Feb. 25, 1958 G. H. VOADEN 2,824,716

MECHANISM FOR LIMITING RUNAWAY SPEED OF ADJUSTABLE BLADE TURBINES

Filed Dec. 27, 1952 7 Sheets-Sheet 1

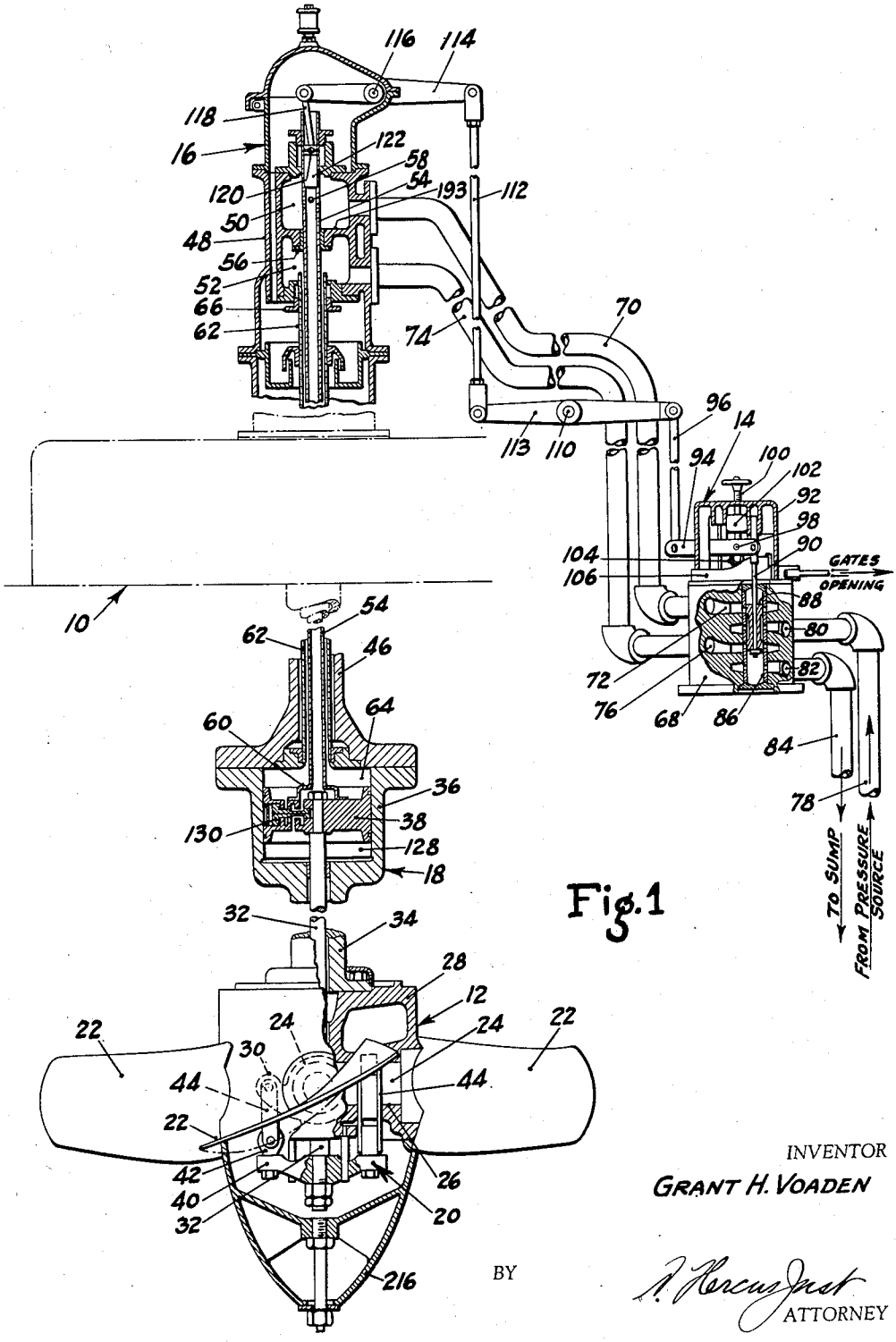


Fig. 1

INVENTOR
GRANT H. VOADEN

BY

ATTORNEY

Feb. 25, 1958 G. H. VOADEN 2,824,716

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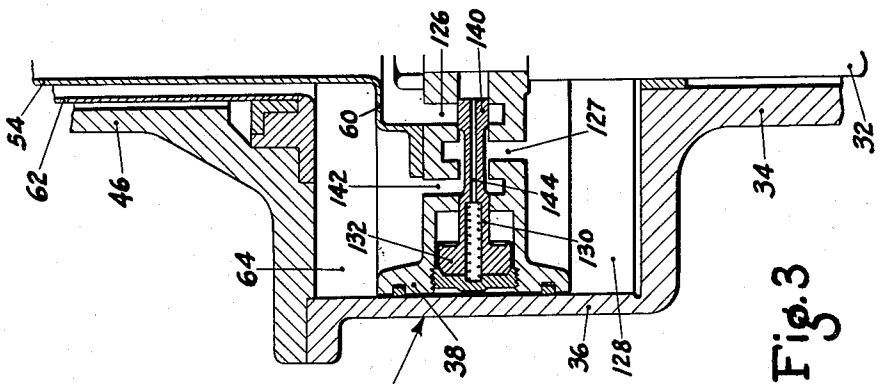


Fig. 3

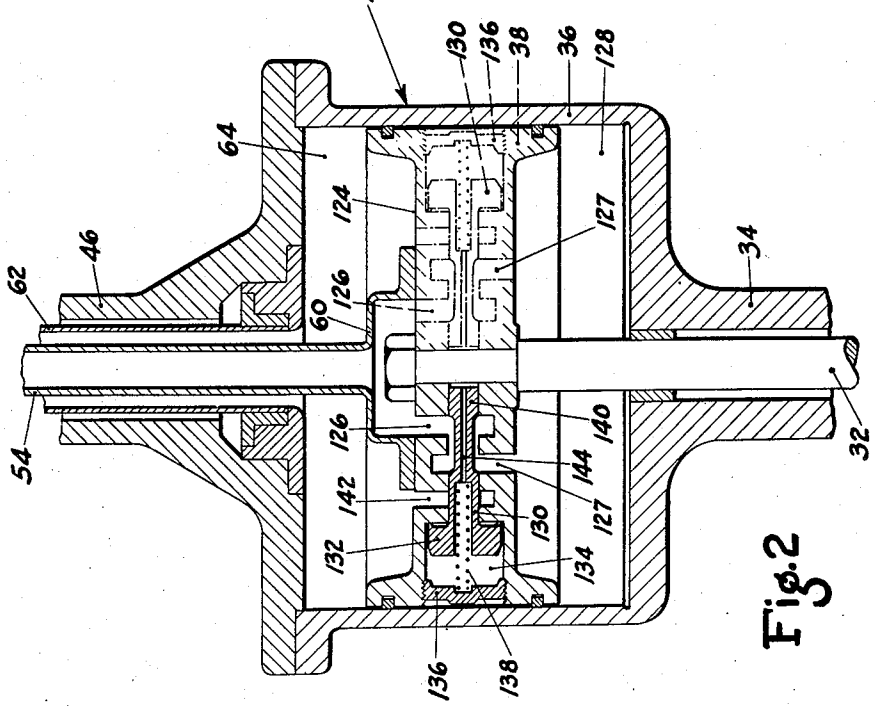


Fig. 2

INVENTOR

GRANT H. VOADEN

BY

ATTORNEY

Feb. 25, 1958 G. H. VOADEN 2,824,716

MECHANISM FOR LIMITING RUNAWAY SPEED OF ADJUSTABLE BLADE TURBINES

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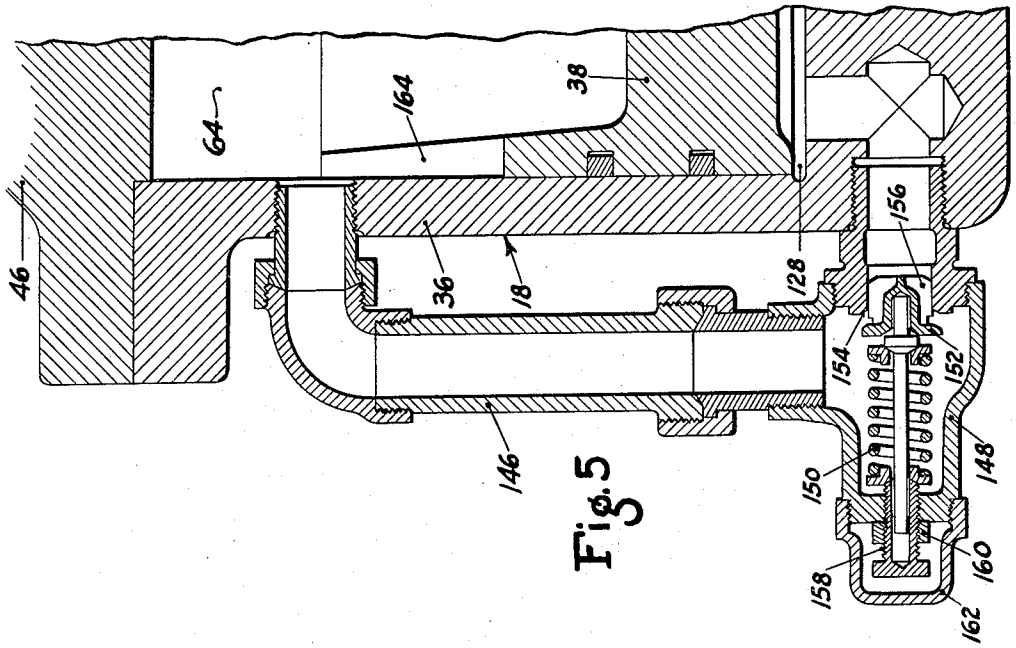


Fig. 5

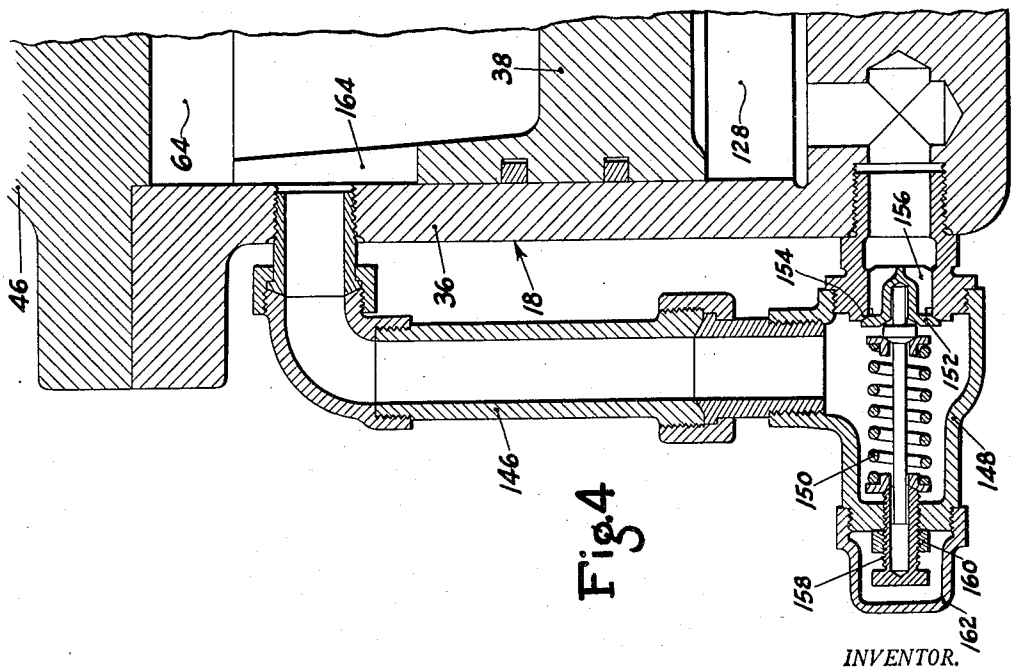


Fig. 4

INVENTOR.
GRANT H. VOADEN
BY
ATTORNEY

Feb. 25, 1958 G. H. VOADEN 2,824,716

MECHANISM FOR LIMITING RUNAWAY SPEED OF ADJUSTABLE BLADE TURBINES

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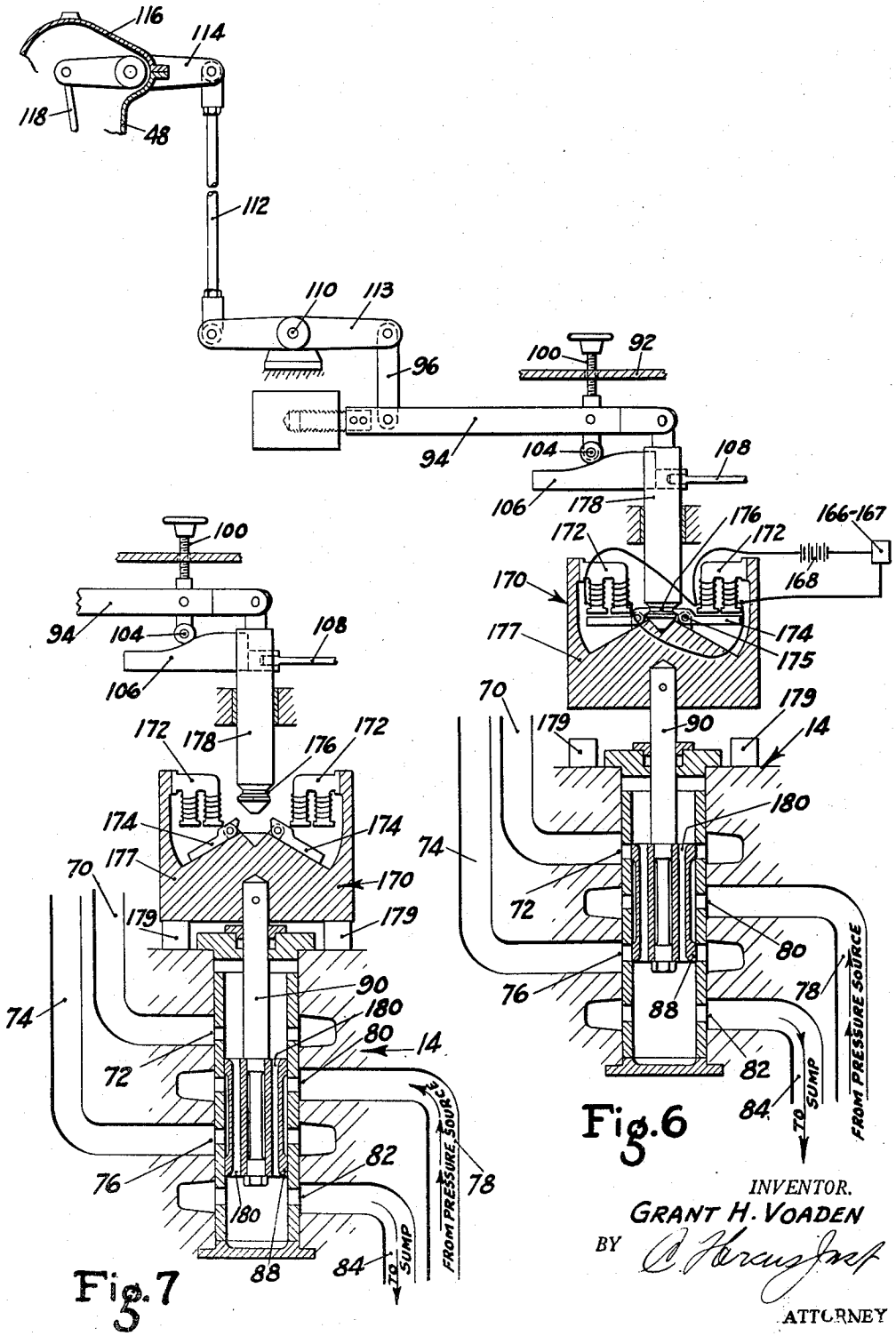


Fig. 6

Fig. 7

INVENTOR.
GRANT H. VOADEN
BY
ATTORNEY

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MECHANISM FOR LIMITING RUNAWAY SPEED OF ADJUSTABLE BLADE TURBINES

Filed Dec. 27, 1952 7 Sheets-Sheet 5

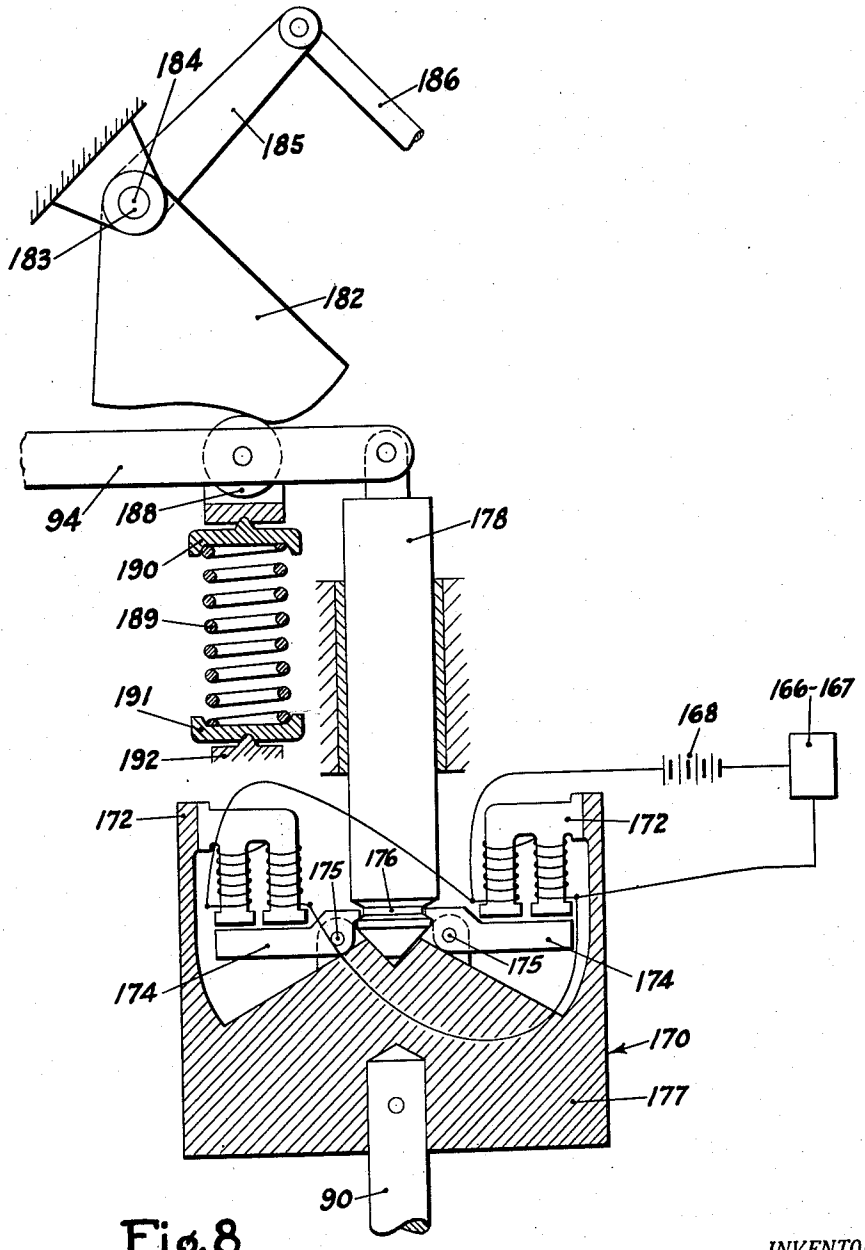


Fig. 8

INVENTOR.
GRANT H. VOADEN
BY C. Hercus Jost
ATTORNEY

Feb. 25, 1958 G. H. VOADEN 2,824,716

MECHANISM FOR LIMITING RUNAWAY SPEED OF ADJUSTABLE BLADE TURBINES

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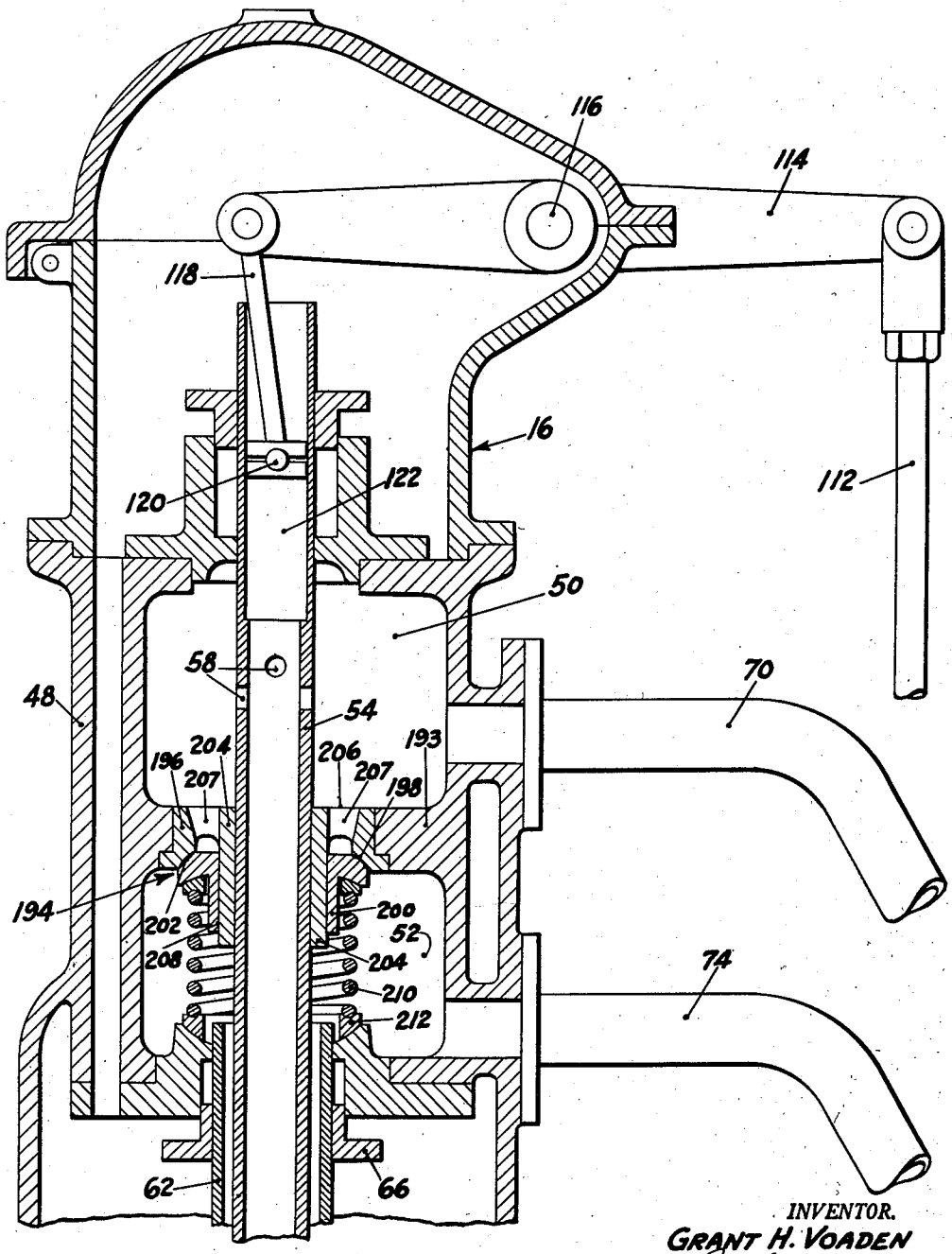


Fig. 9

INVENTOR.
GRANT H. VOADEN
BY
ATTORNEY

Feb. 25, 1958 G. H. VOADEN 2,824,716

MECHANISM FOR LIMITING RUNAWAY SPEED OF ADJUSTABLE BLADE TURBINES

Filed Dec. 27, 1952 7 Sheets-Sheet 7

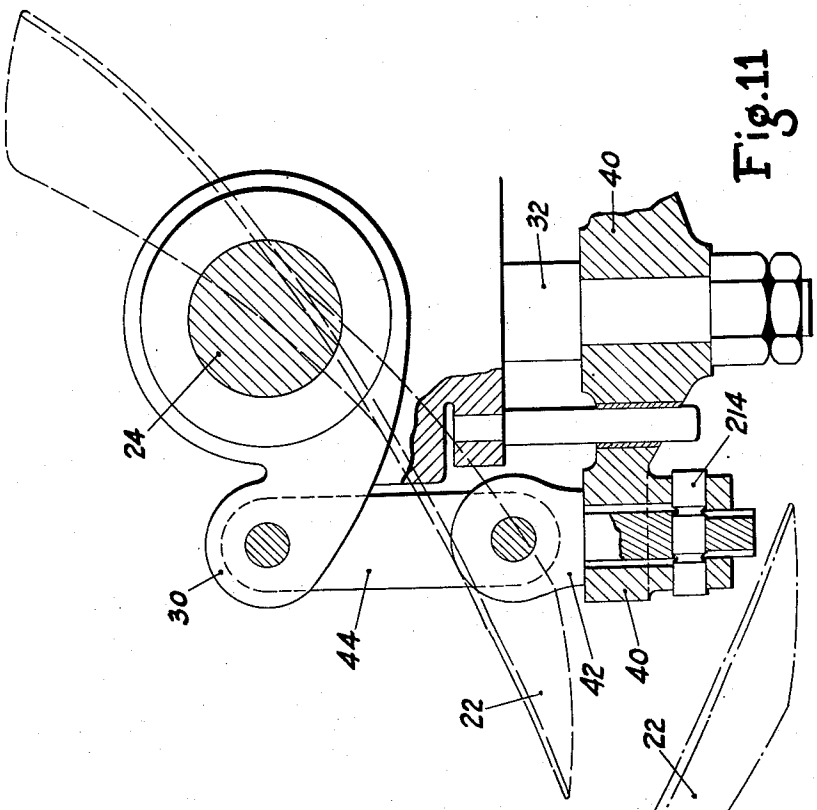


Fig. 11

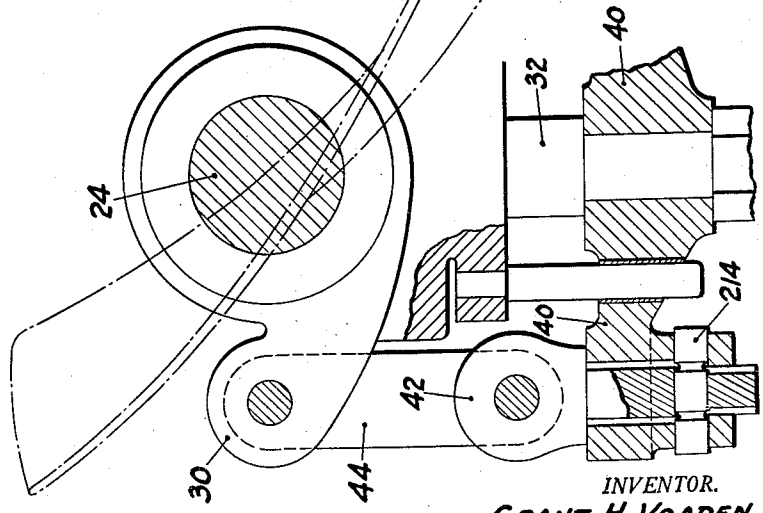


Fig. 10

INVENTOR.
GRANT H. VOADEN
BY
ATTORNEY

United States Patent Office 2,824,716 Patented Feb. 25, 1958

2,824,716

MECHANISM FOR LIMITING RUNAWAY SPEED OF ADJUSTABLE BLADE TURBINES

Grant H. Voaden, York, Pa., assignor to S. Morgan Smith Company, York, Pa., a corporation of Pennsylvania

Application December 27, 1952, Serial No. 328,161

14 Claims. (Cl. 253—143)

This invention relates to an addition to the normal control mechanism of a Kaplan type turbine, said addition comprising positive means for limiting the runaway speed which occurs if said normal control mechanism should fail to function as intended. More particularly, the invention relates to a hydraulic turbine having a Kaplan type runner or propeller on which the blades are movable for disposing the same at various pitches, thereby to obtain the optimum efficiency of the turbine over its entire range of operation.

Adjustable blade runners or propellers have been utilized in hydraulic turbines for a number of years past in hydroelectric power plant installations for purposes of driving electric generators. In such installations, the head of water impounded by a dam will vary during different times of the year and especially between high water or flood stages which usually occur in the spring months and the low water stages which occur during the dry summer months. Also, load requirements vary on generators in such hydroelectric power plants during and between peak periods and off-peak periods occurring each day.

The adjustable blade type Kaplan turbine is admirably suited to meet these various changing conditions in hydroelectric power plants. Water for driving the turbine runner is usually introduced into the runner cavity by an annular arrangement of a plurality of angularly adjustable wicket gates. These gates are adjusted to vary the quantity of water supplied to the runner. In the space between the gates and the runner, the water whirls inward and downward, the whirl component being greatest at low gate openings and decreasing as the gates are opened and the quantity of water increased. It is discharged from the runner in a substantially axial direction. When the wicket gates are adjusted, for example, to meet changing load demands, the blade pitch of the turbine also is adjusted automatically and substantially simultaneously with the gate movement so that at all gate positions the related blade angle is such as to produce maximum turbine efficiency. As the gates open, the blades normally are moved to steeper angles and vice versa.

The turbine runner blades are pivotally supported by a hub from which the blades project uniformly, the hub having bearings which pivotally receive trunnions projecting inwardly from the blades. These blades can be varied in pitch, simultaneously, between predetermined optimum open and closed, or respectively steep and flat, positions within required limits by the use of a hydraulic cylinder or a geared motor usually contained in the turbine shaft. The pitch of the blades may be varied in either direction through a predetermined range, usually, but without restriction thereto, about 26°, as dictated by a governor embodied in the normal control system for each turbine.

If the hydroelectric unit, while operating under load at normal speed is, for example, suddenly disconnected from its load, such as by the opening of the switch connecting the generator outgoing terminals to the power transmission line, the governor and control system normally function to move the wicket gates to closed positions, thus shutting off the water supply so as to stop the unit. The blades of the turbine runner also simultaneously will be moved to closed position. However, if these devices do not function normally and relatively quickly, so-called runaway speed will be developed by the turbine and the generator driven thereby. In turbines of the type described, maximum runaway speeds normally amount to at least two and one-half times the normal speed of the turbine. Obviously it is necessary to design generators to withstand such maximum contemplated runaway speeds.

It is sometimes impracticable and always expensive to design a generator capable of sustaining runaway speeds of the magnitude of two and one-half or more times the normal speed. Considering the fact that generators in many relatively large hydroelectric power installations cost upward from $1,000,000, even any small percentagewise increase in cost amounts to a sizable sum. Further, the cost of a given generator increases at an accelerating rate as the runaway speed for which it must be designed increases. In addition, while there have been relatively few failures of the normal control mechanism resulting in serious damage to hydroelectric units, nevertheless both the generator and the turbine for driving the same must be designed with adequate factors of safety to enable the same to withstand runaway speeds of the maximum magnitude as described above. So designing and building the turbines and generators adds very materially to the costs thereof as compared with the cost of the same if such maximum runaway speeds could be minimized by the use of one or more of the devices described and claimed herein.

The individual blades of the runner customarily have their axes of pitching so located that at the normal or rated rotational speed of the unit, the maximum external force required to increase pitch approximates the maximum force required to decrease pitch.

As the speed of the turbine increases above the rated speed when runaway speed occurs, the net result of the change in the forces tending to open the blades becomes progressively and substantially greater. The effect of this tendency can be calculated from the design of the runner and/or determined by laboratory testing of a working model. Any desired net effect of this inherent tendency can be obtained by proper location of the pitching axes of the blades relative to the blade areas. It is essential for the proper functioning of certain of the various devices covered by this invention that the pivot axes of the blades are so located relative to the blade areas that at any rotational speed above rated speed, the blades have a positive, inherent tendency to increase pitch, i. e. to go open, regardless of their pitch or angular positions. It will be assumed regarding said certain devices, except where specifically stated otherwise, that the blade pitching axes are located to produce this opening tendency at all speeds above rated speed.

A characteristic of one form of adjustable blade turbine, to which this invention applies, is that at a given head, if the turbine is free to rotate at will without load, the runaway speed is lower at the maximum permitted open blade pitch than at lower pitches within the normal pitching range. This is so because the component of force in the direction of rotation of the turbine is reduced as the blades open; for example if the blade position were axial with the turbine shaft, the rotational component of the force is approximately zero. This characteristic is used advantageously in conjunction with various forms of runaway speed control means comprising the present invention.

It is the principal and basic object of the present in-

vention to provide various forms of control means by which the runaway speed of an adjustable blade turbine may be positively limited to an amount above normal, or rated speed, for which it is economically and practically feasible to design and construct the generator while maintaining the usual factors of safety against damage by said runaway speed.

It is another object of the invention to provide in adjustable blade turbines various selective embodiments of control means respectively operable to hold the unit speeds of said turbines within predetermined limits when the normal control means, such as the governing equipment or pressure system, fails to function in its customary manner.

It is still another object of the invention to provide, in adjustable blade turbines having blades which inherently tend to move to the open or steep positions thereof at rotational speeds in excess of rated speed, control means which utilize such tendency to operate or function to hold the unit speeds of said turbines within predetermined limits when the normal control means fail to function as contemplated.

It is another object of the invention to embody actuating means within the control mechanisms referred to above, said actuating means being operable upon the occurrence of overspeed of a predetermined maximum to free the blades from the control of the normal positioning means and permit movement of the blades toward the normal maximum permitted open positions thereof, or beyond said normal maximum open positions, thereby to restrict the speed of the turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

It is still another object of the invention to utilize respectively in various forms of the abovementioned control devices fluid pressures and/or centrifugal forces in excess of predetermined, normal amounts for purposes of actuating the control means to effect movement of the blades toward the maximum permitted open positions thereof so as to maintain the speed of the turbine within safe limits when conditions occur tending to produce runaway speeds.

One further object of the invention is to utilize respectively in the aforementioned various embodiments of control mechanisms, hydraulic, electrical and mechanical means to control and effect movement of the blades toward the open positions thereof so as to limit the speed of the turbine when conditions occur tending to produce runaway speeds.

A still further object of the invention is to provide a number of embodiments of control and actuating means for the blades of an adjustable blade turbine operable to prevent the occurrence of unsafe speeds in excess of normal, said control means being rugged, durable, and positive, capable of ready manipulation, certainty of functioning, easily adjusted, relatively inexpensive to manufacture and install, capable of long use with minimum service requirements, and readily adapted for installation not only in newly manufactured turbines but also to being installed in existing turbines in the field.

Details of the foregoing objects and of the invention, as well as other objects thereof, are set forth in the following specification and illustrated in the accompanying drawings forming a part thereof.

In the drawings:

Fig. 1 is a vertical elevation, partly in section and partly diagrammatic, of an exemplary installation of a hydroelectric unit including a turbine having adjustable blades for purposes of accommodating variations of power requirements from various electrical appliances external to the hydroelectric unit and yet maintain the optimum efficiency of the hydroelectric unit over the operating range. This unit also includes one embodiment of control means which limits the maximum runaway speed of the unit to a predetermined maximum for which the unit has been designed.

Fig. 2 is a fragmentary sectional view on a considerably enlarged scale compared to Fig. 1 showing certain details of the embodiment of control means arranged to maintain the blades against movement except as required during normal operation under the control of the governor.

Fig. 3 is a sectional view similar to Fig. 2 but illustrating only one-half of the mechanism shown therein, said view showing the control mechanism in the position the parts would assume, due to centrifugal force, were the normal speed control mechanism such as the governor and pressure system to fail and thereby permit the unit to reach a predetermined overspeed at which movement of the blades toward open position should be permitted.

Fig. 4 is a still further enlarged sectional fragmentary view of a further embodiment of blade control mechanism utilizing hydraulic principles and operable to permit movement of the blades toward open position upon a predetermined overspeed condition occurring, said condition being evidenced in the normal blade servomotor by an increase in pressure under the piston thereof above a predetermined maximum. In this view, the control means is in a position to maintain the blades against movement except as controlled normally by the governor.

Fig. 5 is a view similar to Fig. 4 except that the control means is positioned to permit relieving of the excess of fluid pressure on the under side of the servomotor piston.

Fig. 6 is a diagrammatic and partially fragmentary as well as partly sectional view illustrating still another embodiment of blade control mechanism responsive directly to rotational speed of the turbine and generator and upon said speed exceeding a predetermined amount in the event of failure of the normal control mechanism, said control mechanism functions to permit the blades of the turbine to move toward open position and thereby reduce the speed of the turbine. This embodiment also utilizes electrical means. In this view, the control mechanism is positioned to maintain the blades of the turbine against movement except as controlled by the functioning of the normal control mechanism.

Fig. 7 is a view similar to Fig. 6 but showing the control mechanism positioned to effect movement of the blades toward open position thereof in response to runaway speed conditions in excess of a predetermined amount.

Fig. 8 is a partly sectional view of an embodiment of control mechanism similar to that shown in Figs. 6 and 7 but including a slightly different type of actuating mechanism.

Fig. 9 is an enlarged sectional view illustrating still another embodiment of blade control means utilizing hydraulically operated means similarly to the embodiment in Figs. 4 and 5 but positioned differently in the unit for greater convenience under certain circumstances. In this figure, an oil head such as illustrated in the upper portion of Fig. 1 is shown, said oil head embodying a fluid pressure relief valve arranged to open automatically when pressures against the under surface of the servomotor piston exceed a predetermined figure as a result of the speed of the turbine exceeding a desired value. In this view the valve is shown in closed position such as when the unit is normally controlled by the governor.

Fig. 10 is a fragmentary vertical elevation partly broken away and partly in section, illustrating the details of still another embodiment of mechanically operable control mechanism responsive to runaway speed conditions of predetermined amount and operable, due to the tendency of the blades to move toward or to the maximum permitted open position upon the occurrence of such conditions, to yield to said tendency of said blades to move so as to permit opening of the blades and effect reduction in the speed of the turbine.

Fig. 11 is a view somewhat similar to Fig. 10 but illustrating a similar type of control mechanism usable in con-

junction with one of the other embodiments of control mechanism illustrated hereinabove to supplement the effectiveness thereof.

Description of adjustable blade turbine with normal controls

In addition to serving the purpose of illustrating one embodiment of the invention, Fig. 1 will first be used as a basis for description of the normal construction and operation of an adjustable blade turbine and the conventional controls therefor. In the exemplary installation shown in said figure, six main elements of such a unit are illustrated. These comprise a diagrammatically illustrated electric generator **10**, a turbine runner **12** of the adjustable blade propeller type which depends from the generator **10** and is rotatably supported to drive the generator, a blade control valve **14**, an oil head **16** which surmounts the generator **10**, a servomotor **18**, and blade actuating mechanism **20**. In this illustration, which is intended solely by way of example, a servomotor **18** has been selected as the means for varying the pitch of the blades **22** of the runner **12** as well as maintaining said blades in any desired operative position between substantially closed position and predetermined, desired, optimum open position in which the blades are at a considerable angle to the closed positions thereof. In Fig. 1 the blades are shown in approximately their mid-travel position in accordance with the particular design and arrangement of runner **12** shown herein. Though this invention is not restricted thereto, the blades move through an angle of approximately 26° in moving between the fullest permitted open position and the closed position thereof.

While not illustrated herein, it will be understood by those familiar with the art that surrounding the runner at a diameter slightly greater than that of the runner blades is a partly spherical, metal housing or discharge ring to contain or restrict the water passing axially through the runner and that the water flow to the runner is controlled by a series of wicket gates each movable about its own axis, but in unison, said axes being uniformly arranged around the axis of rotation of the turbine and said gates being movable between fully closed and fully opened positions for purposes of varying the amount of water directed against the blades of the runner **12**, thereby controlling the force which drives the runner **12** and thereby revolves the rotor of the generator **10**.

The opening of the wicket gates is varied according to the load demand on the generator **10** through the medium of a governor system interconnected with said generator. Such governor is not illustrated in the present drawings but it is believed that anyone skilled in this art is readily familiar with the type of governor referred to. When the wicket gates are adjusted to meet changing load demands on the generator **10**, the pitch of the blades **22** of the runner **12** is adjusted automatically and substantially simultaneously with the gate movement so that at all gate positions, the related blade angle is such as to produce the maximum turbine efficiency. Thus, for example, as the gates open, the blades are pitched toward their maximum open position of steepest angles. Conversely, as the gates move toward closed position, the blades are correspondingly automatically pitched toward their flat or closed position.

When it is desired, by way of further example, to shut down the unit, the wicket gates are closed and the runner blades are moved correspondingly to fully closed position in which the terminal end portions of the blades herein illustrated are disposed substantially within a common plane in the preferred embodiment of the invention although the hub ends of the blades are at an angle to the terminal ends. It will be understood also that the generator **10** and the turbine mechanism for operating the same are supported by customary means within hydroelectric power plants and the runner **12** is supported for rotation within the conventional turbine pit, such supporting structure not being illustrated however.

The actuation of the blades **12** is accomplished, and said blades are also maintained in a desired setting, by positioning means comprising an exemplary blade servomotor **18** which has incorporated therein one embodiment of the present invention. Control and actuation of the blades **22** are achieved through the blade actuating mechanism **20** comprising part of said positioning means. By referring particularly to Fig. 1, it will be seen that each blade **22** has a trunnion **24** rotatably supported within a bearing **26** extending into the hub **28** of the runner **12**. Projecting transversely to the inner end of each trunnion **24** and fixed thereto is a rocker arm **30**. Vertically movable within the hub is an operating rod **32** which extends upwardly through the hub **28** and main shaft **34** to the servomotor piston **38**. The main shaft **34** is attached to the upper end of the hub and to the blade servomotor housing **36** as clearly shown in Fig. 1. The upper end of the operating rod **32** is directly connected to pressure responsive means comprising the blade servomotor piston **38** of the blade servomotor **18**.

Connected to the lower end of operating rod **32** is a crosshead **40**, said crosshead having a plurality of arms corresponding to the number of blades **22**. A link bolt **42** is fixed to each of the arms of the crosshead **40** as clearly shown in Fig. 1. A suitable link **44**, such as a pair of plates, pivotally interconnects the ends of each rocker arm **30** with a link bolt **42** on crosshead **40**. Thus, as the piston **38** moves downward, the blades **22** will be rotated on the trunnion axes by the above described mechanism in blade opening direction. Conversely, as the piston **38** is moved upward, the blades **22** will be rotated on the trunnion axes toward closed position or clockwise as viewed in Fig. 1. A generator shaft **46** is connected to the upper end of the blade servomotor casing **36** and the upper end of said shaft **46** is connected to the rotor, not shown, of the generator **10** so as to drive the same when the runner **12** is rotated.

Surmounting the generator **10** is an oil head **16** referred to hereinabove, the same comprising a casing **48** which is stationarily fixed to the frame of the generator **10**. Supported on the inside of the casing **48** and fixed thereto are two separate hydraulic fluid compartments **50** and **52**. In general, but without restriction thereto, it will be understood herein that oil is the preferred hydraulic fluid. Extending between the compartment **50** and the upper end of piston **38** is an inner oil tube **54**, the upper end of said tube being slidable and rotatable relative to the compartment **50**. A suitable packing box **56** prevents the escape of fluid through the bearing between the compartment **50** and the tube **54** but the upper end of the tube **54** is provided with a plurality of fluid ports **58** through which fluid may pass from the compartment **50** to tube **54** and into a small confining housing **60** connected between the lower end of inner oil tube **54** and the upper surface of piston **38** to which it is rigidly connected. Inner oil tube **54** rotates when the runner **12** is rotated and slides up or down as the piston **38** is moved vertically.

Surrounding the inner oil tube **54** and spaced therefrom is an outer oil tube **62** which communicates at its upper end with compartment **52** and extends downward through the main generator shaft **46** to communicate with the upper end of cavity **64** of the blade servomotor casing **36**. The outer oil tube **62** is rotatable within a packing box **66** at its upper end which prevents the escape of fluid from the compartment **52** except through the space provided between the inner oil tube **54** and the outer oil tube **62**. The outer oil tube **62** rotates with the main shafting but does not move up and down with piston **38**.

Blade control valve 14 is stationarily mounted relative to the generator 10 in close proximity thereto and also in close proximity to, or forming a part of or an appendage to, the governor. Blade control valve 14 comprises a housing 68 to which the ends of a number of fluid conduits are fixed. One conduit 70 extends between valve port 72 of valve 14 and compartment 50 of the oil head. Another conduit 74 extends between valve port 76 of valve 14 and compartment 52 of the oil head. It will thus be seen that by means of conduit 70, compartment 50 of the oil head, and inner oil tube 54, the blade control valve may be hydraulically connected, normally, to the blade servomotor 18 on the lower side of piston 38 and that, by means of conduit 74, compartment 52 and outer oil tube 62, the blade control valve may be hydraulically connected to the blade servomotor 18 on the upper side of piston 38.

Pressure conduit 78 is connected at one end to valve port 80 and the other end thereof is connected to a suitable source of hydraulic pressure such as a pump or air loaded pressure tank, not shown. Valve port 82 communicates with a fluid return conduit 84 which communicates at its other end with a sump tank at atmospheric pressure in accordance with conventional practice. The valve housing 68 has a cylindrical valve sleeve 86 within which a valve piston 88 is slidable axially. The sleeve 86 has suitable conventional openings therein as shown in Fig. 1 which communicate through cored openings or ports in the valve housing.

Valve piston 88 is actuated by a piston rod 90 which extends from the piston 88 into the upper housing 92. One end of floating lever 94 is pivotally connected to the upper end of piston rod 90 and the other end thereof is pivotally connected to the lower end of connecting rod 96. Intermediately of its ends, floating lever 94 is pivotally connected by a pin 98 to cam roller mounting assembly 100. Assembly 100 is provided intermediately of its ends with a guide block 102 slidably mounted for movement between suitable vertical members within the housing 92 as clearly shown in Fig. 1.

Rotatably connected to the lower end of assembly 100 is a cam roller 104, the position of which is controlled by a gate-blade cam 106 which is slidable horizontally relative to the housing 92. When, for example, the wicket gates are moved by the governor, such movement of the gates varies the position of the gate-blade cam 106. For example, as the gates are moving in opening direction the cam 106 will be moved toward the right as viewed in Fig. 1, as indicated by the arrow in said figure, inasmuch as said cam 106 is connected by a suitable link 108 to any convenient movable means on said gates which will effect the desired movement of the cam 106.

Pivotally supported intermediately of its ends by bearing pin 110 which is fixed relative to the generator 10, for example, is a compensating lever 113 which is pivotally connected at its opposite ends respectively to the upper end of connecting rod 96 and the lower end of a second connecting rod 112. The upper end of second connecting rod 112 is pivotally connected to one end of a second compensating lever 114 which is pivoted intermediately of its ends about a pivot pin 116 fixed relative to the upper end of the casing 48 of the oil head 16. The other end of the lever 114 is pivotally connected to a short connecting rod 118, the latter being also pivotally connected at its lower end by a pin 120 to the upper end of inner oil tube 54.

Referring particularly to Figs. 2 and 3 for a better illustration thereof, it will be seen that the confining housing 60, with which the inner oil tube 54 communicates at its lower end, is secured at its perimeter to the upper surface 124 of piston 38. One or more sets of interconnected pairs of fluid ports 126 and 127 extend through the piston 38 for purposes of permitting fluid to pass from the lower end of inner oil tube 54 to the lower compartment 128 within the blade servomotor casing 36.

Under normal conditions, the passages 126 and 127 are open so as freely to permit fluid under pressure to be directed against the lower face or end of piston 38, whereupon movement of the blades toward open position will either be prevented by the oil under pressure within the lower compartment 128 or, if the pressure is sufficient, it will cause the blades to move in closing direction. It will be understood of course that fluid will, under normal circumstances, also be within the cavity 64 in the upper portion of the servomotor casing 36. In general, however, when the turbine is set by blade control valve 14 to maintain the blades 22 at a desired pitch to produce a predetermined speed required by the governor in the control system for the generator 10, the pressure of the hydraulic fluid against the lower end or underface of the piston 38 is positive and adequate to equal or exceed the force tending to urge the piston 38 downward due to the inherent unbalance of the blades 22, as well as the hydraulic forces imposed against said blades to rotate the runner 12.

In the illustration of the valve in Fig. 1, the valve piston 88 is disposed in neutral position whereby pressure through port 80 is nullified due to the valve piston 88 closing the valve ports 72 and 76 from contact by fluid under pressure from valve inlet port 80. The fluid under pressure within the conduits 70 and 74, as well as within the chambers 50 and 52 and the inner and outer oil tubes 54 and 62, will maintain the piston 38 in the position illustrated in Fig. 1 or any other desired position as dictated by the governor. Rotation of the rotor of the generator 10 by the runner 12 will continue at the speed and the load afforded by the pitch of the blades 22 illustrated in Fig. 1, said pitch being dictated by the governor as referred to above.

Assuming, however, that the load requirements of the generator 10 are varied, the gates will either be opened more fully or moved toward closed condition automatically as the situation requires. Such movement will shift the gate-blade cam 106 and cause the valve piston 88 correspondingly to be shifted as a result of vertical movement of the roller 104 engaging the cam 106 and thereby cause movement of floating lever 94. Hydraulic fluid pressure from conduit 78 will then be caused to flow through either conduit 70 or 74 and thence through either inner oil tube or outer oil tube 62 to cause the piston 38 to be moved either upward or downward and thereby correspondingly vary the pitch of the blades 22 as required by the new load condition on the generator.

As the blades 22 are moving about their axes an amount adequate to drive the generator 10 at the rated speed and optimum efficiency under the new load conditions, the compensating mechanism comprising pin 120, levers 113 and 114, connecting rods 96 and 112, and floating lever 94 are actuated by the axial movement of the upper end of inner tube 54 to re-center the valve piston 88 after movement thereof in either direction for the purposes described above. When said piston 88 is so re-centered, no further movement of the blades 22 relative to their pivots can take place.

Assuming, for example, that the generator while operating under any given load at rated speed is suddenly disconnected from its load such as by opening the main switch, the governor normally will be affected in such a manner as to actuate the wicket gates toward the closed position, thus shutting down the unit. Substantially simultaneously, but with a slight lag, the blades would thereby be rotated about their trunnions toward the closed positions thereof. If, however, through some fault in the governor or normal control mechanism or the pressure system, the wicket gates should not move as normally intended, runaway speed of the turbine and generator amounting to as high as three times normal speed will occur and both the turbine and generator must be designed at present on the basis that this maximum runaway speed will occur.

As stated above, by way of example, the usual maximum runaway speed considered for purposes of design is approximately within the range between two and one-half and three times normal speed. Thus, were it possible to provide runaway speed limiting means which would function in such a way and particularly automatically so as to positively substantially limit the amount of maximum runaway speeds occurring under the foregoing circumstances or any other circumstances tending to produce runaway speeds, it will be seen that the cost of generators would be materially reduced, though maintaining the same factors of safety, as compared with the present cost of generators which are usually built to sustain the aforementioned maximum amounts of runaway speeds.

Runaway speed limiting mechanisms

It is the principal purpose of the present invention to provide a number of different embodiments of runaway speed limiting means or mechanisms which have been developed and arranged to function automatically upon overspeeds in excess of a predetermined amount occurring so as to free the blades from the control of the normal positioning means and permit or cause movement of the turbine blades **22** toward their fullest contemplated open position, thereby providing a greater blade opening which produces a lower maximum running speed. This will result in material limitation in runaway speed of the turbine and the generator rotor driven thereby. The need for such runaway speed limiting mechanism, in addition to the normal blade control and positioning mechanism and operable to function independently thereof when overspeeds in excess of a predetermined amount occur, is due to the fact that the blades normally cannot move freely to the fullest permitted open position thereof while controlled by said normal mechanism. For example, when the piston **88** of valve **14** is centered, hydraulic fluid cannot flow from one side of piston **38** of the servomotor to the other nor be relieved to the sump, and the blades **22** will be hydraulically locked against pivotal movement.

Further, under normal speed control of the turbine, and assuming said normal control mechanism is functioning properly, if overspeed should occur, the governor will cause movement of control valve piston **88** to a position to permit flow of hydraulic fluid from the pressure source to lower cavity **128** of the servomotor and from upper cavity **64** thereof to sump, thus closing the blades rather than opening them.

Still further, under normal operation, the governor which controls the operation of the cam **106** to actuate the normal blade control valve **14** includes fly-balls which, upon overspeed occurring, cause the gates to close to reduce or completely shut off the flow of water to the blades of the turbine runner. Thus it will be seen that the normal control mechanism, on disconnection of the unit from its load and as a result of the consequent overspeed, will function to close the wicket gates, and thereby through the gate blade cam and other mechanism illustrated in Fig. 1, for example, also simultaneously close the blades. However, it is the closing of the gates which reduces the turbine speed, not the closing of the blades, since the closing of the blades per se actually would tend to increase the speed. Now, it will be obvious that if the normal control mechanism through a fault fails to close the gates, the blades will be left at whatever position they were in when the original overspeed occurred because control valve **14** is on center and the blades are therefore hydraulically locked, therefore maximum runaway speed of the turbine will occur, possibly as much as three times normal.

The various embodiments described hereafter are not equivalents since the same have particular advantages respectively under various conditions of use and related structure. In all of the preferred embodiments of control mechanism utilizing the present invention, the inherent unbalance of the blades **22** is utilized to insure movement thereof toward the fullest open position of the blades permitted by the actuating mechanism when runaway speeds are sustained by the turbine in excess of a predetermined amount. However, in certain of the embodiments such inherent unbalance, while advantageous, may not be essential.

A common characteristic of all species of runaway speed limiting species to be described hereinafter is that the action thereof is initiated by the turbine reaching a selectable predetermined speed above normal, either directly, or indirectly in response to conditions arising in or affecting the turbine or its controls as a result of said predetermined speed above normal. This predetermined speed above normal may be varied selectively from a relatively small amount to the maximum practical amount. The amount will depend upon the specie of runaway speed limiting mechanism selected for relieving the pressure of the hydraulic fluid on the underside of the blade servomotor. Such relief may be accomplished either by directing the said hydraulic fluid to the sump or by bypassing it to the upper side of the blade servomotor. Such bypassing may be accomplished either directly through the piston or around the same, or between the conduits to and from the servomotor. Various embodiments of the invention are illustrated and described hereinafter to achieve the foregoing objectives.

In the specific embodiment illustrated in Figs. 1 through 3, one type of mechanical runaway speed limiting mechanism is incorporated in the piston **38** of the blade servomotor **18**. The larger scale illustrations in Figs. 2 and 3 best illustrate said embodiment of the invention. It will be seen that this embodiment of runaway speed limiting means comprises one or more slide valves **130** mounted for reciprocable movement radially of the piston **38**. Each slide valve **130** has an integral head **132** reciprocable within a radial cavity **134** of the piston **38**, the outer end of said cavity **134** being closed by threaded plug **136**. Preferably, a compressed spring **138** extends between the plug **136** and the inner end of a suitable cavity extending into the head **132** of the slide valve **130**, as clearly shown in Figs. 2 and 3.

The inner end **140** of the slide valve **130** actually comprises a valve closing means which is arranged, when the slide valve **130** is moved toward the left as viewed in Fig. 2 to the position shown in Fig. 3, to close the fluid passage, consisting of ports **126** and **127**, through the piston **38**. It will be understood of course that the blade servomotor **18** and piston **38** is rotated at the same rotational speed as the runner **12** and centrifugal force will be developed by such rotation. This centrifugal force will be imposed upon each slide valve **130** in opposition to the force of the spring **138** acting against each valve to maintain the slide valve **130** in open position as illustrated in Fig. 2. However, spring **138** is selected so as to have sufficient force to withstand centrifugal forces developed during normal operational speeds of the runner and servomotor but, when overspeeds of a predetermined amount occur, the centrifugal force developed by such speeds and imposed upon each slide valve **130** will be sufficient to overcome the force of each valve spring **138** and thereby force the slide valve **130** to move radially outward to its closed position illustrated in Fig. 3. This thereby closes the fluid passage from the inner fluid tube **54** to the lower side of piston **38** and opens a passageway for fluid from the lower side of the piston to the upper side thereof through ports **127** and **142** as clearly shown in Fig. 3.

Each slide valve **130** has a passage **144** extending axially therethrough to permit the passage of hydraulic fluid or air or a combination thereof freely from one end of the valve to the other, thereby insuring that no impediment is offered to the movement of the valve member **130** by vacuums being established, for example, at either end of the valve member.

It will be remembered that fluid under sufficient pressure

within lower compartment 128 of the blade servomotor 36 and normally forced against the lower side of the piston 38 prevents movement of the blades 22 toward their open positions. Thus, when a passage is provided through the piston 38, the fluid within lower compartment 128 which normally prevents movement of the blades toward the open positions thereof is no longer confined and, upon passing through ports 127 and 142 in the piston 38, permits the inherent unbalance of the blades 22 to move the same toward their fullest permitted open position. Such movement of the blades carries the piston 38 toward the lower end of the servomotor cylinder. Said opening of the blades 22 decreases the speed of the runner 12 as described herein above to that corresponding to the blade angle to which the blades have been moved by said inherent unbalance and, upon the speed being reduced, the spring 138 associated with each slide valve 130 will restore the valve to the normal position thereof shown in Fig. 2. If the condition which orignially caused the speed to exceed said predetermined amount has not been obviated by the time the aforementioned operation has been completed, any succeeding increase in speed above the aforementioned predetermined speed will again open the valve 130 in the piston 38 as described above and the cycle of movements also described above will again occur and the maximum runaway speed will be limited to that corresponding to the angle to which the blades have been moved by their inherent unbalance. The fault in the normal control means which made runaway speed possible might be such that it can not be corrected with the unit rotating. If this is so, the unit can be brought to rest by closing the wicket gates by manual control or by closing the heat gates at the intake.

There will be no relative rotation between the piston 38 and the cylinder casing 36 of the servomotor, although there will be relative sliding movement in a direction axially of said cylinder. Thus, if desired, suitable openings may be formed in the wall of the cylinder casing 36 for purposes of permitting access to the valve 130. Such openings are not shown in the drawings but it will be readily understood that some appropriate form of opening may readily be made in the casing wall, said opening being closed by an appropriate pressure resisting closure. Preferably, one such opening should be provided in the cylinder wall for each slide valve 130. It will be noted that the slide valves 130 and springs 138 are so constructed that they may readily be withdrawn radially outward from the piston 38 when the plug 136 is removed.

It will also be noted that Fig. 2 illustrates one valve member 130 in full lines while a second one, in the right hand portion of Fig. 2, is illustrated in phantom, it being understood that any desired number of such valves may be used in conjunction with a corresponding number of sets of ports 126, 127 and 142 through the piston 38. In the preferred construction, the valves 130 are arranged, when a plurality are used, so as to balance each other relative to the piston 38 and thereby minimize any out-of-balance effect which might otherwise be caused by use of such valves.

The above described pressure by-passing mechanism used for purposes of permitting the blades to open at speeds above a predetermined amount will function regardless of the position of the control valve 14 as the hydraulic fluid need not flow through said valve in either direction during the functioning of the above described fluid pressure by-passing means in the piston 38. For example, assume that the valve piston 88 is in a position such that hydraulic fluid under pressure from conduit 78 can enter the lower compartment 128 of the blade servomotor casing 36 so as to be disposed against the lower end of the piston 38 and the upper cavity 64 of the casing 36 is connected to the fluid return conduit 84 for passage to sump. Assume further that the valve piston 88 would stick in this position. When a speed in excess of said predetermined amount occurs, such sticking of the valve piston 88 would not in any way affect by-passing of the oil through the servomotor piston 38 because the slide vales 130 are free to move irrespective of the setting and effect of the control valve 14 and thereby effect passage of hydraulic fluid through the piston 38, thus permitting the piston to lower and the blades 22 to move to or toward open position due to the inherent unbalance thereof.

Conversely, if the upper end of the piston were subjected to full oil pressure and the lower end of the piston were connected to sump, the blades of the runner would naturally move to full open position even if no oil by-pass means is installed in piston 38. If such by-pass means is installed in said piston, however, the hydraulic fluid pressure would immediately tend to be equalized in cavities 64 and 128, thus permitting the blades to open due to their inherent unbalance, regardless of what the nature of the fault in the normal control mechanism may be.

Displaced oil in lower cavity 128 will pass to upper cavity 54 in accordance with the downward movement of piston 38, it being understood that displacement above and below the piston for any given piston movement will be equal or any difference readily may be compensated for by appropriate details in design common to the art. It is obvious that the exact position of the centrifugally operated valve is immaterial as long as it is in a location to be responsive to the speed of the turbine.

Another embodiment of runaway speed control mechanism is illustrated in Figs. 4 and 5, said means being directly associated with blade servomotor 18 in the specific illustration thereof. This embodiment is of the type responsive to excessive hydraulic pressure developed as a result of overspeeds in excess of a predetermined amount tending to move the blades 22 toward open position by reason of the inherent hydraulic unbalance thereof. Such movement is prevented at normal speeds by hydraulic pressure against the lower side of the piston 38 but excessive overspeeds produce abnormal pressures within the lower cavity 128 of the servomotor casing 36.

In Fig. 4, the mechanism is illustrated in normal operating condition. In this figure, it will be seen that a by-pass conduit is connected at its opposite ends respectively to the upper cavity 64 in the upper end of cylinder casing 36 and the lower cavity 128 in said cylinder casing. Connected in said by-pass conduit 146 is a pressure relief valve 143 including a compressible spring 150 acting against a valve piston 152 which co-acts with valve seat 154. Valve piston 152 is provided with suitable guide fins 156 to insure axial movement of the valve piston.

Under conditions where overspeeds occur in excess of a predetermined amount, for example, about 75% in excess of normal speed, the excessive pressure developed within the lower cavity 128 will react against valve piston 152, which is normally closed against the valve seat 154, and force said piston 152 to open position thereof, illustrated in Fig. 5, compressing spring 150 as a result. It will be understood of course that the spring 150 is suitably adjusted as to pressure by the adjustable threaded plug 158 to permit the same to be compressed and thereby effect opening movement of piston 152 upon a predetermined amount of pressure being exerted against the piston 152 from cavity 128.

As will readily be seen from Fig. 5, when the valve piston 152 is moved to open position, fluid within cavity 128 will flow through the valve opening and the by-pass conduit 146 into upper cavity 64, taking up the space vacated by reason of the piston 38 having moved toward the runner. In any case this will permit the piston 38 rapidly to lower due to the inherent unbalance of the blades 22 tending to pull the piston 38 downward.

As in regard to the embodiment shown in Figs. 1 through 3, it will be understood that, for a given piston

movement within the servomotor casing, the increase in fluid volume in upper cavity **64** must be at least equal to the decrease in lower cavity **128**. Any difference readily may be compensated for by appropriate details in design well known in the art.

Upon the blades opening as the result of said tendency toward fullest permitted open position and in due course reaching same, the speed of the turbine and generator will be reduced and such opening of the blades will be possible regardless of the setting of the control valve **14** or any faulty operation of the normal control mechanism.

When, by reason of said reduction in speed, the pressure against valve piston **152** is reduced so as to be less than the force exerted by spring **150**, said spring will move the valve piston **152** to its closed position shown in Fig. 4 and the turbine will then be in condition to function normally as dictated by the control valve **14** and the governor in the control system as described herein above. Suitable locking means such as a nut **160** may be used to secure the threaded plug **158** in desired calibrated position and the outer end of the threaded plug covered by a cap **162**. The compression of spring **150** is initially set such that the valve piston **152** can not open unless the pressure in cavity **128** exceeds the normal maximum pressure that might be required to pitch the blades while the unit is operating at rated speed or slightly above.

Notches **164** may be provided in the walls of piston **38** to prevent obstruction of the outlet of oil conduit **146** when details of construction so require. Also, while in Fig. 5 the piston **38** is shown at the fullest extent of its normal travel in normal opening direction, it will be understood that in cases where it is desirable to have the blades go to a greater pitch than normal upon runaway speed occurring, the casing **36** of the servomotor **18** would be made appropriately longer to permit such greater movement of the piston.

Still another embodiment of runaway speed limiting mechanism incorporating the principles of the present invention is illustrated in Figs. 6 and 7. This embodiment utilizes electrical means to interconnect certain of the elements of the control system during normal speed operation of the turbine and generator. However, upon overspeed conditions in excess of a predetermined amount occurring, the circuit of electrical actuating mechanism is broken in response to said overspeed conditions, whereupon the blade control mechanism, comprising essentially a by-passing valve, is actuated in such a manner as to effect movement of the blades toward open position and thereby limit the maximum runaway speed of the turbine and generator to that corresponding to the open position to which the blades have been moved by the aforementioned inherent unbalance thereof.

Referring to Fig. 6 particularly, the control valve **14** is illustrated in larger scale than in Fig. 1 and fragmentarily. Further, it will be understood that the control valve controls the flow of hydraulic fluid through conduits similar to conduits **70** and **74**, illustrated in Fig. 1, to the inner and outer oil tubes **54** and **62**, the latter not being illustrated in Fig. 6. In the embodiment shown in Fig. 6, the lever and linkage system illustrated in the upper portion of the figure corresponds substantially to similar mechanism shown in Fig. 1 and the functioning of both linkage and lever mechanisms is similar. In the embodiments shown in Figs. 1 through 5 hydraulic pressure developed during overspeed conditions in excess of a predetermined amount is by-passed through or around the piston **38**, thereby tending to balance the pressure in cavities **64** and **128** for purposes of permitting the blades **22** to move toward or to fullest permitted open position. The embodiment shown in Fig. 6 however does not require the use of such pressure by-pass means through or around the piston of the servomotor **18**. This embodiment affords a complete relief to atmospheric pressure of the lower cavity **128**, said relief being instigated by an electrical centrifugal switch responsive to the R. P. M. of the unit.

In the preferred structure illustrated in Figs. 6 and 7, an electrical centrifugal switch **166**, calibrated to be responsive to generator R. P. M. in excess of a predetermined amount, is driven by the hydroelectric unit, preferably mechanically, so as to be affected by said speed. The switch **166** and the circuit thereof is only diagrammatically shown in Fig. 6. However, said circuit contains contacts so as to interconnect the switch **166** in a circuit comprising a source of current **168** and the magnets of a solenoid type clutch **170**.

As shown in Figs. 6 and 7, the clutch **170** comprises a plurality of electro magnets **172**, the windings of which are electrically connected with each other and with a source of current **168**. Pivoted latches **174** are arranged so that one end of each latch, when held by magnetic force against the magnets **172**, is disposed within an annular groove **176** formed in the lower end of a vertically movable piston rod extension **178**, pivotally connected at its upper end to floating lever **94**. As is clearly shown from Fig. 6, a solenoid clutch weight **177** is fixed to the upper end of piston rod **90** and it will be noted that said weight is relatively massive so as to be capable of readily moving the valve piston **88** downward by gravity when the weight **177** is disconnected from piston rod extension **178**.

The centrifugal electric switch **166** is preferably of the adjustable type and responsive to any selected rotational speed of the hydroelectric unit within an appropriate range. It is contemplated that said switch may be set to open upon a predetermined speed being developed by the turbine and generator **10**. Upon said predetermined speed being attained, which is in excess of normal speed, the switch **166** will open and thereby break the circuit to the electromagnets **172**. Deenergizing said magnets will result in the weight **177** quickly moving the latches **174** about their pivot pins **175** to permit disconnection thereof from extension **178**, thereby permitting the weight **177** to move the valve piston **88** to the lower position thereof shown in Fig. 7. Such movement of the valve piston downward is arrested by suitable stop means such as blocks **179** which directly engage the undersurface of weight **177** as shown in Fig. 7.

When the turbine and generator rotor are rotating at normal speeds, or up to the speed for which the switch is set, extension **178** and piston rod **90** will be interconnected by the solenoid clutch **170** so as to maintain the valve under direct control by the linkage and lever system illustrated in Fig. 6, the same being primarily directed by the governor through cam **106**. As described above in regard to the embodiments shown in Figs. 1 through 3, the position of cam **106** normally is varied in accordance with movement of the wicket gates so as correspondingly to vary the pitch of the blades **22**. Thus, under normal operating conditions within normal speed ranges, i. e., from zero speed up to approximately 50% above normal speed, for example, clutch **170** will maintain the valve piston **88** connected to the linkage and lever control mechanism for normal operation thereof by the governor through cam **106**. Under these normal conditions when the blades **22** are set at a certain pitch corresponding to a desired speed and load on the turbine, the valve piston **88** will be maintained in neutral position illustrated in Fig. 6 or will otherwise function as dictated by the governor. Hydraulic pressure from conduit **78** will be held inactive or be routed as dictated by the governor. When the valve piston **88** is in neutral position as in Fig. 6, it closes the ports **72** and **76** in the control valve **14**, whereby servomotor piston **38** has hydraulic fluid at both ends thereof serving to hold the piston **38** against movement and thereby preventing the pitch of blades **22** from changing.

Upon an overspeed of a predetermined amount occurring from any of a number of causes of abnormal nature, the centrifugal switch **166** will function to disconnect the circuit of the solenoid clutch **170** as described above, whereupon weight **177** will force the valve piston **88** to move to the lower position thereof shown in Fig. 7. Lowering valve piston **88** to the position shown in Fig. 7 connects lower cavity **128** in blade servomotor casing **36** to the sump tank of the hydraulic pressure system, by means of holes through piston **38**, confining housing **60**, inner oil tube **54**, oil head fluid compartment **50**, conduit **70**, port **72**, longitudinal ports **180** in piston **88**, port **82**, and conduit **84**. Since the sump tank is open to atmosphere and is at an appreciably lower elevation than blade servomotor **36**, downward movement of piston **38** is facilitated.

There being a positive inherent unbalance of the runner blades **22** in opening direction at overspeeds, said blades will positively open to fullest permitted extent quickly by reason of the relief of pressure on the underside of piston **38**. This will positively result, regardless of what fault occurred in the normal control mechanism which permitted the overspeed initially to occur.

Rotation of the turbine will continue at speeds less than the predetermined overspeed for which electrical switch **166** was set until such time as the water is shut off from the turbine by some means such as a head gate. Thereupon the fault may be remedied, solenoid clutch **170** restored to interconnect with valve piston extension **178**, solenoid **172** re-energized, and all other conditions restored to normal and the unit restarted.

The mechanism illustrated in Fig. 8 is identical in purpose to that shown in Figs. 6 and 7, i. e. upon runaway speed in excess of a predetermined maximum, it will effect a complete relief to atmospheric pressure of the lower cavity **128**, said relief being actuated by an electrical centrifugal switch responsive to the R. P. M. of the hydroelectric unit. The difference between the mechanism in Fig. 8, and Figs. 6 and 7 is only in the nature of the respective gate-blade cams. Both types of cam are commonly used in the normal control mechanism of adjustable blade propeller turbines and the reason for illustrating them both is merely to show that the present invention, being an addition to the normal control mechanism, may be applied regardless of which type of cam is used.

The mechanism illustrated in Fig. 8 is similar to that shown in Figs. 6 and 7, except that in the mechanism shown in Figs. 6 and 7, the cam **105** is linearly actuated by the governor in unison with movement of the wicket gates. Said cam **182** may be rotated by rotation of cam shaft **183** about fixed center **184**, as dictated by the governor by means of lever **185** and connecting rod **186** or as otherwise convenient.

The normal operation of the control mechanism, i. e. at normal rotative speeds of the hydroelectric unit below a predetermined overspeed, is similar to that described herein relative to Figs. 6 and 7, said description being modified only in regard to the manner in which rotation of cam **182** actuates piston rod extension **178** which is as follows:

When, for example, a load increase upon the generator occurs, the governor mechanism rotates cam **182** clockwise referring to Fig. 8. This forces down cam roller **188**, which is rotatably connected with floating lever **94**, pivoting about the pin connecting its outer end to connecting rod **96**, shown in Fig. 6, and also forcing piston rod extension **178** down, thus causing the blades **22** to be forced in an opening direction by an amount proportional to the original load increase.

The compensation mechanism, similar to that shown in Figs. 1 and 6, functions in response to actual blade movement to move connecting rod **96** down, floating lever **94** pivots about the axis of cam roller **188**, thus raising connecting rod extension **178**, the engaged solenoid clutch **170**, connecting rod **90** and valve piston **88**, recentering the same with valve port **80** and thereby holding the runner blades **22** in the position corresponding to optimum efficiency of the unit at the new load. Compressible spring **189** functions merely to hold cam roller **188** against cam **182** at all times, said spring being supported between upper spring cap **190** supporting the cam roller and the lower spring cap **191** which pivotally rests upon or against a fixed support **192**. However, if the unit is disconnected from its load and the normal control mechanism of the hydroelectric unit fails to function properly, resulting in overspeed of an amount in excess of predetermined maximum for which electrical centrifugal switch **166** is set, then the blades **22** will go to the maximum permitted full open position in the same manner and for the same reasons and purposes as herein described in connection with Figs. 6 and 7.

In the foregoing description pertaining particularly to Figs. 6, 7 and 8, electrical switch **166** is described as being of the "centrifugal" type. This type is well known in the art as having mechanism rotatable at a speed proportional to, in this instance, the speed of the hydroelectric unit, and containing electric contacts which either open or close, whichever may be more desirable, in response to certain degress of centrifugal force induced by the rotation of the switch elements by the machine driving said electrical centrifugal switch.

As another related species of this embodiment of the invention, said electrical switch **166** could be of a type reactive, for example, to hydraulic fluid pressure, i. e. an electrical pressure switch **167**. Switch **167** is such that its contacts open upon pressure in any part of the pressure system, which is hydraulically connected at all times with the lower cavity **128** in casing **36**, exceeding a predetermined maximum value, thereby relieving to sump any pressure holding the blades from moving in opening direction. Considering Figs. 6, 7 and 8 and the descriptions relating thereto to be the same as above but substituting electrical pressure switch **167** for electrical centrifugal switch **166**, it will be seen that the remainder of the equipment will function similarly and with a similar end result as when centrifugal electrical switch **166** is used to control the clutch circuit.

Regarding the embodiment shown in Figs. 6 through 8, it will be evident that release of pressure under the servomotor to sump as a result of movement of the control valve will cause oil to go to sump by gravity from under the servomotor piston. The weights of the piston, the operating rod, and the crosshead, all operating downward by gravity, as well as the weight of oil above the piston and some pressure of the oil above the piston, when pressure under the servomotor is released to sump, will all aid in effecting downward movement of the crosshead. Thus, in the embodiments shown in Figs. 6 throughout 8, the blades **22** of the runner **12** may be balanced for movement about their axes, rather than inherently unbalanced toward open position as in certain of the other embodiments shown and described herein, and the blades will nevertheless be caused to move to open position to reduce the runaway speed of the unit.

Another embodiment of runaway speed limiting mechanism of this invention is illustrated in Fig. 9, this embodiment being similar to that shown in Figs. 4 and 5 in that the hydraulic blade operating fluid is by-passed from the high pressure to low pressure passages at a different location, namely in the oil head **16** rather than at the blade servomotor **18**.

Regarding the embodiment shown in Fig. 9 and, as described above in regard to Fig. 1, hydraulic fluid compartments **50** and **52** in the oil head **16** are separated from each other and respectively communicate with the inner oil tube **54** and outer oil tube **62**. These compartments are separated by a partition **193**. In order to by-pass excessive hydraulic fluid pressure developed during overspeed conditions in such manner as to permit the blades

to move toward the maximum permitted open positions thereof, it is only necessary to hydraulically interconnect lower cavity **128** and upper cavity **64** of blade servomotor casing **36**. This may be accomplished by any of a number of means, in response to excessive pressure in lower cavity **128**, one means being as illustrated in Figs. 4 and 5, another, which is directly responsive to excessive rotational speed of the turbine, is shown in Figs. 1, 2 and 3, and still another, shown in Fig. 9, essentially comprises a by-pass between conduits **70** and **74** at a convenient location.

When lower cavity **128** is subjected to excessive fluid pressure developed during overspeed conditions above a predetermined limit, upper compartment **50** of the oil head will be subject to the same pressure. Of course, when the generator and turbine are operating at normal rated speeds, there will be no excessive pressure anywhere in the system.

The fluid by-passing means of the embodiment shown in Fig. 9 comprises a pressure responsive valve assembly **194**. In the specific embodiment illustrated, a valve seat ring **196** is fixed within a suitable opening formed in the partition **193**. A valve seat **198**, preferably spherical, is formed in the valve seat ring **196** and a slidable valve piston **200** has a complimentary surface **202** which conforms to the valve seat **198**. The seat ring **196** also preferably has a bearing sleeve **204** integrally secured thereto by radial ribs **206** between which are ports **207**. The valve piston **200** has a bearing surface **208** which slides on the bearing sleeve **204** to guide its axial movement. Bearing sleeve **204** also slidably receives the inner oil tube **54** as clearly shown in Fig. 9 and permits axial as well as rotatable movement of said oil tube relative to the bearing sleeve **204**. Valve piston **200** normally is forced upward by a compression spring **210**, the lower end of said spring resting against a relatively fixed seat **212**. The spring **210** is selected so as to exert a maximum force substantially equal to the force upon valve piston **200** from upper compartment **50** that would occur at a predetermined overspeed.

When the turbine and generator are operating at normal speeds the valve piston **200** will be maintained in closed position by the action of spring **210**. Under these circumstances, the control mechanism and oil flow throughout the pressure system will function just as though the pressure responsive valve assembly were not present, i, e., just as though the oil head were constructed as shown in Fig. 1, said functioning being well known in the art. However, should the electrical load on the generator suddenly be disconnected from the same and if, for any reason, runaway speed occurs in excess of a predetermined amount, the inherent unbalance of the blades pulls down abnormally upon piston **38**, creating excessive pressure in lower cavity **128** and upper compartment **50**.

When the pressure in upper compartment **50** increases above a predetermined limit, which corresponds also to the calibrated setting of the spring **210**, the valve piston **200** will be separated from the valve seat ring **196**, thereby permitting fluid under excess pressure from compartment **50** to flow into compartment **52** and from thence through outer oil tube **62** to upper cavity **64** of the servomotor casing **36**. Thus a hydraulic circuit or passage will be provided for the hydraulic fluid from the underside of piston **38** to the upper side, resulting in the turbine blades **22** being able to move in an opening direction by reason of their inherent unbalance at higher than normal rotational speeds. The speed of the turbine and generator will thereby be prevented from exceeding a predetermined limit.

Following such reduction in speed of the turbine and generator, it will be understood that since the excessive fluid pressure in upper compartment **50** has been reduced below the limit at which the valve piston **200** opens, said valve will close so as again to seal compartments **50** and **52** from communication with each other. The above described cycle may reoccur but at no time will the speed of the unit exceed a predetermined limit. If necessary, the water may be shut off from the turbine by means of a head gate or otherwise, to locate and correct the fault in the normal control mechanism which originally permitted the overspeed, and the unit may then be re-started.

One further embodiment of runaway speed limiting mechanism, utilizing mechanical principles, is illustrated in Fig. 10. In this embodiment, no other type of runaway speed limiting mechanism such as herein before described is required and either a conventional blade servomotor **18**, with the customary type of piston therein may be utilized, or a geared electric motor or any other suitable device may be used to pitch the runner blades **22** under normal conditions of operation. When runaway speed conditions occur, as described above in regard to the various other embodiments of the invention, the same inherent unbalance of the blades as well as the unbalance of the hydraulic force imposed upon said blades also exists and tends strongly to move the blades toward their fullest permitted open position but the fluid pressure acting against the lower end of the servomotor piston or mechanical leverage of the gears in case of electric motor drive normally prevents such movement. In the case of the servomotor type of blade operation, the hydraulic pressure acting against said lower end of the piston likewise prevents downward movement of the piston rod **32** and the cross-head **40**. The links **44** and the link bolts **42**, which interconnect the cross-head **40** and the rocker arms **30**, serve to maintain the blades at the pitch required by the governor.

Referring to Fig. 1, under normal operating conditions the force of the operating mechanism when increasing the blade pitch results in tension in link bolts **42** and links **44**. However, if the cross-head **40** is held stationary in an axial direction, the blades, due to their inherent tendency to open upon overspeed, endeavor to open and the link bolts **42** and links **44** are placed in compression. If, however, the blades **22** shown in Fig. 1 were arranged oppositely to the crank arms **30** thereof, as shown in Fig. 10, the link bolts **42** and links **44** would then be under tension as the result of the unbalance of the blades and other forces tending to move the blades toward a greater open position under runaway speed conditions. Thus, the latter arrangement is used in the embodiment shown in Fig. 10 for purposes of providing a runaway speed limiting means of a mechanical nature.

Whereas in the structure shown in Fig. 1 the link bolts **42** may be secured to the ends of the crosshead **40** by threaded nuts, the embodiment shown in Fig. 10 utilizes a shear pin **214** to secure each link bolt **42** to its arm of the cross-head **40**. The shear pins **214** are of such size as to fracture in shear upon a predetermined force being exerted against said pins. Said predetermined force will be that at which it is desired that the blades **22** may be freed for movement to fullest permitted open positions upon a runaway speed of predetermined amount occurring, such as to subject the blades **22** to a corresponding force tending to move said blades toward their fullest permitted open positions.

The shear pins **214** are adequate to resist operating force imposed upon the blades **22** at normal speeds. However, when a predetermined overspeed occurs the pins **214** are sheared and it will be understood that the blades **22** will then move incident to the shearing of the pins **214** regardless of the fact that the operating rod **32** remains stationary.

Obviously with this arrangement of the rocker arms **30** relative to the blades **22**, the normal control mechanism for pitching the blades would be arranged so that pressure oil would be admitted to the underside of piston **38**, i. e. to lower cavity **128**, and upper cavity **64** would be connected to the sump when opening movement of

the blades is required. That is, the cross-head would go up to open the blades and down to close them in normal operation, rather than vice versa.

Furthermore, it will be apparent that if one runner blade were in some way stuck such as by foreign matter being lodged, for example, between a blade and the runner hub **28**, so that it could not open in response to runaway speed exceeding a predetermined amount, the remaining blades **22** of the runner which are not so stuck could go open, thus still preventing any degree of overspeed sufficient to endanger the safety of the unit.

In order to restore the turbine runner blades to normal operating condition so as again to be controlled by the servomotor **18** following shearing of the pins **214**, it will of course be necessary to reconnect the link bolts **42** to the arms of the crosshead by replacing the fractured shear pins **214**. This can readily be done after the turbine has been unwatered and stopped, by mechanics removing the runner cone **216** from the runner **12** so as to gain access to the crosshead **40**.

Another embodiment of the shear pin principle is shown in Fig. 11, and may be incorporated with the various other runaway speed control embodiments illustrated in Figs. 1 through 9 and described hereinabove. However, such shear pin principle is embodiable with said other runaway speed control means for a different purpose than that utilized in the embodiment of Fig. 10. For example, in Fig. 11, wherein the additional shear pin principle is to be used in conjunction with other runaway speed limiting means, it will be noted that the exemplary blade **22** is arranged relative to the rocker arm **30** oppositely to the arrangement in Fig. 10 but the same as in the arrangement shown in Fig. 1. The arrangement of blades and rocker arms shown in Fig. 1 is the same in the various other embodiments of runaway speed limiting means described and illustrated herein with the exception of the embodiment shown in Fig. 10.

Referring to Fig. 11, it is therefore assumed that one of the other runaway speed control means is utilized in the turbine-generator unit such as illustrated in Figs. 1 through 9. Upon the occurrence of runaway speed conditions in excess of a predetermined limit, the aforementioned other runaway speed limiting means will function to permit the piston **38** of the servomotor **18** to be lowered by reason of the inherent unbalance of the blades **22** and correspondingly to permit or effect movement of the blades **22** either toward or to the fullest permitted open positions thereof so as to reduce or limit the speed of the turbine and generator.

If the blades are not stuck by reason of foreign matter being lodged between the blades and some member stationary relative to blade pitching then, under normal speeds and with the normal control mechanism properly functioning and also under runaway speeds in excess of a predetermined maximum permissible limit and with the normal control mechanism not properly functioning, the blades will be pitchable in response to either condition, as required. However, debris of various kinds floating in the head water impounded by a dam will sometimes escape the customary trash racks and enter the spaces between the wicket gates and contact the blades of the runner. In fact, if debris of this nature should become lodged relative to several of the wicket gates so as to obstruct movement thereof when closing of the gates is required due to release of the load from the turbine, runaway speed may result from such condition.

Another deleterious effect may occur from the possibility of engagement of debris articles with the runner **22**. Said possibility arises from the fact that the inner ends of the blades **22**, adjacent the hub **28**, are sometimes so shaped that during movement of the blades relative to the hub between maximum permitted open and closed positions, spaces occur between said inner ends of the blades and the exterior surface of the hub. Should debris, such as a piece of wood, be trapped in such a space when, for example, the blades are disposed at a pitch intermediate of their maximum open and closed positions, and runaway speed conditions occur which because of one of the aforesaid embodiments cause the piston **38** of the servomotor to lower in an effort to permit or cause movement of the blades toward or to their maximum permitted open positions, such lodgement of the debris may prevent the particular blade on which the debris is caught from moving toward open position.

Inasmuch as the crosshead **40** is connected commonly to all of the blades, the wedging of one blade so as to prevent movement would correspondingly prevent movement of all the blades toward open position. Thus, even though the runaway speed conditions demand movement of the blades toward full permitted open position, the functioning of the runaway speed limiting means would be nullified under the foregoing circumstances wherein one of the blades was wedged against moving to such open position and it is remotely conceivable that excessive runaway speeds then could continue with corresponding harm to the generator or turbine, or both.

To obviate the foregoing possible nullifying of the effect of the runaway speed limiting means, the actuating means for the blades **22** may be provided with fracturable shear pins **214** clearly shown in Fig. 11 which extend through the crosshead **40** and the lower ends of the link bolts **42**. Upon runaway speed conditions occurring and the runaway speed limiting means functioning to permit the crosshead **40** to move downward, all of the blades **22** would normally move to the fullest permitted open position. However, if one of the blades had a piece of debris wedged, for example, between it and the hub of the runner so as to prevent such movement of the blade toward its fullest permitted open position, the resistance of said blade to movement would result in shear pin **214** shown in Fig. 11 being fractured so as to separate said blade from control by the crosshead. The crosshead **40** would thereupon be free to continue its downward movement and the rest of the blades could move to the fullest permitted open positions thereof. Of course, the wedge blade presumably will remain in its wedged position but the movement of the remaining blades to their maximum permitted open positions, for example, will insure a reduction in the runaway speed of the turbine to a value below the predetermined limit and thereby prevent damage being sustained by either the turbine or the generator.

As described in regard to the embodiment of Fig. 10, after a shear pin utilized in the embodiment of Fig. 11 has been fractured, it will of course be necessary to shut down the unit for purposes of replacing the sheared or fractured pin **214**. However, it will be necessary to shut down the unit any way for purposes of removing the wedged article of debris from the blade so as to restore the blade to operative condition relative to the hub **28** and to correct the fault in the normal control mechanism which caused the runaway speed initially.

Summary

From the foregoing, it will be seen that the present invention provides a number of various embodiments of runaway speed limiting means which will function automatically to positively prevent runaway speeds in excess of a predetermined amount occurring in an adjustable blade propeller turbine and generator unit, said amount being, for example, 50% to 100% above normal speed, depending upon the embodiment used, instead of up to 200% above normal, which the turbine would be capable of producing were such embodiments not added to the presently used means of normal control. These various embodiments utilize respectively mechanical, hydraulic, and electrical principles to insure movement of the blades **22** toward or to the fullest permitted open positions thereof due to the hydraulic unbalance of the blades at speeds above normal. Such movement will cause the maximum

runaway speed of the turbine to be limited to amounts for which it and the generator may be more economically designed in anticipation thereof.

As a result of incorporating any of the above described and herein claimed runaway speed limiting means and mechanisms in a turbine-generator unit, the cost of manufacturing such turbine generator units is materially reduced as compared with the cost thereof in accordance with present practice wherein the same are usually designed to withstand runaway speeds of the order of two and one-half to three times normal operating speeds. Further, and at least equally important, is the fact that the present invention provides means to insure against possible destruction of mechanism frequently worth millions of dollars at a greatly reduced cost. Such destruction where it has occasionally occurred in the past has in most cases been also accompanied by loss of human lives.

As stated hereinabove under the title "Runaway Speed Limiting Mechanisms," upon overspeed occurring, when the normal control mechanism is functioning properly, the blades will be moved toward or to closed position, whereas the function of the runaway speed limiting devices described and claimed herein is to permit the inherent hydraulic unbalance of the blades to cause movement thereof toward or to the maximum permitted open position thereof. Also, whereas normal blade movement may be from 6° to 32° relative to a plane normal to the axis of rotation of the turbine, when runaway speed occurs, pitching of the blades within a range between 6° to 50° may be desirable in some instances. In these instances appropriate provision would be made in the operating mechanism to permit the additional abnormal travel.

As used hereinabove and in the appended claims, the terms "runaway speed" and "overspeed" in some instances may seem to be synonymous. However it is intended that "runaway speed" be considered as the maximum speed for which the unit must be designed, whereas "overspeed" is considered to be any speed above the normal rated speed of the unit and up to said "runaway speed." Therefore, overspeed is permissible and is used to actuate the runaway speed limiting devices and may be utilized whether the normal controls have functioned or not.

Though a number of embodiments of runaway speed control devices have been illustrated and described herein, the same are not considered equivalents in that they vary as to applicability according to the physical structure and size of the turbine and to the percentage of overspeed at which they will function.

While the invention has been shown and illustrated in its several preferred embodiments, and has included certain details, it should be understood that the invention is not to be limited to the precise details herein illustrated and described since the same may be carried out in other ways falling within the scope of the invention as claimed.

I claim:

1. A hydraulic turbine comprising a runner having pivotally adjustable blades thereon, the blades and pivots thereof being so connected that said blades are hydraulically unbalanced for movement toward the open positions of said blades at speeds above normal, and normal positioning means utilizing hydraulic pressure which varies in proportion to the speed of the turbine, said means being interconnected with said blades and operable normally to maintain said blades in desired positions of operation and vary the same to produce a desired turbine output at normal speed, in combination with runaway speed limiting mechanism interconnected with said normal positioning means and including mechanism responsive to the hydraulic pressure thereof and operable upon said pressure attaining a predetermined amount at speeds of the turbine above normal to free the blades from the control of said normal positioning means and permit the said hydraulic unbalance of said blades to move the same toward said open positions thereof, thereby limiting the runaway speed of the turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

2. A hydraulic turbine comprising a runner having pivotally adjustable blades thereon, the blades and pivots thereof being so connected that said blades are hydraulically unbalanced for movement toward the open positions of said blades at speeds above normal, and positioning means interconnected with said blades and operable normally to maintain said blades in desired positions of operation and vary the same to produce a desired turbine output at normal speed, in combination with runaway speed limiting mechanism including electrical mechanism operable in response to a predetermined speed condition in excess of the normal speed of the turbine and interconnected with said normal positioning means and operable automatically at speeds of a predetermined amount above normal to free the blades from the control of said normal positioning means and permit the said hydraulic unbalance of said blades to move the same toward said open positions thereof, thereby limiting the runaway speed of the turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

3. The turbine structure set forth in claim 2 further including a speed responsive switch in said electrical mechanism and said switch being responsive to the rotational speed thereof and operative upon said speed exceeding a predetermined amount above normal to cause movement of said blades toward said open positions thereof and thereby limit the runaway speed of said turbine to the value aforesaid.

4. The turbine structure set forth in claim 2 further including a pressure responsive switch in said electrical mechanism and further characterized by said control means utilizing a hydraulic system conducting hydraulic fluid under pressure, said switch being interconnected with said hydraulic system and responsive to changes in the pressure therein, said switch being operable upon said pressure increasing a predetermined amount above normal to cause movement of said blades toward said open positions thereof and thereby limit the runaway speed of said turbine to the value aforesaid.

5. A hydraulic turbine including an adjustable blade runner supported for rotation about its axis and comprising a hub, a plurality of blades pivotally supported by said hub and said blades and pivots being so connected that said blades are hydraulically unbalanced for movement toward the open position of said blades at speeds above normal, and blade positioning means including a hydraulically operated servomotor having a piston interconnected to said blades and hydraulic pressure and return conduits interconnected between a source of fluid pressure and said servomotor, said positioning means also including a control valve connected between said conduits and operable to direct pressure against either end of said piston selectively to vary the positions of said blades and maintain them in a substantial range of operative positions, in combination with pressure relief means interconnected with said blade positioning means and upon said turbine exceeding a predetermined speed and developing hydraulic pressure in said system in excess of normal blade control pressures said pressure relief means being operable to direct the hydraulic fluid under pressure from the end of said piston functioning to hold said blades against said movement to the return conduit, thereby to permit movement of said piston and blades toward said open positions thereof and limit the runaway speed of said turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

6. The hydraulic turbine structure of claim 5 additionally including an oil head mounted above said servomotor and interconnected with said hydraulic pressure and

return conduit means, and said pressure relief means in said blade positioning means comprising a pressure responsive valve mounted on said oil head, whereby upon the development of pressure in excess of a predetermined amount in the conduit connected to the end of the servomotor holding the blades against movement in the opening direction, said valve will open and pass fluid from the said end of said servomotor to the opposite end thereof, thus permitting said blades to move toward said open positions thereof to limit the runaway speed of said turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

7. A hydraulic turbine including an adjustable blade runner supported for rotation about its axis and comprising a hub, a plurality of blades pivotally supported by said hub and said blades and pivots being so connected that said blades are hydraulically unbalanced for movement toward the open position of said blades at speeds above normal, and blade positioning means interconnected to said blades and including a hydraulically operated servomotor having a piston operable to vary the positions of said blades and maintain them in a substantial range of operative positions, in combination with a hydraulic fluid by-pass valve mounted in the piston of said servomotor operable upon said turbine being subjected to a selected predetermined runaway speed condition to open and permit passage of hydraulic fluid from the end of the servomotor functioning to prevent said movement of said blades to the other end thereof, whereupon said blades move toward said open positions thereof and thereby limit the runaway speed of the turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

8. The hydraulic turbine structure of claim 7 further characterized by said piston of said servomotor being rotatable with said runner and said by-pass valve being responsive to and operable by centrifugal force generated by the rotation of said piston at a predetermined speed.

9. A hydraulic turbine including an adjustable blade runner supported for rotation about its axis and comprising a hub, a plurality of blades pivotally supported by said hub and said blades and pivots being so connected that said blades are hydraulically unbalanced for movement toward the open position of said blades at speeds above normal, and blade positioning means including a hydraulically operated servomotor having a cylinder and piston therein interconnected to said blades and hydraulic pressure and return conduit means interconnected between a source of fluid pressure and said servomotor, said control system also including a control valve connected in said conduit means and operable to direct pressure against either end of said piston selectively to vary the positions of said blades and maintain them in a substantial range of operative positions, in combination with a hydraulic conduit connected to opposite ends of the cylinder of said servomotor and a pressure responsive relief valve in said conduit, said relief valve being responsive to pressures developed in said control system upon said turbine exceeding a predetermined speed, whereupon said valve will open to permit passage of hydraulic fluid from the pressure end of said cylinder functioning to prevent said movement of said blades to the opposite end thereof, thereby to permit movement of said piston and blades toward said open positions thereof and limit the speed of said turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

10. A hydraulic turbine including an adjustable blade runner supported for rotation about its axis and comprising a hub, a plurality of blades pivotally supported by said hub and said blades and pivots being so connected that said blades are hydraulically unbalanced for movement toward the open position of said blades at speeds above normal, and blade positioning means interconnected to said blades and including a control valve connected by conduit means to a sump reservoir and also to a hydraulically operated servomotor controlled by said valve and having a piston operable to vary the positions of said blades and maintain them in a substantial range of operative positions, in combination with electrically actuated mechanism interconnected with said blade positioning means and including a switch operable to control said electrical means, said switch being mounted relative to said turbine so as to be responsive to the rotary speed of said runner and upon said turbine being subjected to a selected predetermined runaway speed condition in excess of normal, said switch being operable to actuate said electrically actuated mechanism and cause it to actuate said valve so as to connect the pressure in the end of the servomotor serving to prevent said movement of said blades with said sump reservoir, whereby said blades are caused to move toward said open positions thereof and thereby limit the runaway speed of the turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

11. The hydraulic turbine structure of claim 10 further characterized by said control valve having a vertically movable valve member connected by a magnetic clutch to actuating mechanism in said positioning means, said clutch being operable by said speed responsive switch upon said predetermined runaway speed condition occurring, thereby to release said valve member for movement to a position to relieve pressure in the end of the servomotor which prevents said movement of said blades.

12. The hydraulic turbine structure of claim 11 additionally including a weight connected to said valve member and operable upon release of said clutch to shift said valve member by gravity to pressure relieving position.

13. A hydraulic turbine including an adjustable blade runner mounted for rotation about its axis and comprising a hub, a plurality of blades pivotally supported by said hub and said blades and pivots being so connected that said blades are hydraulically unbalanced for movement toward the open position of said blades at speeds above normal, and positioning means interconnected to said blades and operable to vary the positions of said blades and maintain the same in desired normal operating positions, said positioning means including means fracturable to disconnect said blades from said positioning means due to force generated by the abnormal unbalance of said blades upon said turbine attaining a predetermined runaway speed, thereby to permit the unbalance of said blades freely to move said blades toward said open positions thereof and limit the runaway speed of said turbine to a value corresponding to that of the open position of the blades to which they have been moved by said hydraulic unbalance.

14. A hydraulic turbine comprising a runner having a hub and pivotally adjustable blades thereon, the blades and pivots thereof being so connected that said blades are hydraulically unbalanced for movement toward the open positions of said blades at speeds above normal, each of said blades having a rocker arm thereon, and positioning means including a crosshead and fracturable means interconnecting said crosshead and rocker arms on said blades, said positioning means being operable normally to maintain said blades in desired positions of operation and also vary the same to produce a desired turbine output at normal speed, in combination with runaway speed limiting mechanism interconnected with said normal positioning means and operable automatically at speeds of a predetermined amount above normal to free the blades from the control of said normal positioning means and permit the said hydraulic unbalance of said blades to move the same toward said open positions thereof, thereby limiting the runaway speed of the turbine to a value

corresponding to that of the open position of the blades to which they have been moved by said hydraulic unblance and in the event movement of one of said blades toward open position is prevented such as by foreign matter being lodged between said blade and hub of the runner said fracturable means interconnected to said blade will be ruptured by the force exerted by the other blades upon said crosshead, thereby disconnecting said one blade from said crosshead and permitting the other blades still interconnected to the crosshead to move as directed by said inherent unbalance of the other blades to the open position thereof.

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