United States Patent

Klompas

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[54] **ROTOR WITH VARIABLE ANGLE BLADES**

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[73] Assignee: **General Electric Company**

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Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 868,753, Oct. 23, 1969, abandoned.

[52] **U.S. Cl.** **416/156,** 416/145, 416/207, 416/500

[51] **Int. Cl.**.....**F01d 7/02**

[58] **Field of Search**.....416/142, 147, 207, 208, 500, 416/152, 155, 156, 145, 205, 212; 415/129, 133, 149

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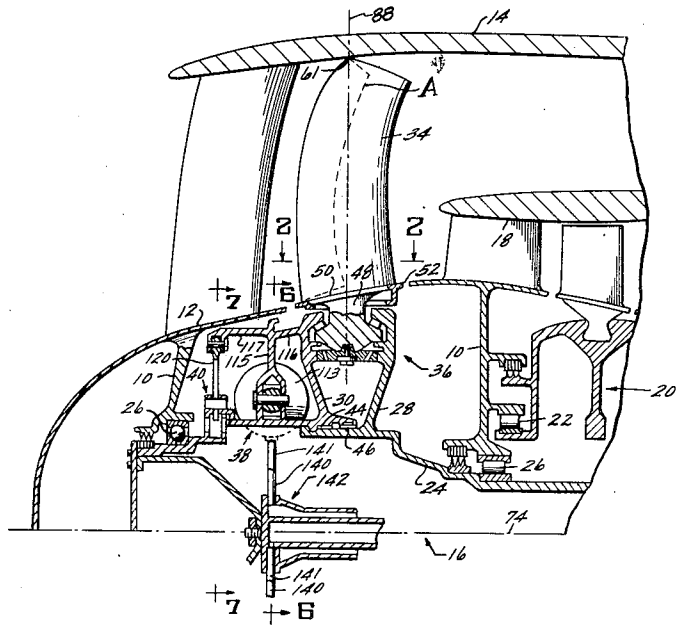
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[57] **ABSTRACT**

A support, actuation, and balancing structure for rotor blades in a variable blade angle axial flow fan, compressor or turbine as may be used in a turbojet or turbofan engine. The supporting structure may include a shaft attached to each rotor blade and restrained from outward radial travel, under high centrifugal loading, by the inwardly facing surfaces of two spaced apart discs. Actuation may be provided by at least one fluid powered piston which controls blade angulation through rotation of one disc member relative to the other. Centrifugally controlled balancing means may also be provided to insure a uniform flow condition through the blades by maintaining the blades in a stable position of angulation upon disengagement of the actuator. Each rotor blade preferably includes a central axis about which the blade angle is varied, and which is in close proximity to the intersection of the leading edge of the blade with the blade tip.

15 Claims, 7 Drawing Figures



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SHEET 1 OF 3

FIG 1

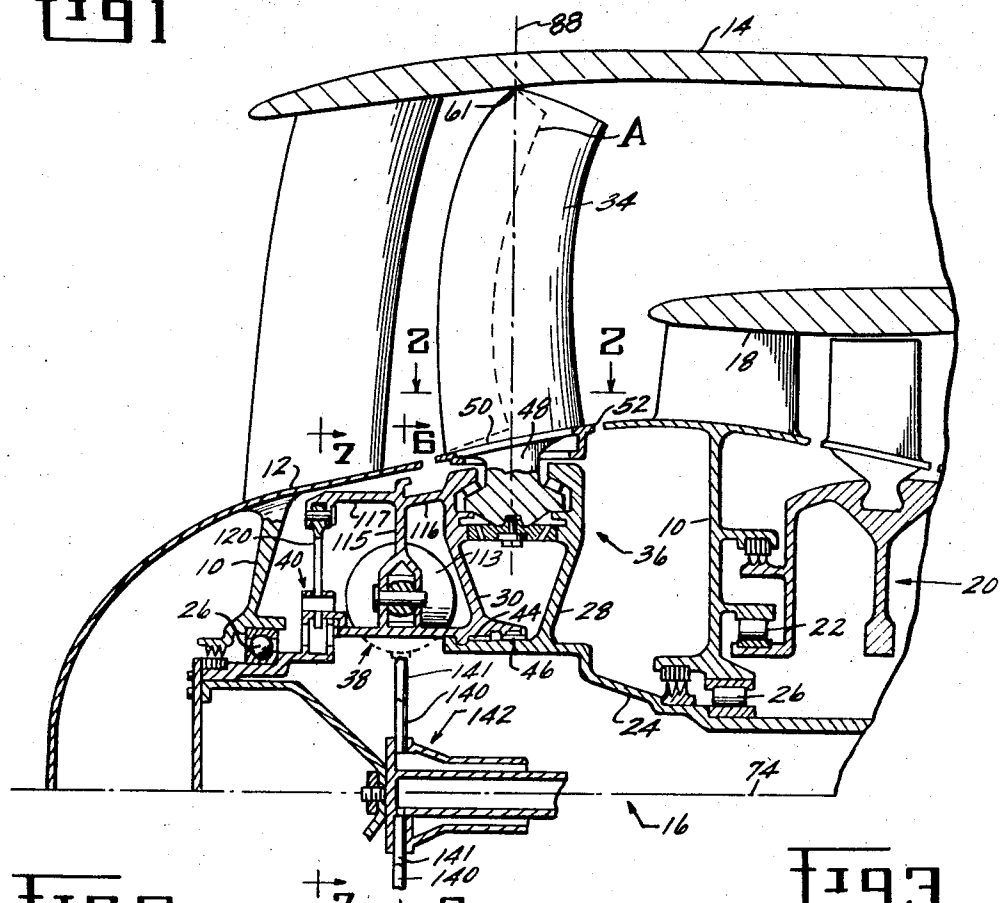


FIG 2

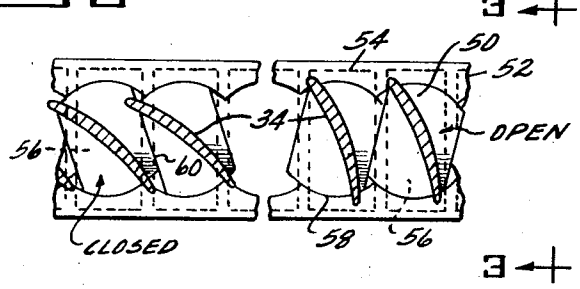
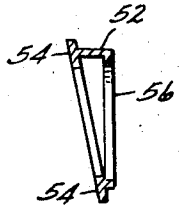


FIG 3



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Fig 4

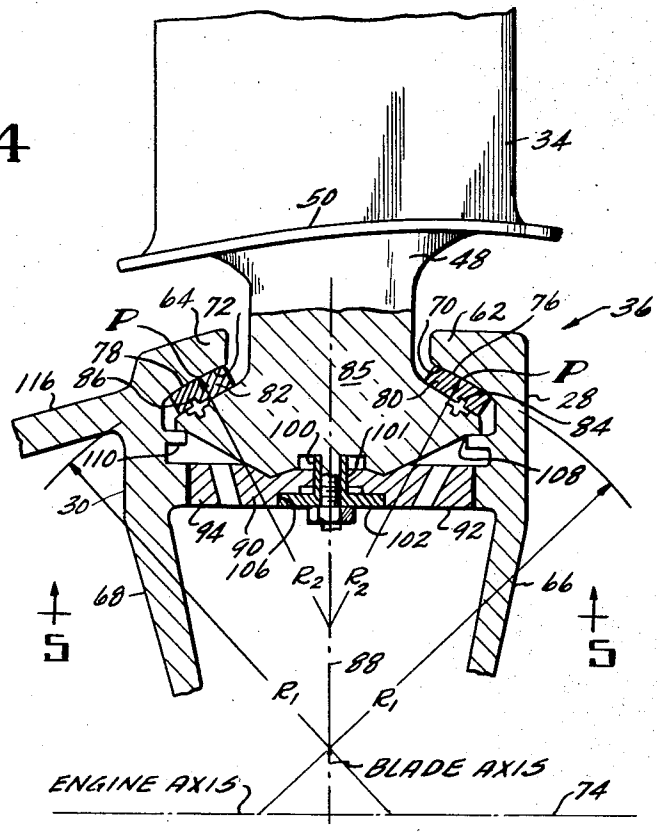
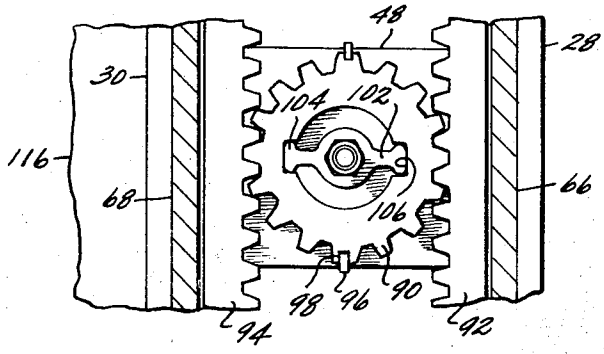


Fig 5



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Fig 6

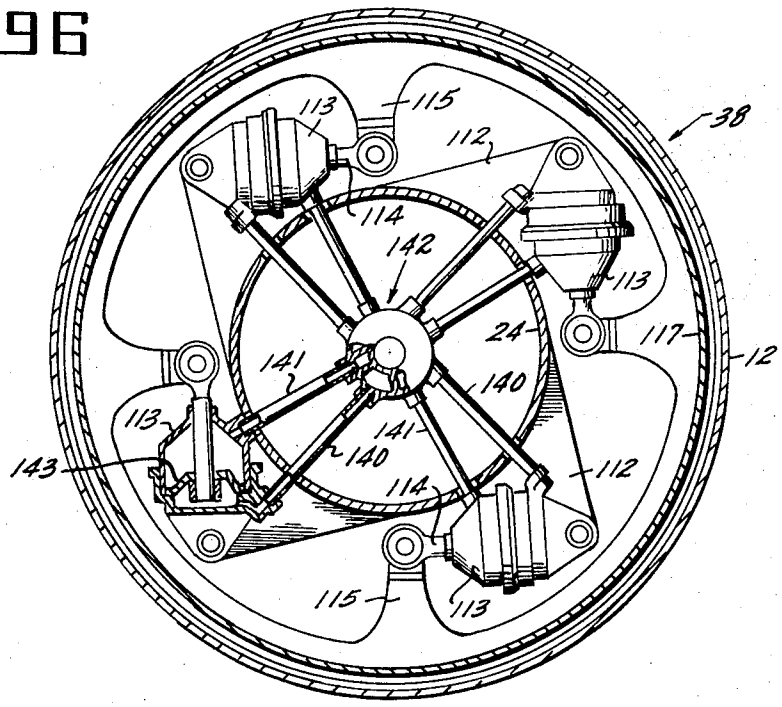
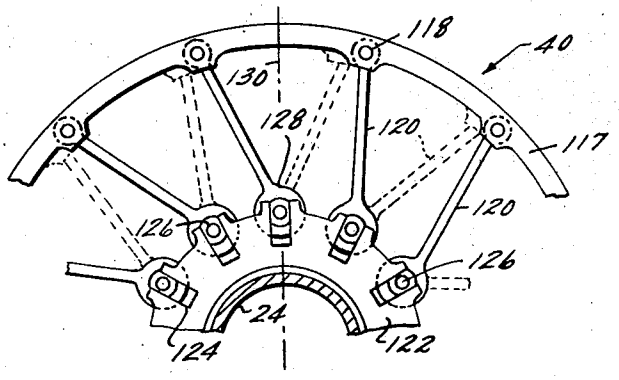


Fig 7



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ROTOR WITH VARIABLE ANGLE BLADES

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation-in-part of application Ser. No. 868,753 filed Oct. 23, 1969 now abandoned.

BACKGROUND OF THE INVENTION

This invention relates to support, actuation, and balancing structures for shafts which are subjected in operation to high axial loads, and more specifically to support, actuation, and balancing structures for variable angle rotor blades in the compressor, fan or turbine of a gas turbine engine, including particular blade configurations suited for variable angle application.

Greater flexibility in operation of gas turbine and turbofan engines over a wide range of operating conditions could be achieved if it were practical to provide rotors with variable angle blading. The shafts upon which variable angle blades are supported become subjected to high axial loads under the influence of centrifugal force. Therefore, one requirement of a variable angle blading system is a relatively low friction bearing structure for supporting rotor blade shafts in a high centrifugal force field. It has been common in the art relating to variable angle aircraft propellers to provide this support with separate relatively large anti-friction bearings at each propeller blade shaft; however, such a technique is not practical in a gas turbine rotor because of the combination of high centrifugal loading on a large plurality of rotor blades and the lack of adequate space in a compressor, fan or turbine rotor for providing individual bearings for each blade. Simple sliding face bearings could be provided to retain blades under high centrifugal load and meet physical space limitations. However, they would involve excessive friction and make variation of blade angle difficult if not impractical during operation of the rotor. What is needed, therefore, is a bearing support arrangement which features the essential rolling contact of an anti-friction bearing combined with the structurally efficient configuration of a sliding face bearing.

Blade actuation structure must also be simple and rugged to withstand the high centrifugal loading upon rotation of the rotor. In addition, should a failure occur in the actuating structure, a means for balancing the blades must be provided for operation independently of the actuating means to insure a stable flow condition eliminating the risk of stalling or other malfunction of the engine.

It has also been found that rotor blade configurations well known to the art are not efficient for variable angle applications. Clearance between the blade tip and stator surface becomes excessive in order to avoid interference between the blade and stator during variations of blade angle. The excessive clearance adversely affects engine performance by increasing flow losses and reducing efficiency.

Therefore it is an object of this invention to provide a variable angle blade support which provides a structurally efficient configuration for restraining outward radial travel under high centrifugal loading while still retaining the rolling contact of a typical anti-friction bearing.

It is another object of this invention to provide a variable angle blade actuation and balancing structure which insures a stable flow condition through the blades upon actuator disengagement.

It is a further object of this invention to provide a novel blade configuration that maximizes the blade efficiency by reducing the effective clearance between the blade tip and the interior surface of the stator casing.

SUMMARY OF THE INVENTION

An axial flow apparatus includes a stator, a rotor, and a variable angle rotor blade system. The rotor blade system includes at least one row of variable angle rotor blades peripherally disposed on the rotor together with support and actuation structure for the blades. The support and actuation structure comprise a support member movably disposed on the rotor together with actuating means for moving the supporting member relative to the rotor. Connecting means transmit movement of the support member relative to the rotor into various angles of blade orientation. Balance means which are responsive to rotation of the rotor, transmit centrifugal force to the support member to maintain the support member in a stable position upon disengagement of the actuator.

The support structure may include a shaft at the bottom of each blade with an enlargement at one end of the shaft. The enlargement may include a spherical bearing surface of radius R_1 symmetrical with respect to the axis of the shaft and restrained from outward radial travel under centrifugal loading by a first disc secured to a rotor and a second disc rotatably mounted to the rotor. Each disc may have an annular ring portion oriented toward the corresponding annular ring on the opposite disc. The annular ring portions each may have a spherical inner surface of radius R_2 (R_2 being larger than R_1) symmetrical with respect to the axis of the rotor shaft. Outward radial restraint is provided by supporting the spherical surface of the enlargement in contact with the inner surface of the annular ring portions.

Actuation means may be provided for rotating the second disc relative to the first disc. Relative rotation between the discs will cause a corresponding rotation of the shaft and a rolling contact between the spherical surface of the enlargement and the spherical surfaces of the annular ring portions.

Each rotor blade includes a central axis about which the blade angle is varied. The central axis is generally normal to the rotor axis and intersects the point of blade attachment to the rotor. The central axis is also preferably in close proximity to the intersection of the leading edge of the blade with the blade tip. The incremental centers of gravity for the incremental blade sections should be arranged so that together they present a balanced force condition at the point of blade attachment to the rotor.

DESCRIPTION OF THE DRAWINGS

While the specification concludes with claims distinctly claiming and particularly pointing out the invention described herein, it is believed that the invention will be more readily understood by reference to the discussion below and the accompanying drawings in which:

FIG. 1 is a partially fragmented section view of the forward end of a turbofan engine embodying the invention;

FIG. 2 is a section view taken along the line 2—2 of FIG. 1;

FIG. 3 is a section view taken along the line 3—3 of FIG. 2;

FIG. 4 is a section view illustrating on an enlarged scale the fan blade bearing structure also shown in FIG. 1;

FIG. 5 is a section view taken along the line 5—5 of FIG. 4;

FIG. 6 is a section view taken along the line 6—6 of FIG. 1 showing the actuation means of this invention;

FIG. 7 is a section view taken along the line 7—7 of FIG. 1 showing the balancing means of this invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

FIG. 1 shows in cross section the forward end of a typical dual rotor turbofan engine in which the invention has been incorporated to provide variable angle rotor blades. While the invention is shown in the environment of a turbofan engine, it is to be understood that it is equally applicable to a gas generator compressor and to turbine rotors for a gas generator turbine, fan turbine, or the power turbine of a turboshaft engine. Additionally, the invention could be similarly applied in any environment in which a thrust bearing and actuation means is required for a shaft (such as the blade shaft) which is subject to high axial loads.

The engine structure comprises an engine frame **10**, an inner casing **12** beginning in the region of the fan inlet guide vane and extending to the first stage of compressor rotor blades, a fan outer casing **14** surrounding the fan rotor **16**, and a gas generator casing **18** shown as surrounding the compressor rotor **20** blading and which would extend beyond the breakoff point of FIG. **1** to the aft end of the engine. The gas generator compressor rotor **20** is suitably supported on the engine frame **10** by bearing means **22**, and the rotor assembly **16** is similarly supported at the rotor shaft **24** by bearing means **26**.

The fan rotor assembly **16** comprises the rotor shaft **24**, a fixed disc **28** which is either integrally formed with or secured to the rotor shaft **24**, a disc **30** rotatably mounted to the rotor shaft **24** so as to permit rotation relative to fixed disc **28**, a plurality of rotor blades **34** supported by the blade bearing and actuation structure **36** which forms one aspect of this invention together with disc actuation means **38** and balance means **40** both of which will be explained in the discussion below. Each blade includes a central axis **88** which is generally normal to the rotor axis **74** and about which the blade angle is varied. Axial location of rotatable disc **30** is maintained by thrust bearing **44** and a radial bearing **46** to provide radial support for disc **30**.

Referring to both FIGS. **1** and **2**, the blade mounting in relation to the inner casing includes blade shafts **48** supported by bearing and actuation structure **36**, platforms **50** located radially outward from the bearing and actuation structure **36** in a position to form a smooth flow path with rotatable annular ring **52** adapted to be movable with respect to the inner casing **12**, and the blade air foil **34** which terminates at platform **50**. The movable ring **52** is provided because, as will become apparent, relative rotation between disc **30** and disc **28** will cause the blade **34** centers to move with respect to fixed disc **28**.

Referring to FIGS. **2** and **3**, ring **52** comprises a pair of scalloped members **54** held in axially spaced relation by structure which includes spacer bars **56**. Platforms **50** are circular segments whose arcuate front and rear edges **58** mate with the scallops on members **54**, and whose straight side edges **60** abut when the blades **34** are in either of the two end positions shown in FIG. **3**, there being freedom for limited rotation between the end positions.

FIGS. **1** and **2** show a single stage of blades **34**; however, in some applications it may be desirable to provide a plurality of stages thereof, it being within the capabilities of a person skilled in the art to design a particular configuration which differs from the embodiment shown in this respect.

Details of the blade support and actuation structure **36** are shown in FIGS. **4** and **5**. Referring first to FIG. **4**, the fixed disc **28** and the rotatable disc **30** include annular ring portions **62**, **64** respectively, which extend toward each other from the disc webs **66**, **68**. The inner surfaces **70**, **72** of annular ring portions **62**, **64** are spherical segments, each having a spherical radius R_1 which originates on the rotor axis **74**. Contacting with spherical surfaces **70**, **72** are spherical surfaces **76**, **78** on the bearing plates **80**, **82**, which are secured to enlargement **85** on the end of each blade shaft **48** by pins **84**, **86**, and have a spherical radius R_2 originating on the central blade axis **88**, R_2 being smaller than radius R_1. **6** Constraint on the axial relationship of the discs **28**, **30** is such that, in the plane of intersection of rotor axis **74** and central blade axis **88**, spherical radii through points P on bearing plates **80**, **82** are colinear with the corresponding spherical radii of disc surfaces **70**, **72**. Since radius R_2 is smaller than radius R_1, points P represent geometric points of contact. It is readily seen that for no sliding at points P rotation of front disc **30** relative to rear disc **28** produces a corresponding rotation of blade **34** about its central axis **88**. The relative motion between bearing plates **80**, **82** and disc surfaces **70**, **72** at any instant may be described by two components—rotation about the common radii through points P and rolling about points P in a direction tangential to rotor **16**. Since the contacting surfaces are completely axi-symmetric, each blade axis remains radial and points P remain in the plane of intersection of the rotor axis **74** and the central blade axis **88**. The contact points P, however, move relative to bearing plates **80**, **82** and the disc surfaces **70**, **72**. In practice, because of the high load and the elasticity of the materials, contact pressure spreads over small circular areas with centers at points P, and the frictional torque developed on the loaded surfaces is only that due to rotation about the points P.

While the drawings illustrate and the discussion above speaks of the surfaces on bearing plates **80**, **82** and the disc surfaces **70**, **72** being spherical, it is not necessary that such be the case, and in some applications other surfaces may be preferable. However, it is necessary that the surfaces mentioned be surfaces of revolution about the pertinent axis and that the surfaces on plates **80**, **82** be non-concentric with surfaces

70, 72 in the area surrounding points P to preserve the point contact between mating load bearing surfaces. For example, surfaces 70, 72 could be conical and the outer surfaces on plates 80, 82 can be segments of an appropriate surface of revolution about the blade axis.

Surfaces 70, 72 and the outer surfaces 76, 78 of bearing plates 80, 82 may be asymmetric with respect to the colinear radii passing through points P to provide a non-circular contact between the contacting bearing surfaces, or may be machined with a small degree of asymmetry in the unloaded condition to provide allowance for blade or disc deflection. One possible reason for providing non-circular loaded contact between the bearing surfaces would be to change the blade root stiffness, as, for example, by providing for a tangentially oriented elliptical contact in the loaded condition.

To synchronize the motion and to absorb torque and tangential loading on each blade 34, a beveled pinion 90 at the inner end of each blade shaft 48 engages mating gears 92, 94 fixed to each disc 28, 30. The greatest portion of the load on the gear teeth is due to blade centrifugal torque. To transmit this torque, two lugs 96 are provided on the inner end of the blade shaft 48 and engage with corresponding slots 98 on the pinion 90. Lugs 96 are located nominally in a plane normal to the rotor axis 74, thus leaving the bearing plate 80, 82 spherical surfaces 76, 78 free to rock on the inner surfaces 70, 72 of the disc annular ring portions 62, 64 in a mode in which the central blade axis would be free to deviate from a rotor radius passing through the blade center at its root, i.e., the blades would be free to move in response to forces having components in a plane which is normal to both the rotor axis 74 and the central blade axis 88, hereinafter referred to as tangential forces. These tangential forces, which are relatively low, are transmitted to the beveled pinion 90 through an adapter 100 which extends through a center hole 101 in pinion 90 and is securely fastened to the end of the blade shaft 48. Adapter 100 includes radially extending webs 102 having male splines 104 on their outer ends which mate with corresponding female splines 106 formed in a recess on the pinion 90 face. Webs 102 are nominally oriented in a direction normal to the direction of the tangential forces (i.e., parallel to rotor axis 74) and have a degree of flexibility adequate to effectively isolate the gear teeth from possible loading due to blade 34 vibration or misalignment, yet have sufficient rigidity to maintain the central blade axis 88 colinear with a rotor radius passing through the blade root under steady state load conditions.

To assure synchronization of the pinion 90–gear 92, 94 interaction with the rolling contact interaction between the bearing plate 80, 82 surfaces 76, 78 and the disc annular ring portion surfaces 70, 72, the pitch cone of the beveled pinion 90 is constructed to coincide with the surface of revolution which would be generated by rotating the lines connecting points P with the intersection of central blade axis 88 and rotor axis 74 about the central blade axis 88.

During rotation of the fan rotor 16, centrifugal loading on the blades 34 will urge them radially outwardly from the center of the rotor and assure the contacts at points P between bearing plates 80, 82 spherical surfaces 76, 78 and the disc annular ring portion spherical surfaces 70, 72. However, mechanical support means should be provided to aid in maintaining the radial position of the blades 34 when the rotor 16 is at rest. To this end, the enlargement 85 on each blade shaft 48 has been provided with an annular surface 108 facing the rotor axis 74, and a circular ridge 110 is provided on each of webs 66, 68, which ridges 110 are in slidable contact with the flat annular surface 108 when rotor 16 is stationary.

Referring now to FIG. 4 only, response of central blade axis 88 to relative rotation between rotatable disc 30 and fixed disc 28 can be visualized. As the discs 28, 30 are moved relative to each other, pinion 90 and gears 92, 94 will interact in a rack and pinion type interaction, and the central blade axis 88 will move about the rotor axis 74 with respect to a point on fixed disc 28. Thus, the need for the movable rings 52 described in connection with FIGS. 1 and 2 is established.

Referring again to FIG. 1, it is desirable that the clearance between the tip of blade 34 and the outer casing 14 be minimized throughout the blade's range of angulation to maximize performance. In order to reduce the aforementioned clearance, each blade of this invention is formed so that the central axis 88 about which the blade angle is varied, is in close proximity to the intersection of the leading edge of the blade with the blade tip at 61. For known blades, the intersection of the leading edge with the blade tip must be substantially forward of the central axis in order to avoid unbalanced force conditions which act to cause high bending stresses at the point of blade attachment to the rotor. The result is an excessive clearance over that portion of the blade tip that is forward of the central axis.

The novel blade configuration of this invention, however, achieves a central axis of angulation in close proximity to the intersection of the leading edge of the blade with the blade tip at 61, in combination with a balanced force condition at the point of blade attachment to the rotor. The balanced force condition is achieved by adjusting the incremental centers of gravity for each incremental blade section in the following manner. Broken line A illustrates the locus of incremental centers of gravity for the incremental blade sections wherein each incremental blade section lies in a plane orthogonal to the central axis 88. The locus is arranged so that the sum of the centrifugal forces acting on each point on the locus are balanced and in the aggregate cause no bending moment at the point of blade attachment to the rotor. The locus is also displaced relative to the plane of intersection of rotor axis 74 and central blade axis 88 so as not to produce any bending moments that would act to distort the curvilinear surface of each blade face. The outer radial limits for the blade tip are defined by a spherical surface generated by a radius originating at the intersection of the central blade axis 88 with the rotor axis 74 and terminating at a point which passes through the intersection of the leading edge with the blade tip at 61. This limit is imposed in order to avoid interference between the blade tip and stator casing for all blade positions. Thus, the blade may be balanced at the point of blade attachment to the rotor and the steady bending stresses due to curvature of the airfoil, fail to distort the curvilinear surfaces of each blade face.

Referring now to FIG. 6, the configuration of disc actuation means **38** is illustrated in more detail. Brackets **112** are secured to rotor shaft **24**, and fluid operated actuators **113** are pinned with a clevis arrangement to brackets **112**. The actuation rods **114** of actuators **113** are connected to inwardly directed extensions **115** of annular member **116** which is connected to rotatable disc **30** (see FIG. 1) so that motion thereof will be transmitted to disc **30**. Actuators **113** can be controlled from a remote location and include end stops designed to give positive positioning of the actuation system at two end points. Actuator fluid is supplied through the generally radial conduits **140**, **141** such that an imbalance in fluid pressure acts to cause translation of slidable balance piston **143**. Conduits **140**, **141** are in flow connection with a central discharge hub assembly **142** which extends along the rotor axis **74** and supplies actuator fluid from a source which is not shown.

FIG. 7 illustrates the balance means **40** provided to utilize the rotational energy of rotor shaft **24** to maintain the actuation structure in selected positions without requiring pressurization of the actuator **113** fluid except for transition from one position to another. The balance means operates to insure a stable flow through the blades upon disengagement of the actuator whether by design or by accident. An annular support member **117** is secured or integrally formed with rotatable disc **30** (see FIG. 1) and includes a plurality of peripherally spaced holes **118** to which a like plurality of links **120** are rotatably pinned. A second annular support member **122** is secured to rotor shaft **24** and includes a plurality of radially oriented elongated slider tracks **124** around its periphery. The inner ends of links **120** are each pinned to a slider **126** which is slidably engaged in a track **124**, and each link **120** further includes a relatively large mass of material **128** symmetric with respect to the pinned joint. When the balance system is in the position shown by the solid lines of FIG. 7, each link is oriented angularly in a counter-clockwise direction with respect to a rotor radius **130** passing through a track **124**. The centrifugal forces generated by the rotating masses **128** are transmitted through links **120** and establish a counter-clockwise tangential force component acting on the first annular support member **117**. Thus, the transmitted tangential force component urges and maintains the actuation structure in its counter-clockwise end position even upon disengagement of the actuator.

When the actuator **113** (see FIG. 6) is pressurized to transition the actuation structure from the solid line position shown in FIG. 7 to the over center, clockwise position shown by broken lines, each slider **126** moves radially inward in its track **124** until the actuation structure is in its approximate center position, and then moves radially outward in its track **124**. It is readily seen that the same centrifugal forces which urged the actuation structure to its counter-clockwise end position will also urge the structure towards its clockwise end position. Thus balance means **40** will assure stability of the actuation system should the actuators **113** for some reason lose fluid pressure in that there will be stable maintenance of the structure in one end position or another.

Having thus described one embodiment of the inventive combination and sub-combinations, what is desired to be secured by letters patent is as follows:

1. A support and actuation structure for a rotatable shaft subject to high axial loads, said structure comprising:

a structural frame;

a first shaft rotatably supported in said frame;

a first disc secured to said first shaft and having a web portion and an annular ring portion, said annular ring portion having an inner surface of revolution about the axis of said first shaft;

a second disc rotatably mounted to said first shaft and having a web portion and an annular ring portion oriented toward the annular ring portion of said first disc, said latter mentioned annular ring portion having an inner surface of revolution about the axis of said first shaft;

a second shaft having an enlargement on one end thereof, said enlargement including surfaces of revolution about the axis of said second shaft;

means supporting the surfaces of revolution of said enlargement in contact with the inner surfaces of said annular ring portions, said contacting surfaces being further characterized by being non-concentric with each other in the plane defined by intersection of the axes of said first and second shafts;

means constraining the discs and the second shaft so that the respective points of contact between said enlargement and said annular ring portions are symmetrically located with respect to the axis of said second shaft; and

means for rotating said second disc relative to said first disc.

2. The structure recited in claim **1** wherein the inner surfaces of said annular ring portions are spherical surfaces having equal radii R_1 and the surfaces of revolution on said enlargement are spherical surfaces having equal radii R_2, R_2 being smaller than R_1.

3. The structure recited in claim **2** wherein the means supporting said enlargement in contact with said annular ring portions and constraining said second shaft includes an annular surface on said enlargement and a surface on each web in slidable contact therewith.

4. The structure recited in claim **3** wherein a bevel gear is attached to each web portion, said gears facing each other, and a beveled pinion is secured to said second shaft and meshes with said bevel gears, the pitch cone of said pinion being defined by revolution of the lines joining the points of contact between the spherical surfaces to the intersection of said shafts, about the axis of said second shaft, whereby said gears and pinion will provide positive synchronization of the relative motion between said discs with the corresponding rolling motion between the spherical surface of said enlargement and the inner surfaces of said annular ring portions.

5. In an axial flow apparatus having a stator and a rotor, a variable angle rotor blade system comprising at least one row of variable rotor blades peripherally disposed on said rotor with each blade having a central axis about which the blade angle may be varied, and support structure for said blades, said structure comprising:

a first disc attached to said rotor and having a web portion and an annular ring portion, said annular ring portion having an inner surface of revolution about the axis of said rotor;

a second disc rotatably mounted to the rotor and having a web portion and an annular ring portion oriented toward the annular ring portion of said first disc, said latter mentioned annular ring portion having an inner surface of revolution about the axis of the rotor;

a blade shaft attached to the inner end of each blade, said blade shaft having an enlargement on its inner end, said enlargement including surfaces of revolution about the axis of said blade shaft which face outwardly with respect to the rotor axis;

means supporting the surfaces of revolution of said enlargements in contact with the inner surfaces of said annular ring portions, said contacting surfaces being further characterized by being non-concentric with each other in the planes defined by intersection of the central axes of said blade and said rotor;

means constraining the central axis of each blade in a plane substantially normal to the rotor axis;

means constraining the spacing of said discs so that the respective points of contact between each enlargement and said annular ring portions are symmetrically located with respect to the central blade axis, so that said blades will be radially supported against centrifugal loads by essentially point contact between each enlargement and each annular ring portion and relative rotation between said discs will cause said blades to rotate about their central axes with minimum friction.

6. The system recited in claim **5** wherein the means supporting each enlargement in contact with said annular ring portion and constraining said central blade axis includes an annular surface on each said enlargement normal to the blade axis and facing the rotor axis, and a surface on each said web in slidable contact with said annular surface.

7. The system recited in claim **6** wherein a bevel gear is attached to each web portion, said gears facing each other, and a beveled pinion is secured to each blade shaft and meshes with said bevel gears, the pitch cone of said pinion being defined by revolution of the lines joining the points of contact between said contacting surfaces and the intersection of the central blade axes and rotor axis, about the central axis of said blade whereby said gears and pinion will provide positive synchronization of the relative motion between said discs with the corresponding rolling motion between the surface of revolution of said enlargement and the inner surface of said annular ring portions.

8. The system recited in claim **7** wherein said surfaces of revolution are spherical.

9. The system recited in claim **7** wherein the means securing said pinion to said blade shaft comprises:

a surface formed on the end of said blade shaft;

a seat formed on a first face of said pinion and mated with said surface;

a recess formed in the second face of said pinion opposite said first face, said recess including at least two female splines;

a center hole defined through said pinion;

an adapter extending through said center hole and secured to said blade shaft, said adapter comprising two webs having male splines at their ends engaged with said female splines, said adapter additionally holding said surface and said seat in contact with each other, said webs being nominally located in a direction parallel with the axis of said rotor and being further characterized by flexibility adequate to isolate said meshed pinion and gears from loading caused by lateral vibration or misalignment of said blade shaft;

a pair of lugs on the inner end of said blade shaft and radially spaced from the axis thereof in planes nominally normal to the axis of said rotor; and

a pair of slots in said pinion mated with said lugs.

10. The system recited in claim **5** including actuating means for rotating said second disc relative to said first disc wherein said actuating means include at least one fluid powered actuator having a range of positions from one end stop to another end stop.

11. The system recited in claim **10** wherein said actuation structure further includes centrifugal balance means responsive to the rotation of said rotor for urging the blades to and maintaining the blades at the nearest end stop upon disengagement of the actuation force.

12. The system recited in claim **11** wherein said balance means comprises:

a first annular support member secured to said second disc;

a second annular support member secured to said rotor shaft;

a plurality of links pinned to said first annular support member for rotation with respect thereto and secured to said second annular support member for limited radial sliding motion and rotation with respect thereto;

said links further including a mass of material for generating centrifugal force located at their inner ends.

13. The system recited in claim **5** wherein the central axis about which the blade angle may be varied is in close proximity to the intersection of the leading edge of the blade with the blade tip and the locus of incremental centers of gravity for the incremental blade sections which lie in planes orthogonal to the central blade axis is arranged so that the centrifugal forces acting on all points on the locus balance about the central blade axis to cause no bending moment at the point of blade attachment to the rotor.

14. In an axial flow apparatus having a stator and a rotor, a variable angle rotor blade system comprising at least one row of variable rotor blades peripherally disposed on said rotor and support and actuation structure for said blades, said structure comprising:

a support member movably disposed on said rotor;

actuating means for moving said supporting member relative to said rotor;

connecting means for transmitting movement of said support member relative to said rotor into blade angle orientation;

and balance means having a second support member secured to said rotor shaft and spaced radially inward of said first support member together with a plurality of links each of which is rotatably attached to said first and second support members wherein one of the rotatable attachments of each link is adapted for limited radial sliding motion with respect to the support members and each link

includes a mass of material for generating a centrifugal force for urging the blades to a stable position thereby insuring a stable flow through the blades upon disengagement of actuator force.

15. The axial flow apparatus of claim 14 wherein the second support member is an annulus having a plurality of radially oriented elongated slider tracks around its periphery and the inner radial end of each link is rotatably pinned to a slide which slidably engages a slider track to provide said limited radial sliding motion.

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