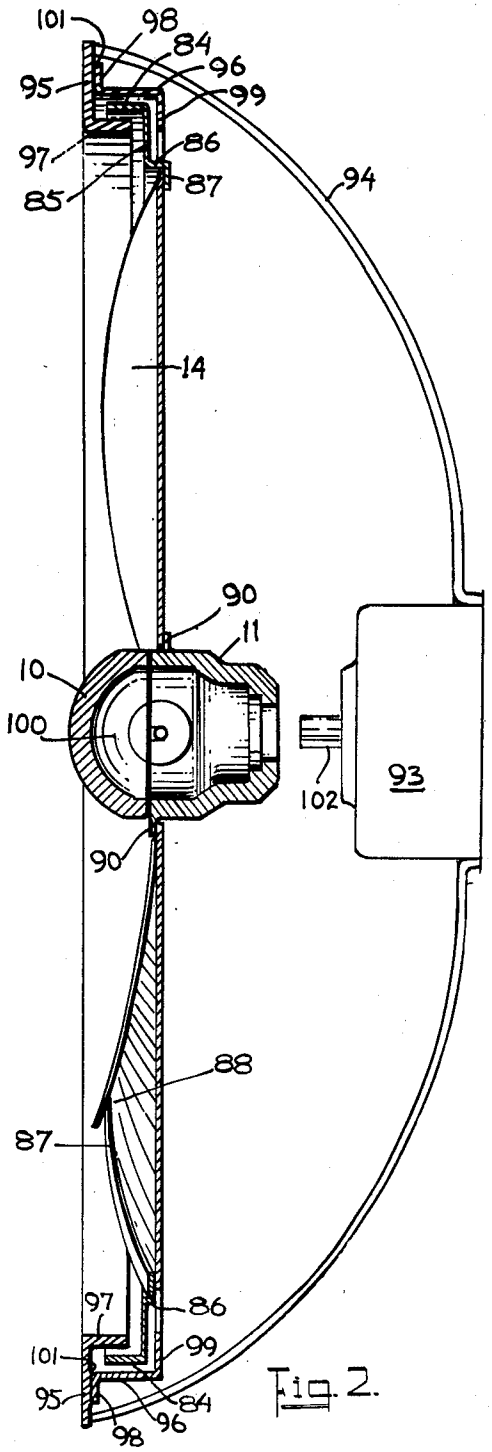


Fig. 2.

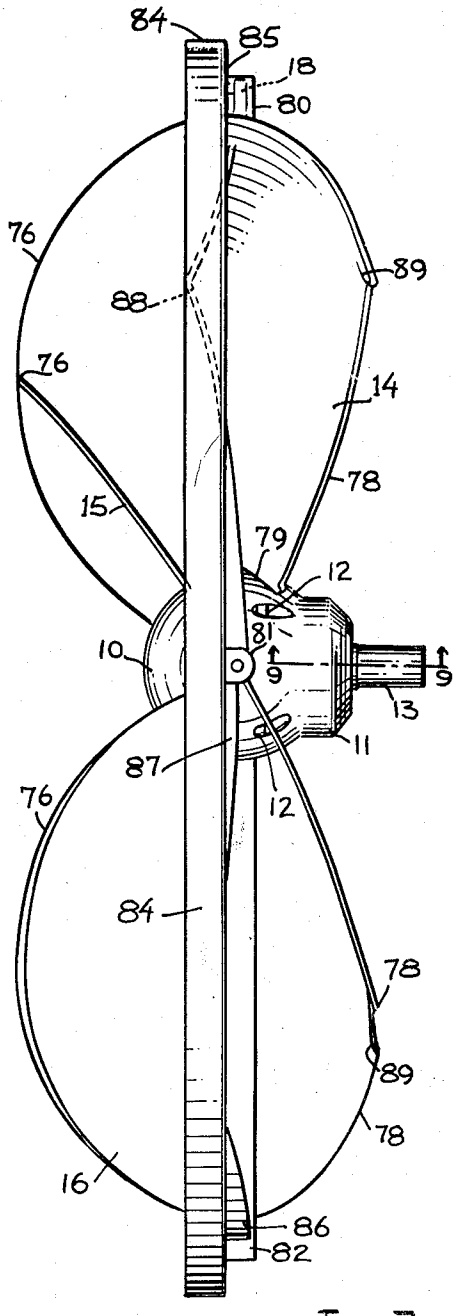


Fig. 3.

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VARIABLE PITCH IMPELLERS AND CLOSURE SEALS THEREFOR

Filed Feb. 13, 1957 5 Sheets-Sheet 3

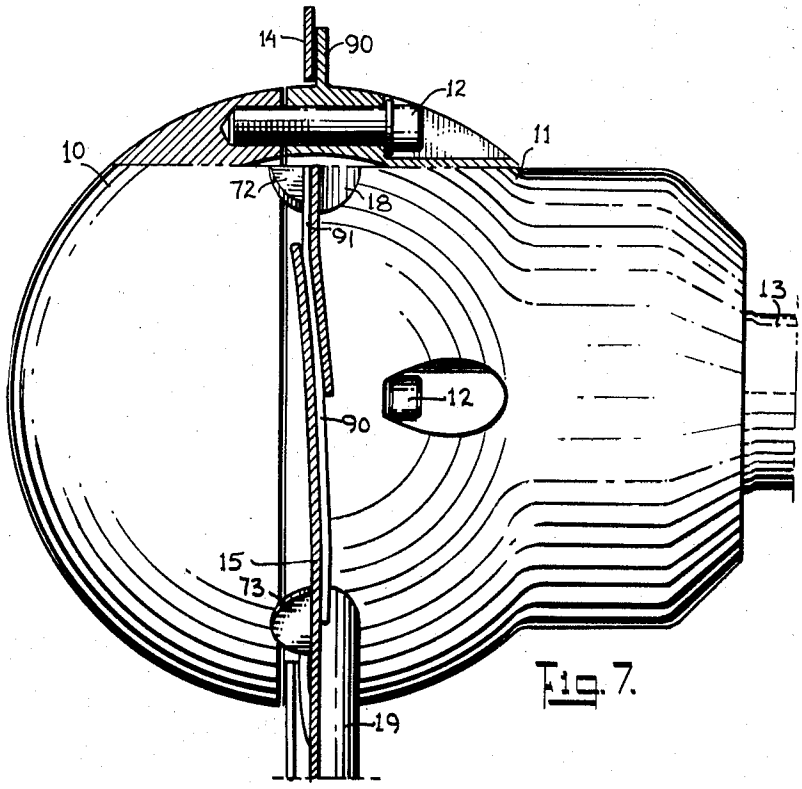


Fig. 7.

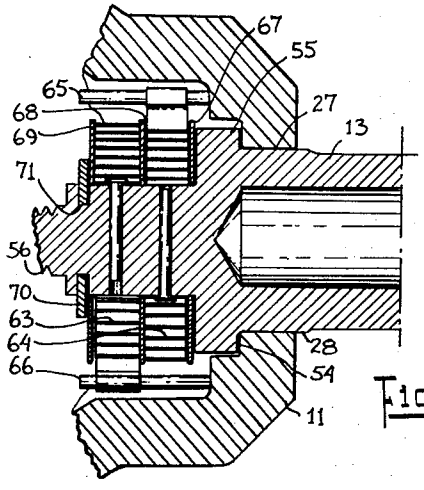


Fig. 9.

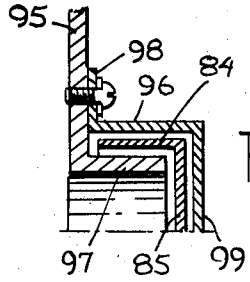


Fig. 4.

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VARIABLE PITCH IMPELLERS AND CLOSURE SEALS THEREFOR

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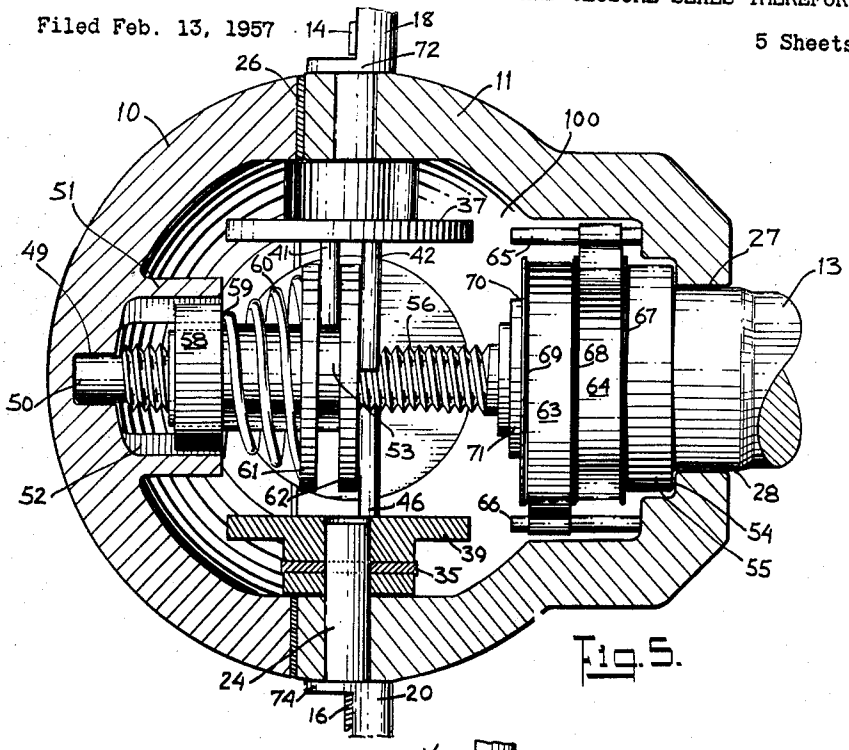
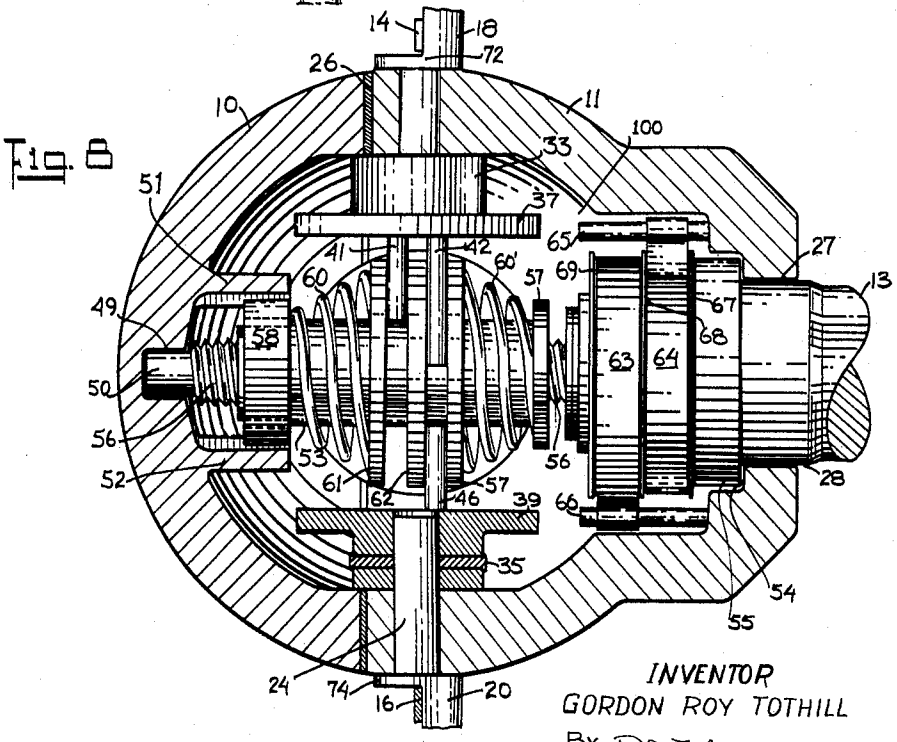


Fig. 5.

Fig. 8



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VARIABLE PITCH IMPELLERS AND CLOSURE SEALS THEREFOR

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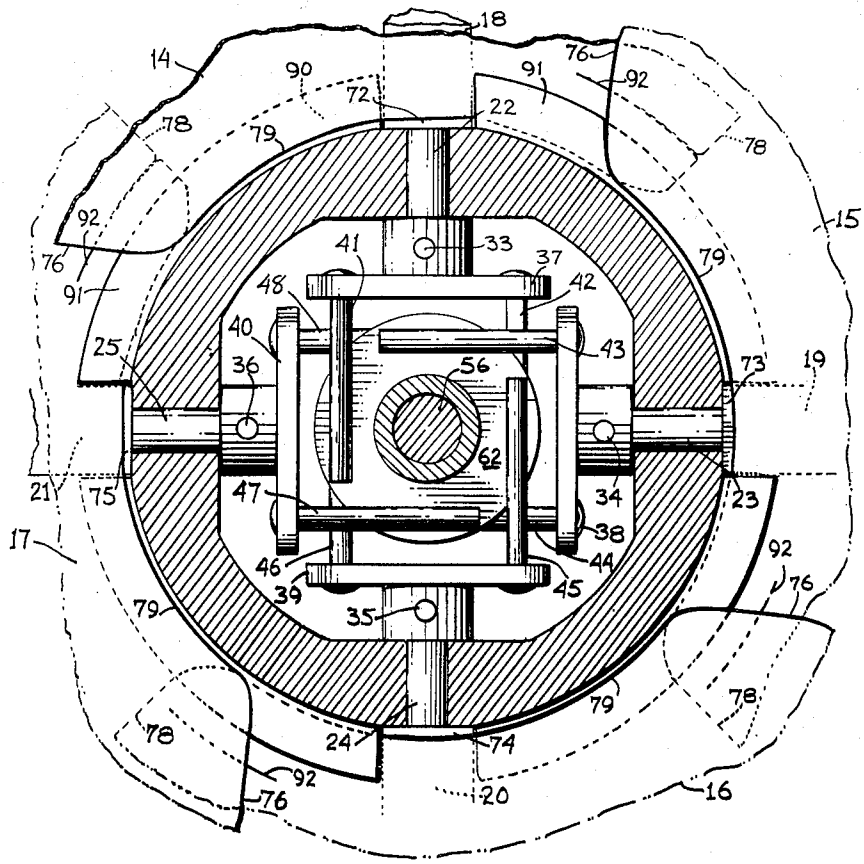


Fig. 6.

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2,956,739

VARIABLE PITCH IMPELLERS AND CLOSURE SEALS THEREFOR

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Filed Feb. 13, 1957, Ser. No. 639,879

6 Claims. (Cl. 230—270)

This invention relates to variable-pitch impellers and blade sealing arrangements, and more especially relates to impellers or propellers whose blade pitch is automatically varied in accordance with the idle or rotating condition of the impeller.

According to the invention an impeller system may be realized for use in association with an apertured housing, wherein the blades themselves form shutters or closures for the aperture when the blades are not rotating, and wherein automatic pitch adjustment means respond when the blades are rotated to open the blading.

It is a primary object of the invention to provide an impeller system such that the blades assume a minimum or reference pitch when stationary, and when rotated by a powered means have an increase in pitch automatically effected to a predetermined maximum, which maximum pitch is held in accordance with angular acceleration forces and aerodynamic loading of the blading.

Another object of the invention is to provide an apertured enclosure and multi-bladed impeller or propeller coaxial therewith and of such form that in the minimum pitch condition of the blades the flow of gas or fluid through the aperture is prevented, without the use of check valves or shutters.

A further object is to provide a simple, light, compact and durable pitch changing mechanism for a system of blading which is economical of manufacture and easy of assembly.

Yet another object is to provide a sturdy blade pitch adjusting mechanism, characterized by a large mechanical advantage, adapted to be entirely housed within an impeller or propeller hub of relatively small diameter, and characterized further by a minimum of working parts.

Still another object is to provide a pitch-adjusting mechanism enclosed within a hub body which gradually effects an opening of blades journalling in the body from a closed position on starting and which draws no power from the driving source when the impeller or propeller has reached an equilibrium speed of rotation.

A further major object is the provision of a pitch-adjusting mechanism capable of applying a large turning moment to each blade and inherently capable of taking up frictional wear in the coupling to the blades so that all blades of a set are maintained at equal pitch values.

Still another object is the provision of a pitch-adjusting mechanism which is independent of rotational direction applied by the drive means so that starting up from rest position in either a clockwise or counterclockwise direction is effective to increase the blade pitch from minimum to maximum.

A further object is the provision of a simple, lightweight and efficient impeller blading system and an associated enclosure which requires no external agency or mechanism for preventing the forward or backward flow of gas or fluid therethrough in the stationary blade condition, and which requires no external control to effect a sealing action in the rest condition.

With these and other objects in view as well as other

advantages which will become apparent in a discussion of the improved construction, the invention consists in several novel features and combinations thereof set forth in the appended claims, wherein it will be understood that the several constructional elements constituting the invention may be varied in proportion and arrangement without departing from the nature and scope of the appended claims.

To enable others skilled in the art to comprehend the underlying features of this invention that they may embody the same by suitable modifications of structure and relation of parts to effect the various practical applications contemplated for the invention, the following description is to be read in conjunction with drawings showing a preferred embodiment of the invention, namely:

Fig. 1 is a front view of the impeller and enclosure showing the general arrangement of the embodiment as applied to an exhaust or ventilating fan;

Fig. 2 is a sectional view of the impeller and enclosure taken on the line **2—2** in Fig. 1 and omitting the pitch-changing mechanism;

Fig. 3 is a side view of the impeller as illustrated in Fig. 1 with the blades in the open or maximum pitch position showing curved blade seal surfaces on the enclosure;

Fig. 4 is a detailed enlargement of the sectional view through the peripheral seal assembly of Fig. 3;

Fig. 5 is a partial sectional view of the spinner or hub illustrating the pitch change mechanism;

Fig. 6 is a partial sectional view of a spinner substantially on the line **3—3** of Fig. 5, showing the detail of the arrangement of blade rotating elements and axial screw actuator;

Fig. 7 is a partial sectional view of a spinner showing the method of assembling and securing the parts of the spinner cap and base;

Fig. 8 is a sectional view similar to Fig. 5 corresponding to another embodiment of the invention;

Fig. 9 is a cross-section taken on line **9—9** of Fig. 3 showing the impeller shaft and coil springs mounted thereon.

General description of hub

Referring to the figures of drawing, a practical embodiment of the invention generally comprises an assembly of a shaft adapted to be coupled to a drive motor, a hub coaxial with and enclosing blade pitch-adjusting mechanism and journalled on the shaft, a set of blading supported rotatably in the hub having their length and pivot axes substantially perpendicular to the shaft, and an enclosure associated with a motor housing coaxially mounted with respect to the shaft having sealing flanges co-operating with a seal ring carried at the outer margins of the blades.

Having regard first to Figures 1, 2, and 3 of the drawing, a propeller or impeller hub is realized as a hollow, somewhat pear-shaped housing which is preferably a body of revolution comprising a forward spherical shell portion **10** assembled with a larger rear hub portion **11**, as by means of paraxial screw devices **12**, to enclose a space generally designated as **100**. A compression sealing gasket **26** is interposed between the abutting faces of the two shell portions of the hub to provide a dust-proof and oil-retaining protective enclosure for mechanism when the parts are assembled. The hub surrounds the end portion of shaft **13** and is captively supported thereon in coaxial relation. Coaxially of the hub body **11** a bore **27** is provided wherein a cylindric bearing surface **28** of the shaft **13** is journalled. A blind bore **49** is formed in the forward hub shell **10** coaxially therewith and opening inwardly to journal the forward end **50** of the shaft **13** therein. Supported about the periphery of the hub portion **11** are a set of propeller or impeller blades, these being designated **14**, **15**, **16**, and **17** in the

case of the four-blade assembly depicted. The blades are mounted upon respective shafts **18**, **19**, **20**, and **21** disposed substantially at right angles to the axis of the shaft **13**, and including respective reduced diameter portions **22**, **23**, **24**, and **25** adjacent their inner ends. These reduced diameter portions as best shown in Fig. 6, are journalled in bores extending through the spherically shaped part of shell **11** and equiangularly spaced about the periphery thereof. Supported upon and secured to the terminal portions of the blade pivot axles just described are the respective collars **29**, **30**, **31** and **32** which are restrained against axial movement, as by use of grubscrew or similar fastening devices **33**, **34**, **35**, and **36**. The collars include lateral extensions as for example respective integral flange portions **37**, **38**, **39**, and **40**, each collar carrying its respective pair of pins of the sets of pin pairs **41**—**42**, **43**—**44**, **45**—**46**, and **47**—**48**. These pins are affixed to extend inwardly of their mounting collar in a direction substantially parallel with the pivot axis of the blade shaft associated with that collar, and the pins are moreover spaced equidistantly in opposite directions along a line passing through the blade pivot axis.

The outermost faces of the collars **29** to **32** inclusive are suitably shaped to conform with the inside of the rear shell **11**; in a preferred embodiment the inside faces of the shells are machined smooth in the region surrounding each bore to provide flat bearing surfaces adequate to withstand centrifugal forces when the impeller is rotating. Alternatively the inside of the shells in the vicinity of each bore in which the shafts **22**, **23**, **24**, and **25** are journalled are made spherical and the outer end of each of the aforesaid collars is convexly spherical.

The inner face of the forward shell **10** carries axially protruding posts or shaped guides **51** and **52**, preferably formed integrally with the shell, and spaced laterally from the bore **49**. As will be described later, these posts serve as ways or guides to prevent rotation of a carrier **53** forming part of the pitch-adjusting mechanism presently to be explained.

Alternatively the guide may be realized as a cylindric boss having either internal or external splines or equivalent forms designed to provide for axial freedom of movement for a guided body while restraining relative rotation therebetween. In one embodiment such boss has a squared shoulder against which the restoring spring may be seated.

The rear shell **11** is machined to provide a stepped bore including the axial bearing **27** and an annular thrust bearing disc face **54** concentric therewith on the inner side of the shell.

Blade pitch adjusting mechanism

Primary shaft **13** which is preferably machined from a single length of rod of steel or other suitable material, is formed with a collar **55** adjacent a bearing surface **28**, the collar serving as a thrust bearing member in association with the disc face **54** to prevent outward axial movement of the shaft. The outer end of the shaft is of reduced diameter and is finished to provide an axle **50** journalling in the end bearing **49** by which it is also restrained against forward axial movement. Between the small end and the collar the shaft has a portion **56** formed of intermediate uniform diameter, this portion being threaded concentrically with the shaft axis. The thread may be realized in any thread form and in any suitable pitch, preferably as a fine pitch thread of close tolerance. In applications requiring a large diameter threaded portion it may be advisable to use a self-centering thread; however in the majority of applications as for example in space venting fans the thread may preferably be National Fine machine thread. In general as the number of threads per inch is increased the mechanical advantage of the pitch-adjusting mechanism is proportionately increased with the result that the torque effect for a given loading of the impeller system effective to alter the blade pitch is increased. In some applications a coarse pitch will be found preferable to obtain rapid and large changes in blade pitch.

A body **53**, hereinafter referred to as the actuator nut, has an axial bore threaded to mate with the thread of the shaft portion **56**, upon which it rides. Sufficient clearance is allowed for smooth engagement of the threaded parts with low friction and a minimum of axial play. An outer end of the actuator nut has an integral flange **58** extending radially of the nut, the flange being milled axially to engage the ways **51**, **52** in the end shell **10** of the hub. An inner shoulder **59** formed on the flange serves as a support for restraint of the outer end of compression spring **60** which is coaxially disposed on the nut **53**. The other end of the spring rests against an apertured flanged disc **61** which is freely slidable on the cylindric portion of nut **53**. The inner end of the nut is formed with a circular integral disc flange **62**. Pressure of the spring **60** against disc **61** forces the latter against pins **41**, **43**, **45** and **47**, thereby developing a torque tending to rotate each blade shaft into a rest position in which the said pins are pressed between the discs **61** and **62**. At this rest position the respective pin pairs such as **41** and **42** on collar **37**, lie in contact with the front and the back face of the flange **62**.

While a conical spring **60** is shown in the drawing, to provide for a shorter fully compressed spring length, the use of straight springs is envisaged in those applications where the relative compression of the spring need not be large. Between the shaft flange **55** and the reduced diameter portion **56** there are mounted two opposed flat power springs **63** and **64** whose inner ends are suitably secured to the shaft and whose outer ends are captive on posts **65**, **66** secured to the inside of shell **11**. The coils are restrained and guided by washers **67**, **68**, and **69**, and by spring retainer **70** which engages a groove **71** in the shaft. Tensioning of the coils is arranged to hold the shaft **13** in a rest position at which pin pairs such as **41** and **42** are in contact against opposite faces of flange **62**.

Blading and seals

As has previously been set forth the impeller blading is affixed to shafts **18**, **19**, **20** and **21**. As will be most directly apparent from Figure 8, the shafts, which may conveniently be realized from rod stock, are cut away to form approximate semi-cylinders extending outwardly from the base portions **72**, **73**, **74** and **75**, which are left in their original rod diameter size to prevent axial play of the shafts. The blades are realized preferably as rigid bodies of sheet metal or other material having suitable characteristics and in the case of metal may be affixed by rivets or by spot-welding to the flat faces formed on the shafts.

Referring additionally to Fig. 1, the outline of each blade as viewed from the front is seen to include a leading edge **76** which rounds smoothly into the circular outer edge **77**, a trailing edge **78** lying substantially along a radius, and a circular inner edge **79** conforming to the periphery of the hub. In the drawings Figs. 1, 2 and 3 the rotation of the blade assembly is anticlockwise and the rotation of each individual blade is such that leading edges move toward the front or viewing position while the trailing edges swing away oppositely, with increasing blade pitch.

The forming and arrangement of the impeller blades is such as to permit the trailing edge of each blade to lie in close contact with the back surface of an adjacent blade, the line of contact extending substantially the full length of the blade. The contact line will be observed to lie at the leading edge of the overlapped blade. The exact shape and number of the blades may be varied considerably; as herein described the blades are essentially concave to the front but are uncurved along any radius through the hub axis, and the leading edge portion over-

of the hub but rests against the inner face of flange **58** of the nut **53**. In the centered rest position as illustrated, pin **42** is gripped between integral flange **62** and the slidable disc **61'** under pressure developed by axial compression of spring **60'**. Similarly pin **41** is gripped between the opposite face of flange **62** and slidable discs **61** under the pressure of coaxially compressed spring **60**. Since the pins **41** and **42** are therefore held in close contact against flange **62**, as pertains to all other pin pairs also, the respective blades are firmly maintained in their reference shuttering position.

If due to relative rotation of shaft **13** with respect to the nut **53** the flange **62** is shifted to move pin **42** to the right, the pin **41** will experience a rotatory displacement to the left causing spring **60** to be compressed further, the deflection in the Fig. 5 embodiment being twice the displacement of flange **62**. In the embodiment of Fig. 8 it will be obvious that the spring **60** will be compressed equally with spring **60'**. At the same time, due to the relative rotation of shaft **13** as has been indicated, coil spring **64** will wind up and spring **63** will unwind an equal and opposite amount. There will consequently be a torque on hub part **11** transferred by the pull of spring **64** on post **65**, tending to spin the hub and assembly to follow the shaft rotation.

Dynamic operation of blade-adjusting mechanism

Referring particularly to Figs. 1, 5, 6, and 8, the operation of the mechanism contained within hub space **100** will be discussed in the following description. Let it be assumed that there is applied to shaft **13** a torque tending to rotate the blading anticlockwise as viewed in Fig. 1. For this direction of rotation the blading if opened would tend to urge air through the aperture in housing **95** from the front to the rear. Inasmuch as the usual drive motor means **93** accelerate from rest condition to driving speed in a relatively short interval of time, the inertia of the hub and blade assembly will resist rapid increase in rotational velocity. Accordingly, flat spring **64** will wind up as shaft **13** rotates relatively to the hub and spring **63** will unwind. At the same time the relative rotation of threaded shaft portion **56** within actuator nut **53** will cause the latter to be drawn axially to the right in Fig. 5 away from shell portion **10**, moving pins **42, 44, 46** and **48** to the right under pressure of flange **62**. The rotation of collars **37, 38, 39** and **40** will be opposed by the reciprocal opposite displacement of pins **41, 43, 45** and **47** which compress spring **60** by forcing disc **61** to the left. It will be apparent that the extent of relative rotation of the shaft **13** with respect to the hub assembly is limited by the loading due to winding of coil spring **64** and the compression of spring **60**. After a certain angular relative rotation has occurred the coupling between the hub and shaft **13** will be relatively rigid and the hub takes on the rotational speed of the shaft, with the blading opened to its greatest pitch. As soon as the driving source has reached its equilibrium speed due to the load, the loading due to acceleration of the hub assembly substantially disappears, with the result that a slight decrease in blade pitch will possibly occur due to unwinding of flat spring **64** and a winding up of spring **63**. At the same time axial decompression of spring **60** in Fig. 5, or of springs **60** and **60'** in Fig. 8, brought about by the decrease of blade pitch, reduces the loading. The resistance or drag of the moving blading against the air stream developed and the reaction forces due to accelerating the stream remain effective to hold the blading open at an equilibrium pitch setting.

The extent of drag forces on the blading and the torque effect tending to hold the blades open will in general be determined by the airfoil characteristics of the blade forms chosen. Where the center of pressure lies between the leading edge and the blade pivot shaft over the working range of blade pitch angles there will be a couple tending to increase the plate pitch.

With conventional drive motors of shaded pole induction type or series wound brush type their equilibrium rotational velocity under variable loading is generally related inversely with the magnitude of the loading; consequently where such drive is employed with the blade pitch-adjusting mechanism according to the invention fluctuations in approach velocity of the air stream tending to decrease the loading on the blades will co-operate with the regulation characteristic of the motor to maintain a relatively steady fan spin rate.

The embodiments in Figs. 5 and 8 may be used in those applications where it is desired to move air or gas in either direction through the aperture in housing **95**; for example, where the flow is desired from the rear or motor side towards the front the direction of motor drive will be reversed by means not forming any part of this invention, causing the blading to spin clockwise in Fig. 1 view. Under this condition the flange **62** compresses spring **60** as it urges pins **41, 43, 45** and **47** to the left against disc **61**, and flat spring **63** winds up to an equilibrium coupling condition, while the blade pitch increases as described before.

If the direction of drive is again reversed while the blade assembly is spinning to drive air from the rear to the front, the actuator nut **53** will be rapidly axially moved through the zero blade pitch condition, thereby arresting the flow of air, and the same blade pitch angle as before will be quickly re-established as the coupling between shaft **13** and the hub assembly becomes more firm.

In some applications it may be acceptable to permit shaft **13** to rotate a large number of revolutions before one of the spiral springs **63** or **64** is firmly wound up to provide a rigid coupling. Where the actuator nut **53** may travel a considerable amount, sufficient leeway must be provided between the end of the shaft thread **56** and the nut travel limit to prevent jamming, or a suitable cushion or stop may be provided to arrest the nut.

I claim:

1. A variable blade pitch impeller comprising a drive shaft having a threaded portion, a hollow hub enclosing said threaded portion and journalling said drive shaft therein, a plurality of impeller blades having radial blade support shafts journalled in said hub to turn about respective blade shaft axes, collar means fixed to each blade shaft within the hub each carrying a pair of pins extending inwardly parallel with respective blade shaft axes and spaced oppositely from said axes, a flanged actuator nut threadedly engaging said drive shaft reciprocable in guide means in said hub preventing relative rotation of said nut, said flange extending between respective pins of each pin pair, spring means biasing one pin of each pin pair against said flange, and resilient means coupling said hub with said drive shaft whereby when said hub and said drive shaft are disposed in predetermined angular relation said flange is in contact with both pins of each pin pair to hold said blades in a reference pitch setting.

2. A variable blade pitch impeller as in claim 1 wherein said resilient means comprise a pair of opposed spiral torsion springs coaxial with said drive shaft and each having one end fixed thereto and its other end fixed to the hub, said springs being wound in opposite senses.

3. An impeller organization comprising a drive shaft having a length portion threaded adjacent one end, a hollow hub journalled for rotation upon the shaft to enclose said portion, a plurality of blades having blade shafts radially arranged about the hub and having their inner ends journalled for rotation in said hub about axes lying in median planes transverse to said blades, a pair of crank pins extending within said hub and secured to the inner end of each blade shaft, said crank pins being spaced equally on opposite sides of a blade axis, torsion spring means yieldably coupling said hub with said drive shaft for rotation together in either direction and to establish a reference angular relation therebetween said hub and

lapped by an adjacent blade is curved forwardly outwardly while the trailing edge is curved similarly forwardly. It is to be understood that a blade may alternatively be formed on dies conforming to the surface of a cylinder, of which the generating line moves about an axis normal to and passing through the drive shaft axis.

The outer ends of the blade shafts are journalled in respective cups or sockets 80, 81, 82 and 83, secured in any suitable manner to the back surface of an annular disc portion 85 which lies radially inwardly of a carrier ring 84 surrounding the blading and forming part of a rotative seal. Lying radially inwardly of the disc 85 is a discontinuous cylindric band 86 whose axial length varies around the circumference and which extends in certain regions more to the forward side of disc 85 than on others. A radially inwardly directed blade-edge seating flange generally designated 87 is integral with band 86 and is curved from the plane of disc 85 to conform with the curvature of blade edges 77. In Figure 2 the flange areas 88 are displaced a maximum distance forwardly from the disc 85 while at the discontinuities of the band 86 in the immediate proximity of sockets 80, 81, 82, and 83 the width of band 86 and the displacement of the seal flange 87 are approximately equal to the blade material thickness. In general, it will be a design choice to axially shift the relation of flange 87 with respect to disc flange 85, for example to dispose areas 88 and the socket areas equally and oppositely from flange 85.

Flange 87 is preferably formed by a forming die shaped between areas 88 to conform to a surface of a cylinder whose axis is normal to the axis of shaft 13 and disposed forwardly of the hub. The outer margins 77 of the blades are therefore preferably formed with a slight forward arcuate bulge conforming to the overlap of a rearwardly disposed area of flange 87.

Each blade lies in its rest position with that part of its outer edge which extends between the support shaft and the leading edge lying in contact with the forward side of flange 87, while the remainder of the outer edge lies in contact with the rear side of the flange 87. In the vicinity of the junction of the trailing and outer edges of each blade, a depressed tab 89 is formed integrally with the blade having a maximum depth measured at the trailing edge equal to the thickness of the flange area 88. The depth tapers to zero along the shoulder which forms the boundary between the tab and the unformed body of the blade. The contour of this shoulder substantially follows the outline of the flange area 88, and is preferably produced by a forming die, or alternatively by partial shearing in a die. The front side of the tab lies in close contact with the rear face of the flange area 88 to provide an effective seal, while the rear side of the blade overlaps the flange area 88 to complete the peripheral seal.

As previously indicated the radial seal between adjacent blades is along a line of contact which extends to the shoulder of the tab 89. Any leakage at the outer edges of the blading is therefore limited to the relatively narrow gap between the shoulder of tab 89 and the flange outline at the area 88. Such leakage may be reduced to a very low value by careful design and precision.

At the inner edges of each blade a somewhat similar peripheral seal provision is made, which is best understood by reference to Fig. 7. A discontinuous peripheral flange comprising segments 90, 91 is formed integral with and disposed about the hub 11. That part of the inner edge 79 of each blade which extends between the shaft and the blade's leading edge lies to the front of and in contact with the seal segment 90. The remainder of the inner edge lies to the rear of and in sealing contact with the seal segment 91. The segmental lengths are so arranged that only sufficient clearance remains between the end of a segment and a blade support shaft 18, 19, 20 or 21, to allow for free shaft rotation. A relative axial displacement is provided between the end of segment 90 adjacent to an end of segment 91, parallel to the axis of shaft 13, equal to at least the blade thickness.

A depressed tab 92 is formed at the junction of leading edge 78 and inner edge 79 integrally with each blade, with a depth measured normal to the trailing edge substantially equal to the thickness of segment 91, and gradually tapering to zero along the shoulder of the depression, which conforms to the outline of segment 91. Each of the segments 90 and 91 may be, and preferably is, warped axially to protrude forwardly at its mid-length position, or more specifically is formed with a ridge lying in line with the blade trailing edge, to conform with blade curvatures.

Referring again to Fig. 2 the blade assembly including the ring 84 and annular disc element 85 will be seen to be located with respect to an enclosure 95 by a support frame 94 in which a motor 93 is mounted. The exact form of the frame is not critical and may comprise, for example, a mesh of wires or rods of adequate strength and rigidity secured to the enclosure and to the motor mounting. Motor 93 may be of any suitable type having a drive shaft 102 upon which fan shaft 13 may be coaxially fitted and secured in drive relation.

Ring 84 is disposed coaxially within an annular cavity formed between an inner flange 97 of the enclosure 95 and a formed ring 96 secured to the back of the enclosure. It will be apparent that a labyrinthal passage is thereby provided to restrict flow of air or fluid around the edge of the ring 84. Preferably the ring 96 is flanged reversely with a forward disc flange 98 extending radially outwardly and a rearward disc flange 99 directed parallelly with the disc 85. The radial length of the flange 99 is such that a clearance remains between the outer ends of sockets 80, 81, 82 and 83 and the inner margin of the disc 99. Between the flange 98 and the enclosure 95 a resilient gasket 101 may preferably be interposed at the time of assembly and the parts are secured together by any suitable means, for example by metal screws or bolts and nuts.

The seal ring 84 with its integral disc flange 85, cylindric flange 86, and blade edge seal 87, may preferably be realized by a stamping and drawing step employing sheet metal stock. Alternatively injection moulding or die-casting processes may be employed where a one-piece body is desired. Similarly, flange ring 96 may be stamped from light guage metal or may be realized by any other conventional process of adequate precision.

In applications where the flow of gas or fluid must be relatively completely prevented or where the head existing between opposite sides of the impeller is relatively large in the stationary closed-blade position, the fabrication of the various parts will require to be carried out with a greater regard to rigidity and precision of fits. The surfaces of the blading and all surfaces of the seal flanges may be coated with a resilient film of a soft rubber or plastisol, preferably by dipping or by spraying before assembly. In certain fluid pump and compressor applications the inner surface of the rotary ring 84 will be arranged to actually ride against the outer surface of the panel flange 97, to produce a sleeve bearing type of seal. Where an anti-friction large bore type of ball race bearing (not shown) is provided between the relatively rotating elements and carries its own seal, the flanged ring 96 may be eliminated.

An alternative embodiment is shown in Fig. 8 wherein means are provided for centering the actuator nut 53 more positively to achieve a relatively more rigid closed-impellor position. In the sectional view it will be apparent that the mechanism housed within the hub is similar to that of Fig. 5 with the primary difference that a second compression spring 60' is provided, retained between a further axially movable disc 61' and an end flange 57 on the actuator nut 53. It will also be noted that the outer end of spring 60 is not seated against any part

drive shaft are at rest, actuator nut means engaged with said threaded length portion guided in said hub for axial reciprocation when said hub over-runs said drive shaft in either direction, said nut having an integral disc flange extending radially between crank pins of each pair whereby to swivel said blades from substantially zero pitch setting corresponding to a reference nut position along said drive shaft when said hub and drive shaft are at rest to a working pitch setting when said drive shaft turns in said hub in either rotational sense, and means resiliently biasing corresponding crank pins of each pair to bear against one side of said flange.

4. An impeller organization as in claim 3 wherein opposite crank pins of each pair are resiliently biased in axial directions toward each other, and wherein said torsion spring means comprise a pair of spirally wound springs each having their one ends fixed to the drive shaft and their other ends fixed to said hub, said springs being wound in opposite senses.

5. In a variable pitch impeller organization the combination of a hollow hub, a drive shaft journalled for rotation in said hub and having a threaded portion enclosed therein, means yieldably coupling said hub with said shaft to establish a reference angular relation therebetween when at rest, a plurality of impeller blades having axles whereof the inner ends are swivelled in said hub about axes substantially perpendicular to and equiangularly spaced about said drive shaft, a pair of crank pins fixed to each of said inner ends, the pins of a pair being spaced equally and oppositely from the respective blade axes, an axially reciprocable rotationally restrained nut engaged with said threaded portion of said shaft for relative screw rotation thereon when said drive shaft over-runs said hub in either rotational sense, a coaxial disc flange carried by the nut disposed between crank pins of each of said pairs and occupying a reference axial position along said shaft corresponding to said rest

angular relation of said hub and drive shaft, and means biasing a corresponding pin of each pair against a side of said flange whereby to establish a zero pitch blade setting when said hub and drive shaft are at rest and whereby displacement of said nut in opposite axial directions from said reference axial position effects the swivelling of said blade axles in the same direction to increase the blade pitch.

6. A variable blade pitch impeller as in claim 5 wherein said yieldable coupling means comprise a pair of opposed torsion springs coaxial with said drive shaft each having their one ends joined with said drive shaft and their other ends joined with said hub, said springs being wound in opposite senses.

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