July 25, 1933. L. F. MOODY 1,919,376

REVERSIBLE PUMP TURBINE

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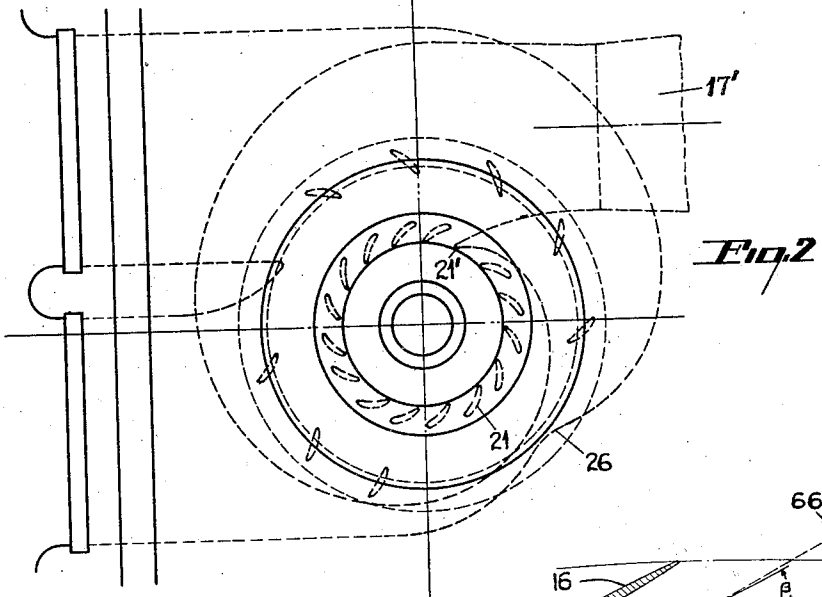


Fig.2

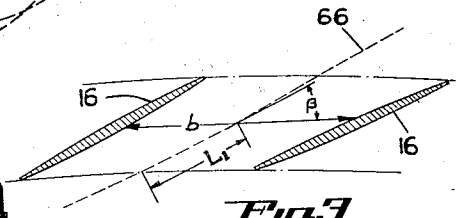


Fig.3

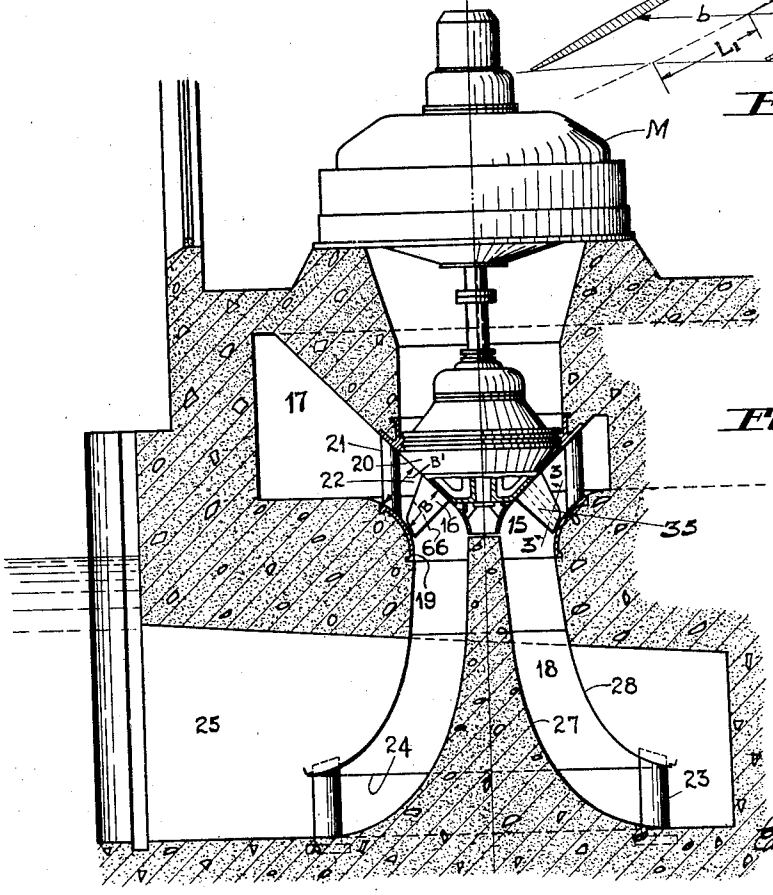


Fig.1

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REVERSIBLE PUMP TURBINE

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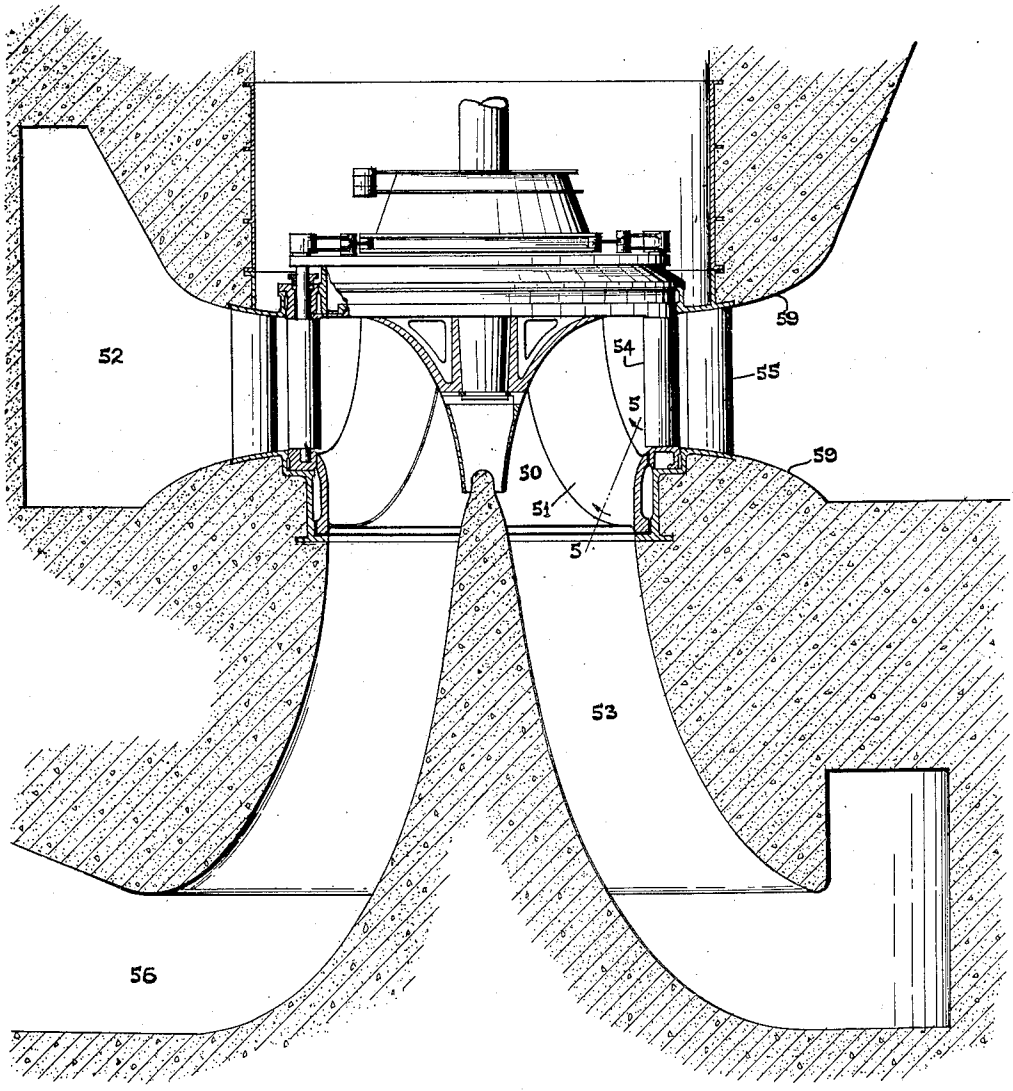
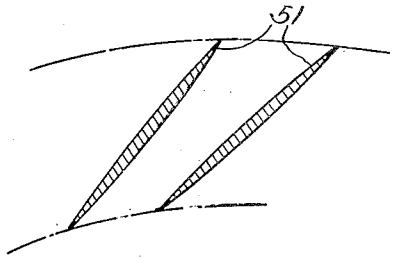


Fig. 4

Fig. 5.



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REVERSIBLE PUMP TURBINE

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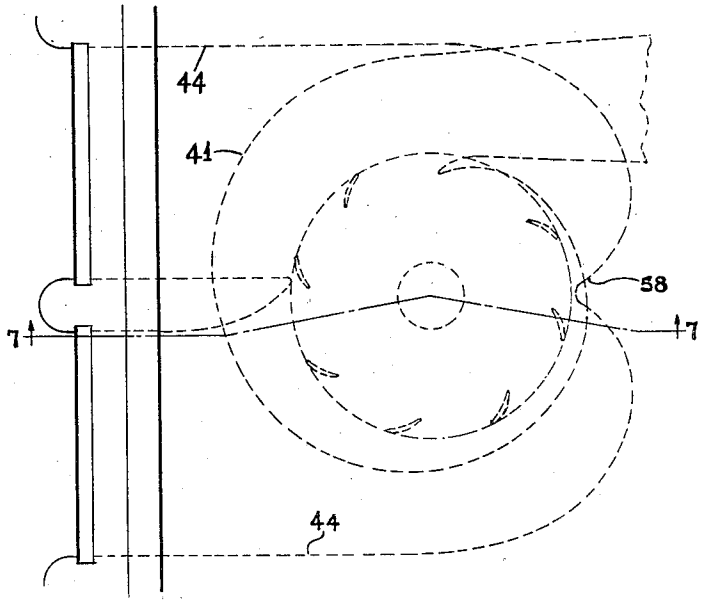


Fig 8

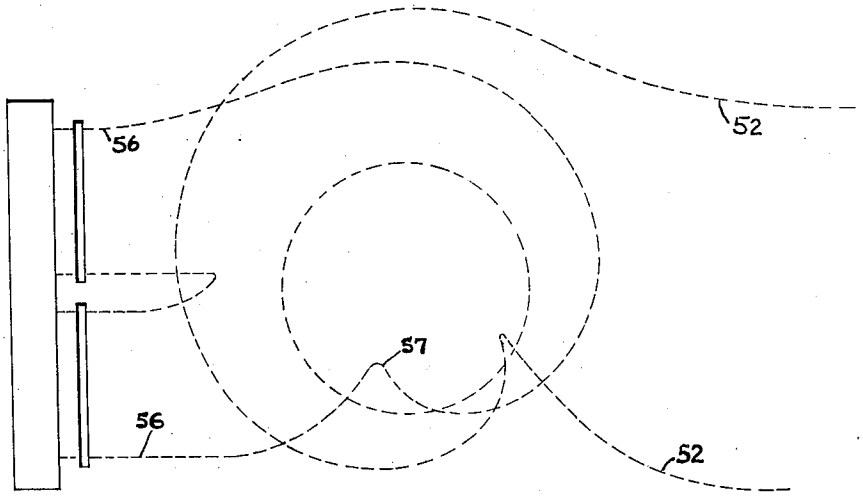


Fig 6

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REVERSIBLE PUMP TURBINE

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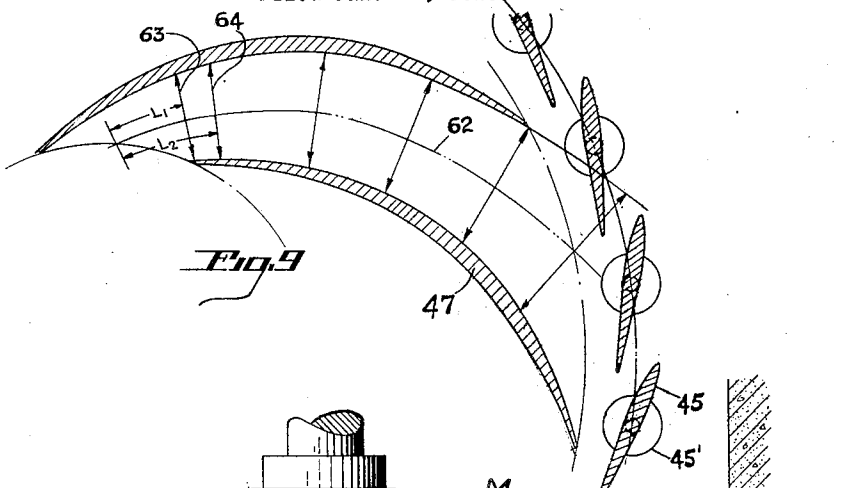


Fig. 9

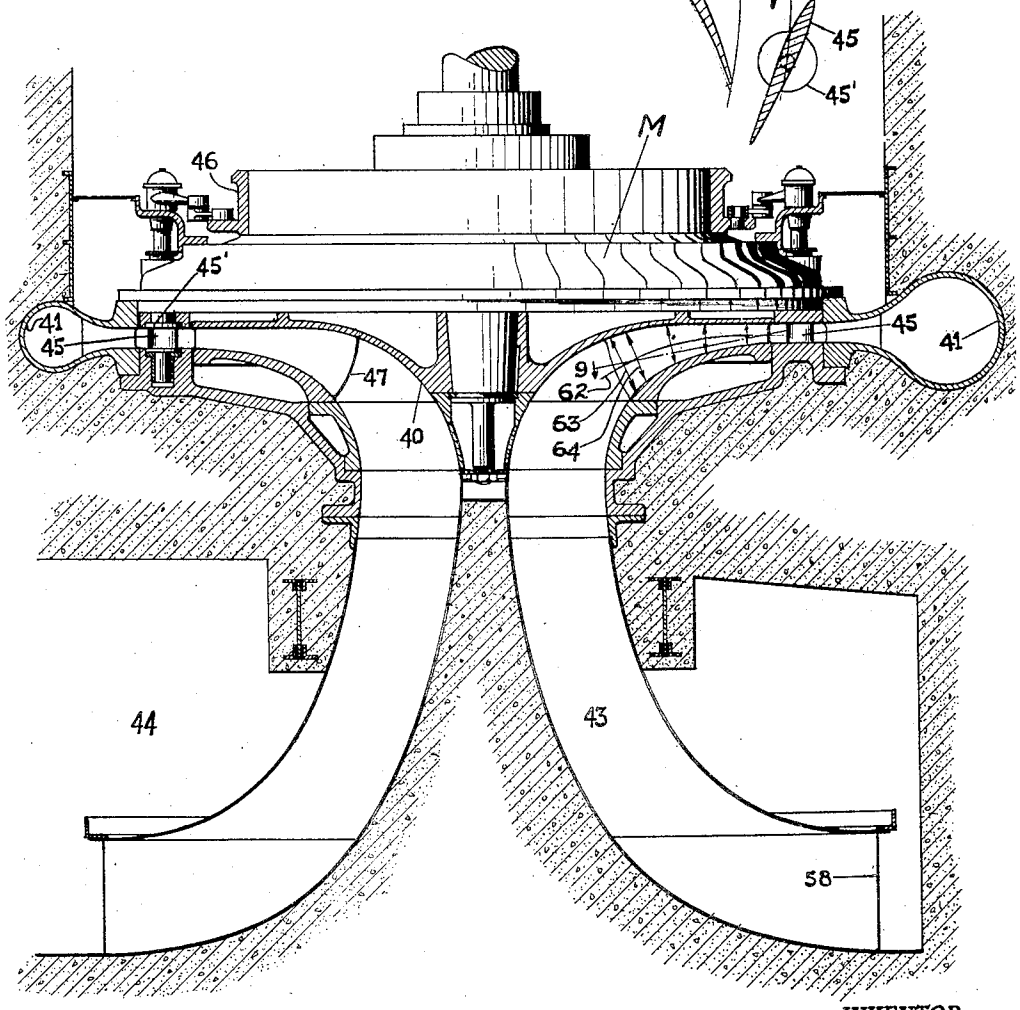


Fig. 7

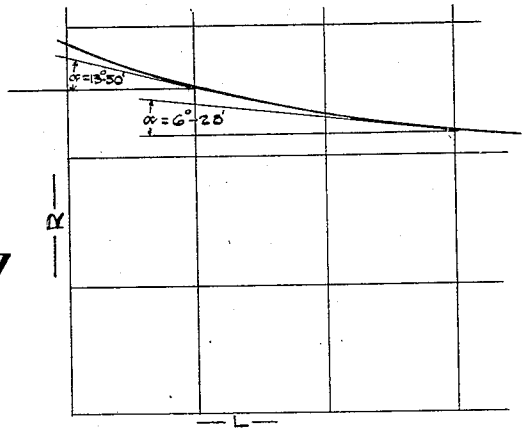
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REVERSIBLE PUMP TURBINE

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Fig 10



EQUIVALENT CONICAL TUBE CORRESPONDING TO GUIDE VANE PASSAGES

Fig 13

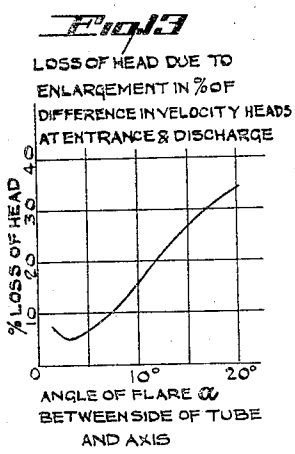
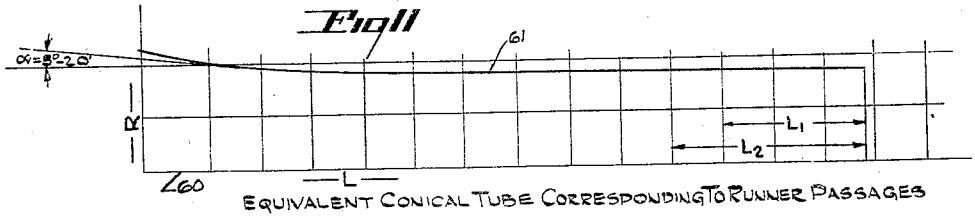
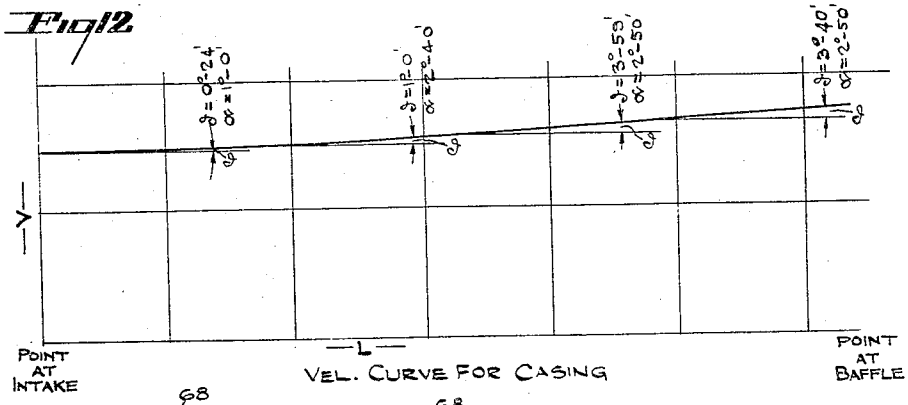


Fig 11



EQUIVALENT CONICAL TUBE CORRESPONDING TO RUNNER PASSAGES

Fig 12



VEL. CURVE FOR CASING

Fig 14

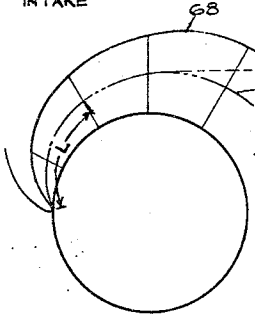


Fig 15

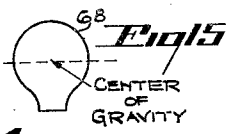


Fig 17

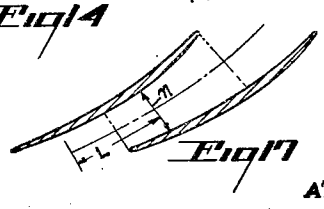
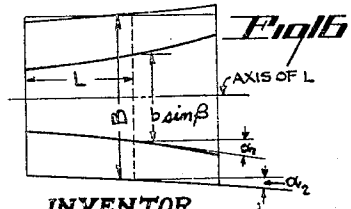


Fig 16



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July 25, 1933. L. F. MOODY 1,919,376

REVERSIBLE PUMP TURBINE

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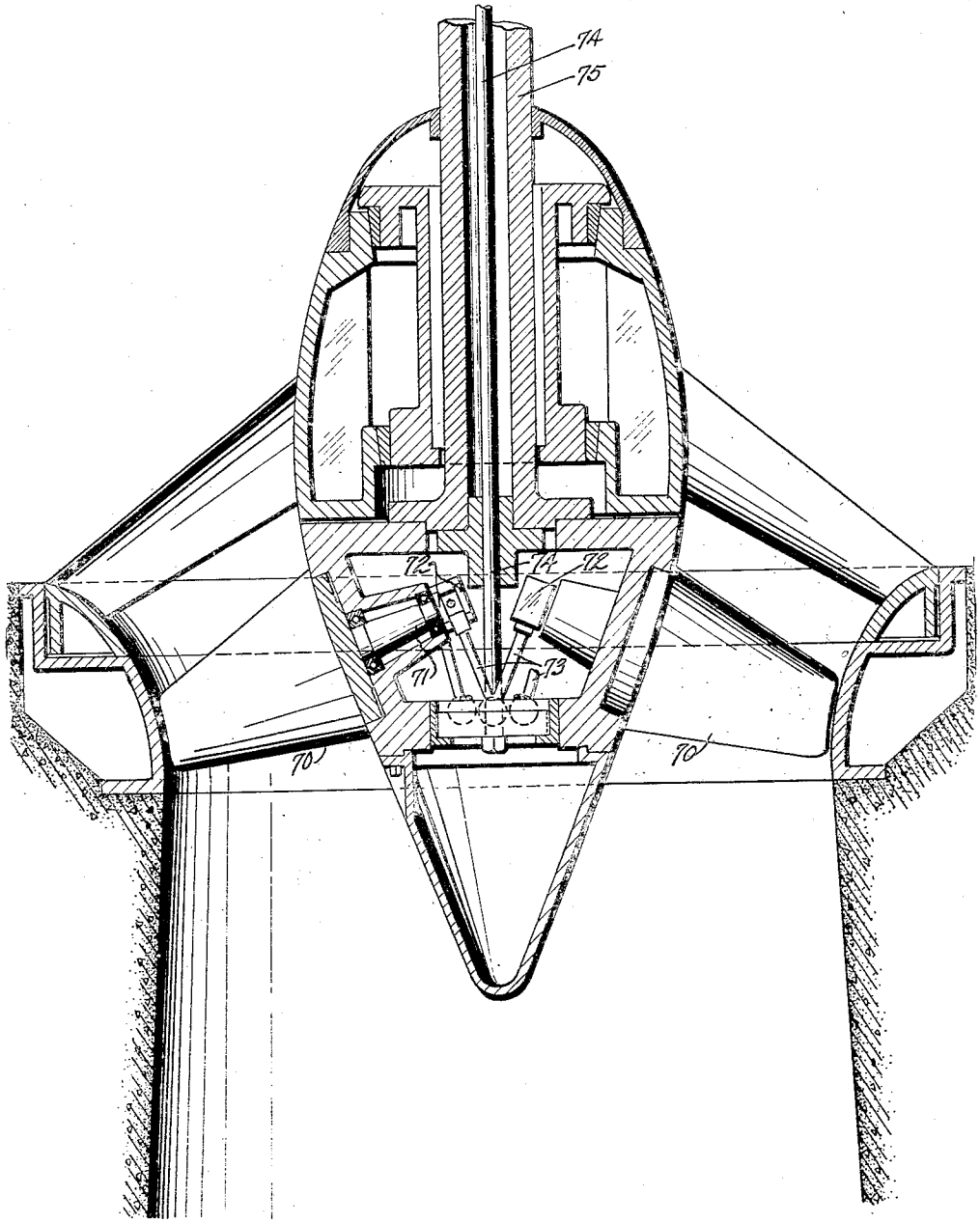


Fig. 18

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Patented July 25, 1933 1,919,376

UNITED STATES PATENT OFFICE

LEWIS FERRY MOODY, OF PHILADELPHIA, PENNSYLVANIA

REVERSIBLE PUMP TURBINE

Application filed January 11, 1929. Serial No. 331,910.

This invention relates generally to hydraulic machines of the reversible pump-turbine type and one object of this invention is to produce a hydraulic machine which will operate efficiently and effectively at times as a turbine or hydraulic motor for converting hydraulic powers into mechanical power, and at other times as a pump, utilizing mechanical power to impart head to water or to raise water against a head.

Another object is to produce such a machine adapted to work as a turbine under a given head, or range of heads, and at a given speed, which will be reversible so as to be capable of pumping effectively against approximately the same head or range of heads, or against a head only moderately in excess of that head, while running at substantially the same speed or at a speed not greatly in excess of the first value.

Another object is to produce a machine which will be interchangeable in action from turbine to pump by reversing at the same time both the direction of flow of the working fluid and the direction of revolution of the rotating element, and which, by adhering to the principles hereinafter described, may be applied to either low or high heads by suitably proportioning the parts.

Another object is to produce a machine which is capable of operating either as a pump or turbine, and which when used as a turbine will be regulatable to control its speed or load.

Still other objects and advantages will appear from the following description of the accompanying drawings in which;

Fig. 1 is a sectional view of a high specific speed propeller type unit embodying the principles of my invention,

Fig. 2 is a plan view taken just below the generator of Fig. 1,

Fig. 3 is a developed conical section through the blades taken substantially on line 3—3 of Fig. 1,

Fig. 4 is a sectional view of a mixed flow Francis type unit embodying the principles of my invention,

Fig. 5 is a section similar to Fig. 3 and taken on line 5—5 of Fig. 4,

Fig. 6 is a diagrammatic plan view of Fig. 4,

Fig. 7 is a vertical section of a lower specific speed Francis type turbine embodying the principles of my invention, the section through the collector passage being taken for simplicity along the line 7—7 of Fig. 8,

Fig. 8 is a diagrammatic plan view of Fig. 7,

Fig. 9 is an enlarged substantially horizontal section of two adjacent runner vanes and associated guide vanes taken on line 9 of Fig. 7,

Figs. 10 and 11 are equivalent conical tube curves of the runner and guide vane passages, corresponding to Figs. 7–9,

Fig. 12 is the velocity curve for determining the cross-sectional areas of the casing or inlet, for the turbine of Figs. 7 to 9,

Fig. 13 is a curve showing the relation between loss of head and rate of enlargement of the passage,

Figs. 14 and 15 are a diagrammatic plan and section of a volute casing or passage,

Fig. 16 is an equivalent rectangular passage,

Fig. 17 is a diagram of a moderate specific speed runner having vanes with considerable overlap and showing a method of obtaining the proper passage between the two successive vanes,

Fig. 18 is a vertical sectional view of an adjustable blade runner and blade adjusting mechanism.

Frequently in hydraulic power developments, and sometimes in connection with steam power stations, opportunities arise, for storing or accumulating power during periods of low power demand on the system, so that this stored power may be utilized during periods of peak loads. This may be feasible when a natural reservoir is available or when one may be created by dams, and when a supply of water or a second reservoir is available, together with a difference of level between water surfaces to give a suitable head. Such power storage may be feasible both in hydraulic developments on rivers and in tidal power developments, and may sometimes be valuable in connection

with steam power systems, even when little or no hydraulic power is available beyond that created by storage.

In hydraulic developments on rivets interconnected with large steam power systems, when a sufficiently large reservoir exists above the dam, it may often be economical to install a number of units operable as pumps to store power for peak load utilization in the steam power system.

The pumping and utilization of the stored water may be accomplished by using separate pumps and turbines, the pumps being operated when excess plant capacity or river flow is available and the turbines being operated at times of high load demand. According to the present invention it is proposed to use the same by hydraulic unit coupled to an electric generator or motor, both for pumping and turbine operation. When pumping the electric motor operates as a motor and when utilizing the stored water the same electric motor runs as a generator, the direction of rotation of the rotating element of the hydraulic and electric unit being reversed.

The hydraulic machine may be described as hydrodynamically reversible. Centrifugal pumps as customarily built are not adapted to run efficiently as turbines, and turbines of customary form are not adapted to operate effectively as pumps, or without prohibitive hydraulic losses. The usual forms of the rotating element or runner are not adapted to operate reversibly with the desired degree of efficiency. Pumps of usual type if run as turbines are not only unsuited in form of runner but do not have the means for regaining the kinetic energy of the water discharging from the runner with reversed flow and therefore would develop but little power and efficiency; and moreover lack regulating means for governing speed and load. Turbines of customary forms, such as the Francis turbine, are incapable of effective use as pumps, because with reversed flow the runner vanes are so formed that the passages between them enlarge so rapidly in cross-section that prohibitive "enlargement losses" would occur; and in addition to this the turbine casings are not adapted to handle reversed flow without causing large losses in eddies and turbulence, and are not capable, without radical changes in form and proportions, of acting as efficient diffusers or regainers of the kinetic energy of the high velocity discharge from the runners if it should be attempted to run them as pumps. Moreover, the incorrect hydraulic conditions would promote rapid deterioration through hydraulic corrosion or "pitting".

In the hydraulic machine of this invention, the water passages to and from the runner are so formed and proportioned that the velocity gradually accelerates in the entrance passage as the runner is approached, and gradually decreases in velocity or decelerates as the fluid flows away from the runner; so that when the flow is reversed the gradual deceleration in what was before the discharge passage becomes a gradual acceleration, and the gradual acceleration in what was before the entrance passage becomes a gradual deceleration; and in each passage sudden enlargements and rapid curvatures of walls which would permit the flow to separate from the walls are avoided for both directions of flow. By these means the machine incorporates a discharge passage which will act as an efficient diffuser or regainer of kinetic energy whichever direction of flow is considered. The runner is so formed that the cross-sectional area of the vane passages, measured normally to the relative velocity of flow, either remains approximately constant, or increases at so gradual a rate when the runner is operating as a pump impeller that the relative velocity of entrance to the impeller will be efficiently decelerated.

For convenience of identification the usual turbine inlet and draft tube pasages are referred to respectively as the inlet or entrance passage and the discharge passage even though the reverse functions are performed by these passages when operating as a pump.

In one specific aspect of the invention the above principles are embodied in a high specific speed propeller type unit as shown in Figs. 1–3 wherein the unshrouded propeller runner 15 has a usual form of conical hub carrying a relatively small number of diagonal blades 16 which preferably overlap in the manner shown in Fig. 3, that is, with respect to a line normal to the path of flow 66. This overlapping of the blades effects efficient diffusion and gives stability of operation. A spiral inlet passage 17 commencing at the intake 17′ and an annular spreading discharge passage 18 conduct water relative to the runner which is disposed at the juncture of the passages, this juncture being defined by the indicated turbine parts such as the curb ring 19 and the head cover 20 and the stay ring 21. This ring has fixed guide or stay vanes 21 suitably arranged and shaped to cooperate with the flow in accordance with the principles of this invention, these principles also applying to baffle 21′ which is the terminal of the spiral inlet. The runner is spaced from the guide vanes to form a transition space 22, while lower stay vanes 23 are also provided a substantial distance from the runner. The spreading annular passage 18 terminates at 24 in a collector passage 25 which extends downstream, the wall of the collector approaching the discharge edge 24 as at 26. The conical core 27 preferably extends up to the runner hub to form a relatively smooth continuation of the outer surface thereof while the outer wall of passage 18 comprises either plate metal 28, or may, as

shown in Fig. 4, be made wholly of concrete. The runners in all forms herein disclosed are connected, as shown, directly to the shaft of the motor or generator M, the function thereof being dependent on whether pump or turbine operation is had.

In Figs. 7 to 9 there is shown a low specific speed turbine to which the principles of my invention have also been applied. The runner 40 when operating as a turbine receives water from a casing 41 and discharges into an annular spreading draft tube 43 and collector passage 44 while relatively thin wicket gates 45 are adapted to vary the angle of whirl imparted to the flow, these gates being adjusted by usual mechanism including a shifting ring 46. As is usual in low specific speed turbines the blades 47 are relatively large in number in comparison to the propeller runner where usually not more than about six blades are used, but are fewer in number than in a usual non-reversing turbine of corresponding specific speed, this being illustrated by the disclosure of Fig. 9 wherein the spacing of the blades is such as to require only eight blades. It will also be noted that in the low specific turbine of Fig. 9 the blades have a greater degree of curvature in the direction of flow than does the propeller runner and that the blades are long in direction of flow to permit gradual deceleration. In the forms of the invention shown in Figs. 1 and 7 the outer wall of the runner passage in meridian section has a large radius of curvature which is conducive to efficient and stable operation, it being noted that in general the radius of curvature of the outer surface is not less than approximately two-thirds of the throat diameter, noting in particular Fig. 1 whereas in Fig. 7 the radius of curvature is approximately equal to the full radius of the throat diameter. The throat is considered as being that portion of the passage having the smallest diameter adjacent the runner. It will be further observed in all three forms that the flow passage and/or runner passages from the lower side of the runner up to the guide vanes do not increase in width when measured in meridian planes, or as expressed in another aspect, the transverse width of the flow passage adjacent the guide vanes when measured in a meridian plane either falls within or is not greater than the transverse width of the passage adjacent the lower portion of the runner. For instance, the guide vanes 54, Fig. 4, are of appreciably less length than would normally be the case with a usual non-reversible Francis type turbine while in connection with Fig. 1 the flow passage leading upwardly from the runner to the point adjacent the guide vanes 21 is of substantially uniform width.

In Fig. 4 a moderately high specific speed or mixed flow type of Francis unit is shown having a runner 50 provided with a moderately large number of spoon-shaped blades 51, a spiral entrance passage 52 and spreading annular discharge passage 53. The runner blade passages and entrance and discharge passages are designed in accordance with the principles herein outlined as is also the case with the adjustable guide vanes 54 and fixed vanes 55. The collector passages may be varied in their design, for instance, in Fig. 6 collector 56 has its spiral portion start at a lateral point 57 while in Fig. 8 it starts at a central point 58. The upper and lower walls 59 at entrance have gradual relative divergence.

As a guide in proportioning both the entrance and discharge passages and the runner and guide vane passages, equivalent conical tubes may be laid out in diagrams, an example of which is as follows: Along an axis 60, Fig. 11, distances are measured representing to scale distances measured along the middle flow line of the passage, which is here the runner passage of Fig. 9. At each point so located on the axis 60 a diameter (or radius R as in Fig. 11) is drawn perpendicular to the axis representing to the same scale the diameter or radius of a circle having an area equal to the transverse area of the passage at the corresponding point, taken perpendicularly to the flow. These diameters or radii R determine the walls 61 of a conical tube; and if efficient diffusion or deceleration is to be accomplished, the angle of flare, or angle of the conical wall to the axis of the cone, should be between about 4° and 7° for the discharge passages, and should not materially exceed 7°; and in the equivalent cone corresponding to the runner vane passage, the angle of flare should not exceed about 5° on account of the high relative velocity of flow through the runner. In placing the limiting angle of flare for efficient deceleration at about 7° it should be understood that an exact limit can not be prescribed for the reason that the loss of head accompanying deceleration varies with respect to the angle in accordance with a curve, reaching its extreme minimum value in the neighborhood of about 3° or 4°, and increasing at a progressively increasing rate as the angle is increased. The character of the relation is shown by the curve in Fig. 13. From this it will be seen from the rounded form of the curve that angles somewhat greater than the best angle cause only a moderate increase in the loss, but as the angle is further increased the loss increases more and more rapidly, so that if one departs from a restricted zone, excessive losses will be incurred. In practice, in large machines, the most economical angle will not be that for the extreme minimum shown on the curve, but considering the final discharge loss and the increased dimensions and cost accompanying the use of very long gradually tapering passages, the best angle in most

cases can be advantageously made greater than 3° or 4°, and may be from 5° to 10° in practice, a value of about 6° or 7° being found from experience to be consistent with economy and high efficiency. Preferably the angle of flare of the equivalent cone corresponding to the runner passage may be made about 3° or 4°, so that when operative as a turbine the runner will have a slightly and gradually contracting passage between successive vanes; and when operating as a pump there will be a gradual deceleration of the relative velocity during the passing of the water through the impeller and while it is increasing in static pressure due to the transformation of mechanical energy into head; but as pointed out above slightly greater angles, namely 6° or 7°, can be advantageously used in many cases as consistent with both economy and high efficiency.

Applying to the Fig. 7, 9 form of runner passages, as an illustration, the above outlined method for determining the equivalent conical tube, distances L_1, L_2 etc., are measured along the middle flow line 62 and the transverse areas at the lines 63, 64 etc. are determined. The radii of circles having areas equal to these transverse areas are then laid off from the axis 60, Fig. 11 as radii R and thus the outline 61 of one half of the conical tube is obtained.

To plot the equivalent cone for the diffuser channel between two blades 16 of the propeller runner, transverse areas are taken equal to $B \times b \sin \beta$ (Figs. 1 and 3) at selected points along the center line of flow 66 at distances L_1 etc. from the end of the passage. (The center line of flow is the mean flow line at the center of the annular area at about seven-tenths of the tip radius.) Then from

$$\pi R^2 = B \times b \sin \beta \text{ find}$$

$$R = \sqrt{\frac{B \times b}{\pi}} \sin \beta,$$

and plot R vs. L as in Fig. 11. Then α is the angle of flare of the equivalent cone, and α should not exceed about 5°.

To plot the equivalent conical tube for the discharge passage, one specific method which may be followed for an annular spreading passage is disclosed in my Patent No. 1,315,232 while the equivalent conical tube for the guide vanes as shown in Fig. 10 is determined by the same methods used for the runner vane passages. The equivalent conical tube for draft tubes of elbow or other formation could be determined in accordance with the principles outlined herein.

In designing the volute casing for efficient deceleration, instead of laying out an equivalent conical tube having cross-sectional areas equal to those of the casing, as in the case of the runner vane and guide vane passages and the draft tube, a modified procedure must be followed on account of the fact that the volute casing or entrance passage 68, diagrammatically shown in Figs. 14 and 15, is continuously receiving additional quantities of fluid around the turbine periphery due to the lateral entrance of fluid from the periphery of the runner and guide vanes. This varying quantity of the fluid flowing in the casing is taken care of by plotting a curve Fig. 12 of velocities at successive points in the casing with respect to distances L, measured along the mean line of flow 69 (drawn to pass through the centers of gravity of the successive casing sections) see Figs. 14 and 15. Now taking the radius of a circle having an area equal to that of any section of the casing

$$= \sqrt{\frac{\text{area}}{\pi}},$$

and calling this R, a curve is plotted between V and L, (Fig. 12) to scale, V being the velocity at the section considered in feet per second. Calling the scales to which diagram is plotted

S_V and S_L where

S_V = scale for velocities = inches on diagram representing 1 ft./sec.

S_L = scale for lengths = inches on diagram representing 1 ft.

then the corresponding angle of flare α of an equivalent conical tube having the same rate of deceleration is calculated from the formula

$$\tan \alpha = \frac{R}{2V} \times \left\{\frac{S_L}{S_V}\right\} \times \tan \delta;$$

in which δ is the angle of slope of the velocity curve at the point considered, measured from the diagram as indicated in Fig. 12. The values of α so determined, should be small and preferably not materially greater than about 7° at any point in the casing to give sufficiently gradual deceleration for efficient regain of velocity head.

In a turbine the velocity of flow in the casing is cutomarily made low relatively to the head on the turbine, the corresponding velocity head being in usual practice between 2% and 5% of the head on the turbine. In such a machine used solely for turbine operation, no loss is incurred in rapidly accelerating the velocity from the casing to the runner, since fluid may be rapidly accelerated in a contracting passage without causing turbulence or loss. In a machine designed for interchangeable pump operation however the velocity leaving the runner is high compared to the head, and gradual deceleration of this velocity is essential if turbulence and material loss of head is to be avoided; this requires that this gradual deceleration must be effected throughout the transition from the runner to the casing, and throughout the casing itself, and consequent-

ly the casing velocity must in many cases be much higher than that provided in the usual turbine and must gradually decrease throughout the casing. In a pump of moderate or high specific speed the velocity head required in the volute casing will in general be of the order of 10% to 25% of the head, or several times that used in turbines, and to make the machine efficiently operable as a pump, the sections of the volute casing will have to be made much smaller than in normally designed turbine, with casing velocities two or more times the values used for normal turbines.

As another criterion for the form of the runner vane passage, if a diagram is constructed, Fig. 16, by rectifying the centerline of flow between two successive vanes so as to obtain a straight axis such as 60 in drawing the equivalent cone, and the width of passage from vane to vane and the breadth perpendicular to the shroud ring and hub are laid off to form a rectangular tube, the walls of the tube in either direction should not diverge from each other by more than about 10° if there is divergence in both directions. In general, however, α_2 (Fig. 16) will be negative; that is, the transverse breadths of the runner vane passage measured in a plane containing the runner axis (meridian plane) as shown at 63, 64 in Fig. 7, will gradually decrease in the direction of the flow when pumping; so that the walls formed by the hub and shroud ring will gradually converge. At the same time the transverse widths from vane to vane in a section normal to the meridian section (the conical section of Fig. 9) will gradually increase in the direction of pump flow, giving a positive angle of divergence, α_7, which should not exceed about 10°; so that the entire divergence of the walls formed by the back of one vane and face of the next will not exceed about 20°.

To construct the equivalent rectangular passage for the propeller runner, the values of B and $(b \sin \beta)$ respectively are plotted against L as shown in Fig. 16. In this rectangular equivalent tube neither α_1 nor α_2 should exceed 10°.

In an ordinary form of runner in which the vanes have a substantial amount of overlap, the channel area may be calculated as above throughout, or the middle portions as shown in Fig. 17 may be taken as $B \times n$, so that

$$R = \sqrt{\frac{Bn}{\pi}};$$

and the rectangular equivalent tube drawn by plotting B and n against L.

In order to provide regulating means for the turbine and also to permit the discharge during pump operation to be adjusted to compensate for the change in condition due to this reversed operation at substantially the same speed and head, as frequently desired, the movable guide vanes or wicket gates 45 or 54 may be provided at entrance to the runner (running as a turbine). These are shaped to conform to the gradually decelerating flow of the runner discharge during pump operation, so that the areas between these guide vanes, when in approximately their normal position for pumping, must increase at a rate consistent with an equivalent cone or tube as shown in Fig. 10 and according to the same rule as for the other passages; and the guide vane surfaces must be relatively flat and free from sharp curvatures, and must be relatively thin and have gradually tapering discharge edges for both directions of flow, to enable the streams passing between the guide vanes to merge and reunite in a single continuous stream after discharging from them. To permit the use of relatively thin guide vanes, relatively large collars 45′ may be provided to obtain sufficient strength between the guide vanes and shafts therefor.

Theoretically considered, a machine following the principles above described can be reversed in direction of revolution and flow, all the velocities remaining the same in amount but reversed in direction, and the vector diagrams of velocities will then remain the same with only the directions reversed, and the flow will continue to agree with the runner vane angles at entrance and discharge and no "impact losses" due to disagreement between the flow direction and the vane direction will occur. This reversal at constant speed of revolution would however require the head acting to be different for pump operation from that for turbine operation, to give identical velocity diagrams. The head acting upon the machine, from entrance to discharge or from discharge to entrance, minus the head loss in the machine due to friction, eddies and discharge velocity head losses, may be termed the effective head on the runner and denoted H_e. When operating as a pump the total head on the machine is H_p and calling the head lost in the machine H_{Lp}, then

$$H_e = H_p + H_{Lp} = \frac{H_p}{E_p},$$

in which E_p is the pump efficiency; but when operating as a turbine the total head on the machine is H_T and calling the head then lost in the machine H_{LT}, the relation is: $H_e = H_T - H_{LT} = H_t E_t$, in which E_t is the turbine efficiency. Hence to obtain the same effective head on the runner and the same velocity diagrams when the machine is reversed the head on the machine would have to be changed from H_p for pump operation

to H_t for turbine operation, according to the relation:

$$H_e = \frac{H_p}{E_p} = H_t E_t;$$

that is the head would have to be changed in the ratio:

$$\frac{H_p}{H_t} = E_p E_t.$$

Under usual requirements, however, the head will remain approximately the same, or be slightly lower for turbine action than for pump action due to conduit losses, outside of the machine. To obtain exactly similar velocity diagrams in this case it would therefore be necessary to correct the speed of revolution to keep

$$\frac{u}{\sqrt{H_e}}$$

the same (where u is the linear velocity of a point of the runner). In the case of $H_t = H_p$, we should have H_{ep}, the effective head on the runner for pump action, greater than H_{et}, that for turbine action, according to the relation:

$$H_p = H_{ep} E_p = H_t = \frac{H_{et}}{E_t}$$

so that

$$\frac{H_{et}}{H_{ep}} = E_p E_t$$

The speeds would then have to be adjusted to give:

$$\frac{U_p}{\sqrt{H_{ep}}} = \frac{U_t}{\sqrt{H_{et}}}$$

or would bear the ratio:

$$\frac{U_t}{U_p} = \frac{\sqrt{H_{et}}}{H_{ep}} = \sqrt{E_p E_t}$$

in which U_t and U_p are the linear velocities of a point on the runner for turbine and pump action respectively, these being proportional to the revolutions of the runner. That is, for equal total heads, the speed for turbine operation should be

$$\sqrt{E_p E_t}$$

times the pump speed, that is the pump speed multiplied by the geometric mean of the efficiencies. Taking for example, a turbine having an efficiency of 90% which would operate as a pump with an efficiency of 80% the turbine speed should theoretically be reduced to

$$\sqrt{.90 \times .80}$$

times the pump speed, or to about 85% of the pump speed. The loss in supply and discharge conduits can be taken into account by using for E_p and E_t the overall efficiencies including these conduit losses, instead of the efficiencies of the machine proper.

Now for large machines where high values of efficiencies may be counted upon, this speed correction may probably be kept somewhat within the 15% reduction required by the above example; and by operating the pump slightly below its exact speed for best efficiency and running the turbine somewhat overspeeded, such a speed variation may be divided between the pump and turbine operation so that each operation will be only slightly off-speed; and since the efficiency curves of both turbines and pumps are usually relatively flat in the maximum efficiency region they may be operated slightly off-speed without serious reduction in efficiency. Moreover, if the turbine is equipped with guide vanes these may be opened somewhat above their position for normal pump operation, thus increasing the discharge as a turbine and improving the efficiency as well as increasing the power developed to a value more nearly approaching the motor capacity required for pumping. It will, therefore, be practicable under most conditions to maintain the same speed for both turbine and pump operation. Of course the above relations involve the efficiencies actually to be attained, and these can be reliably determined only by tests on a definite design. When large machines are built, the efficiencies both as pump and turbine will increase with the size (according to a rule verified by tests, the percentage of loss in geometrically similar machines varies inversely as the fourth root of the linear dimensions), so that for large installations the variation from best speed will be relatively less. In some cases, for example if the total head for pumping must be considerably greater than for turbine operation, as when long supply conduits or pipe lines are used, it may be desirable to use a variable speed motor or generator such as one having pole-changing provisions, to maintain speeds close to the maximum efficiency conditions.

The guiding rules set out above for proportioning the runner vane channels and discharge and intake passages to give the most efficient deceleration are directed more particularly to machines of moderate or low specific speeds, applicable in usual practice to heads greater than about 60 to 75 feet. For lower heads, below these values, the machine will be more economical if of the propeller type, with axial or diagonal flow unshrouded runners, and of high specific speed. The diagonal propeller type machine will probably be applicable to heads even higher than 75 feet, as soon as there is opportunity to gain experience in actual installations under such heads. The angles of flare of the equivalent conical tubes used for a basis of design of the runner vane channels and discharge passages will be subject to modification in propeller machines, a larger range of choice of these angles then being called for on ac-

count of economical consideration due to the greater need of restricted dimensions of the structures in low head machines to conserve cost, but the general principles outlined for moderate or low specific speed machines will apply to the high specific speed machines.

In high specific speed machines for low or moderate heads, it may be advantageous, either in addition to movable guide vanes or instead of them, to make the blades of the propeller runners adjustable to aid in adjusting the machine to the best efficiency conditions when changing from turbine to pump operation, as well as to increase the range and flexibility of operation for variations in head or quantity of flow for either turbine or pump use. In reversible high specific speed propeller turbines for low or moderate heads, adjustable blade runners will have advantages in some cases, both for pump and turbine operation, with the blades adjustable either while running or by shutting down the unit for a short period. The blade adjustment will permit the most favorable conditions to be secured to suit the head, either for increasing the efficiency or the capacity, and by this means the change in conditions when reversing from turbine to pump operation or vice-versa can be provided for and the best blade angle secured for each mode of operation under the particular value of the head then existing. The reduction in power as a turbine compared to pump operation can be more completely compensated by turning the blades toward the vertical position and farther from the horizontal position when operating as a turbine, in order better to utilize the capacity of the electrical machine (generator-motor) or conversely the blades can be turned to a more horizontal position to avoid overloading the motor during pumping. The use of adjustable runner blades may be advantageously applied to propeller type pumps of high specific speed, whether reversible or not, whenever a variable discharge is desired. Adjustable runners and operating mechanism suitable for the above purposes and as shown here in Fig. 18 are disclosed more specifically in my Patent Nos. 1,837,568 and 1,853,139, it sufficing to state herein that adjustable blades 70 are mounted on shafts 71 journalled in the runner hub, which shafts have arms 72 connected by links 73 to a common axially movable rod 74 centrally disposed in a hollow runner shaft 75, thus effecting angular adjustment of the runner blades upon axial movement of the rod 74. Under many conditions however, and probably under the more usual requirements, the machine may be made to give satisfactory performance for reversible operation without going to the expense of providing runner blade adjustment, with its mechanical complication.

Reverting to the conditions in a low or moderate speed pump or turbine, it will be clear from the previous explanations that turbines of ordinary form involve too rapid acceleration of the relative velocity of flow while passing through the runner, so that if pump operation is attempted the rapid acceleration then becomes a rapid deceleration which cannot be efficiently secured. When the flow passes through rapidly enlarging passages it does not actually decelerate at a rate sufficient to take up a velocity equal to the quantity divided by the passage area, but continues to flow at nearly its original velocity and the flowing stream only partly fills the passage, leaving regions of eddies and backward flow; and instead of efficient deceleration there is merely a reduction of velocity by dissipation of velocity head and loss of energy. The requirements for a reversible machine are therefore the formation of the runner with vane shapes consistent with a very gradual change of relative velocity and with discharge velocity not greatly different in amount from entrance velocity, thus avoiding unstable operation, corrosion and loss of efficiency.

The conditions of gradual change in relative velocity and flatly curved blades with little change of direction, and discharge velocity nearly equal to entrance velocity, are all consistent with obtaining the most efficient design of high specific speed turbine and pump runners of the propeller type. To avoid cavitation, or separation of the streams from the blade surfaces with resulting unstable operation, vibration and pitting of the surfaces exposed to the flow, a blade area substantially equal to, or not materially less than, the disk area should be provided, and wide openings between successive blade edges should be avoided. This will insure proper guidance of the flow and avoidance of regions of high local vacua during both pump and turbine operation; and is particularly important in the pump operation, since there will then be a certain degree of deceleration between the blades and this necessitates even a greater degree of guidance or "cell" effect—that is, constraint of the flow within enclosed passages—than does the turbine operation, particularly if the runner must be set a considerable distance above the tailwater or sump water surface.

When the machine is equipped with guide vanes, these are preferably placed at entrance to the runner for turbine action, that is at the runner discharge for pumping. These will be fully effective for turbine regulation or governing by a usual governor control as set forth in my copending application previously referred to herein, and may also be used for adjusting the pump load or discharge; but the latter adjustment will be effective only within a relatively small range of guide vane angle since the pump is de-

pendent for good efficiency on the deceleration in the diffuser or casing, the guide vanes not being long enough to provide anything approaching complete deceleration between them within practicable limits of size, while the casing may be designed to provide a long path of flow for the entire stream with correspondingly gradual deceleration.

To find the relation between the specific speed of the machine running as a turbine and its specific speed as a pump, the following is the theoretic basis for this transformation, considering the case where the total head is changed to give equivalent hydraulic action, that is similar vector diagrams of velocity, or where the speed is corrected to give this equivalent action. The pump specific speed is defined by the formula

$$N_{sp}=N_p\frac{\sqrt{Q_p}}{H_t^{3/4}}$$

in which N_p is the number of revolutions per minute and Q_p the cubic feet of water per second discharged when pumping. The turbine specific speed is defined by the formula

$$N_{st}=N_t\frac{\sqrt{HP_t}}{H_t^{5/4}}$$

in which N_t is the revolutions per minute and HP_t the horsepower delivered on the shaft when operating as a turbine. Expressing the last formula in terms of the water quantity Q_t discharged by the turbine instead of in terms of

$$HP, N_{st}=N_t\frac{\sqrt{\frac{W}{550}Q_tH_tE_t}}{H_t^{5/4}}=\sqrt{\frac{W}{550}}N_t\frac{\sqrt{Q_tE_t}}{H_t^{3/4}}$$

in which W is the specific weight of the water in pounds per cubic foot. Now referring back, we have

$$H_c=\frac{H_p}{E_p}=H_tE_t$$

and

$$\frac{H_p}{H_t}=E_pE_t;$$

and this was based on the same velocities of water and points on the runner for both turbine and pump action, and since the passage areas remain the same throughout, as well as the runner diameters, this means that the revolutions and water quantity remain the same or $N_t=N_p$ and $Q_p=Q_t$—Accordingly we have

$$\frac{N_{st}}{N_{sp}}=\sqrt{\frac{W}{550}}\frac{N_t}{N_p}\sqrt{\frac{Q_t}{Q_p}}\left(\frac{H_p}{H_t}\right)^{3/4}\sqrt{E_t}=$$

$$\sqrt{\frac{W}{550}}\times1\times1\times(E_pE_t)^{3/4}\sqrt{E_t}=\sqrt{\frac{W}{550}}E_p^{3/4}E_t^{5/4}$$

With W=62.4 this becomes

$$\frac{N_{st}}{N_{sp}}=\frac{E_p^{3/4}E_t^{5/4}}{2.97}$$

This relation will be useful as a guide in approximate calculations, but is subject to variations from the exact final conditions which should be determined by tests on models geometrically similar to any proposed installations, since exactly similar flow conditions, as here assumed, will not be secured in reversed operation, due to differences in the distribution of velocities across the channel sections in the runner and elsewhere; and such tests are also necessary to determine the exact values of E_p and E_t; but if the machine is proportioned in accordance with the principles here set out this relation should furnish a useful guide in preliminary calculations and provide a basis for the design.

The relation between specific speeds for turbine and pump action may also be expressed in terms of the specific speed based on quantity for the turbine, a similar expression to that used for pumps, (as described in paper by me in A.S.M.E. Trans. 1921: "The present trend of hydraulic turbine development"), namely:

$$N_{sqt}=N_t\frac{\sqrt{Q_t}}{H_t^{3/4}}$$

instead of the usual specific speed, based on power

$$N_{st}=N_t\frac{\sqrt{HP_t}}{H_t^{5/4}}.$$

The above derivation using N_{sqt} instead of N_{st} then becomes—

$$\frac{N_{sqt}}{N_{sp}}=\frac{N_t\frac{\sqrt{Q_t}}{H_t^{3/4}}}{N_p\frac{\sqrt{Q_p}}{H_p^{3/4}}}=\frac{N_t}{N_p}\sqrt{\frac{Q_t}{Q_p}}\left(\frac{H_p}{H_t}\right)^{3/4}=1\times1\times(E_pE_t)^{3/4}$$

hence the relation expressed in these terms becomes

$$\frac{N_{sqt}}{N_{sp}}=(E_pE_t)^{3/4}$$

This way may be used for the same purpose as the one previously derived.

In the design of the machine for pump operation it is sometimes desirable, particularly in the case of high specific speed, to direct the flow into the runner with an initial whirl, that is with rotational velocity components around the runner axis, which for high specific speed should be in the same direction as the runner rotation. If entrance guide vanes are placed in the entrance passage for this purpose, they should be located at some distance from the impeller, and preferaly at a large radial distance from the axis and at a point of lower velocity than at the impeller, to provide a large intermediate transition space between the guide vanes and impeller, since when the machine runs as a turbine the rotational components in the discharge flow from the runner will differ somewhat in amount from those required for

pump action and will vary with the guide vane opening, and this difference would result in loss of head if the guide vanes were placed close to the runner.

The setting of this machine required for reversible operation will differ in each case from that required for turbine operation alone, or required for a non-reversible turbine of equal capacity and speed. Owing to the loss of head inherent in the decelerating action in the draft tube during turbine operation the absolute pressure at the entrance to the draft tube, that is at the discharge from the runner, is reduced to an extent equal to that portion of the velocity head which can be reconverted into pressure head by the decelerating action of the tube, and this can never amount to 100% of the velocity head, but in most cases will be in the neighbourhood of 80% or somewhat more. The turbine can be raised with respect to tailwater, when determining its setting, by an amount corresponding to this loss of head or in the neighbourhood of 20% of the velocity head or somewhat less, compared to the setting which would be necessary with perfect regain of velocity head. On the other hand, when the machine is pumping, the absolute pressure at the upper end of the draft tube, that is at entrance to the runner, is reduced by the full extent of the velocity head below the static pressure corresponding to the elevation, and by an additional amount corresponding to the loss of head in the tube, which in this case is mainly that due to pipe friction. Therefore the elevation of the runner for pump operation must be lower than that permissible for turbine operation by an amount equal to the sum of the losses of head in the tube during both pump and turbine operation, to avoid the occurrence of an absolute pressure less than the vapor pressure of water, with resulting "cavitation".

The commercial feasibility of utilizing a reversible hydraulic machine as here proposed will depend to an important extent on the efficiency attainable during both pump and turbine operation, since all of the power stored by pumping is subject to the loss occurring during pumping and again to loss during utilization, that is during turbine operation and with a given capacity of reservoir the power which can be developed from the stored water is proportional to the product of the efficiencies of pump and turbine actions; so that in most cases these efficiencies will be a controlling factor in determining whether an installation is economically justified.

The few types of runners herein disclosed have been chosen as examples to which the principles of my invention may be applied but it is to be understood that other specific forms of runners may be used when the blades and passages are properly proportioned in accordance with the theory and principles herein set forth. Examples of a few other forms of runners adapted to be modified in accordance with this invention are shown in my Patent No. 1,583,415 and application, Serial No. 373,545, filed April 13, 1920.

The propeller type turbine of Figs. 1 and 2 may also be equipped with movable guide vanes such as shown in Fig. 4 between the stay vanes 21 and transition space 22, the stay vanes being placed farther from the turbine axis and the contour of the upper end of curb ring 19 being suitably altered to accommodate these movable guide vanes, which would be arranged similar to vanes 54 in Fig. 4.

Instead of annular spreading types of draft tube shown in the figures, other efficient forms of draft tube may be employed such as those of improved elbow type, but it is essential in all cases to preserve gradual deceleration in the draft tube and to employ forms having high efficiency of regaining kinetic energy.

It should be noted that the above principles for obtaining gradual deceleration of the water leaving the runner when pumping result in a modified intake passage up to the runner when considered as a turbine, differing from usual turbines which are not reversible. Thus in Fig. 1 the transverse distance B′ across the passage, which has the value B at the runner, changes gradually and continuously in passing through the transition space 22, stay ring 21 until reaching the casing 17, in order that the passage area transverse to flow in the plane of the figure shall increase gradually and consistenly with a limited rate of deceleration of the meridian components of velocity, that is the components in the plane of the figure. This consideration results in a transverse width across the passage at 21 and 22, and a height of guide vanes or stay vanes 21, which is materially smaller for a given size of runner than is employed in irreversible or usual turbines of the same specific speed. In general, in high specific speed reversible propeller turbines, the transverse width of the passage at B′, Fig. 1, at the guide vanes or stay vanes, will be less than, or not greater than, the minimum width B at the runner.

In hydraulic power plants having sufficient pondage or reservoir capacity to permit the useful storage of power by pumping during off-peak load demands, while at the same time having considerable river flow for direct power generation, it will in general be the economical solution to install a limited number of the reversible units described above, in combination with a number of non-reversible turbines which would be shut down during the pumping operation, and would be utilized during a limited portion of the load cycle including peak load periods. That is, in such developments only a portion of the installed units would be of this reversible

type, depending on the economical amount of pumped water untilization in comparison with the total plant capacity required for direct power generation and peak load capacity.

It will of course be understood by those skilled in the art that various changes may be made without departing from the spirit of the invention as set forth in the appended claims.

I claim:—

1. A hydraulic reversible pump-turbine comprising entrance and discharge passages and a runner having blades forming passages therebetween, the walls of all of said passages being arranged whereby the flow areas thereof are so proportioned that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as by being within a range of values with a maximum limit of approximately 10° whereby in either direction of runner rotation the fluid flow throughout will be accelerated or decelerated at a relativly small rate.

2. A hydraulic reversible pump-turbine comprising entrance and discharge passages and a runner disposed therein provided with vane passages in which any rate of enlargement in one direction of flow is below a predetermined relatively small value, and the rate of enlargement of said passages in directions away from said runner is also of small value, said small value being between limits ranging from approximately 3° to 10° determined by laying out the passages as equivalent conical tubes and measuring the angle between the conical wall and the cone axis.

3. A hydraulic reversible pump-turbine comprising a runner having vane passages adapted to effect a relatively small rate of change in velocity of the fluid flow therethrough, and means providing entrance and discharge passages for said runner in either direction of runner rotation, said vane passages and entrance and discharge passages being so proportioned that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as between limits ranging from approximately 3° to 10° whereby said passages are adapted to effect relatively low rates of acceleration and deceleration, the discharge passage during pump operation which comprises the entrance passage during turbine operation being of volute formation.

4. A hydraulic reversible pump-turbine comprising a runner having vane passages with an angle of equivalent, conical flare falling below approximately 5°, and passages leading therefrom in opposite directions and adapted to provide gradual deceleration of fluid discharged in either direction of runner rotation.

5. A hydraulic reversible pump-turbine comprising a runner having vane passages in which the angle of equivalent conical flare falls below approximately 5°, and passages leading therefrom in opposite directions with an angle of equivalent conical flare falling below approximately 7°.

6. A hydraulic reversible pump-turbine comprising a runner having vane passages, entrance and discharge passages and guide vanes spaced to provide passages, all of said passages being proportioned so that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as between limits with a maximum value of approximately 10° to effect at the most only relatively small rates of change in the velocity of the flow therethrough during either direction of runner rotation.

7. A hydraulic reversible pump-turbine comprising a runner having vane passages in which the angle of equivalent conical flare falls below approximately 5°, means providing a gradual flow decelerating discharge passage for said runner in either direction of runner rotation, and guide vanes spaced to provide passages through which pump discharged flow passes with a relatively small rate of change of velocity, said discharge passage having an equivalent conical flare of preferably about 7°.

8. The combination as set forth in claim 7 further characterized by having said guide vanes adjustable.

9. A hydraulic reversible pump-turbine comprising a runner, flow passages therefor and guide vanes, the flow areas through said elements being proportioned so that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as between limits ranging from approximately 3° to 10° thereby to effect relatively small changes in the rate of any acceleration or deceleration of flow therethrough, and means whereby said runner may normally operate as a pump at a higher rate of speed than during turbine operation.

10. A hydraulic reversible pump-turbine comprising a runner having vane passages in which the equivalent angle of conical flare falls below approximately 5°, and flow passages therefor being adapted to effect a gradual rate of deceleration of fluid discharged during either turbine or pump operation, whereby said runner is adapted to normally operate in either direction of runner rotation at substantially the same normal speed.

11. A hydraulic reversible pump-turbine comprising a runner having vane passages in which the angle of equivalent conical flare falls below approximately 5°, entrance and discharge passages each adapted to effect relatively gradual deceleration of flow discharged from the runner, and angularly disposed guide vanes in said entrance passage.

12. The combination set forth in claim 10 further characterized by having guide vanes also in the flow passage which forms an entrance passage during turbine operation.

13. A hydraulic reversible pump-turbine comprising a runner, and entrance and discharge passages one of which is of materially greater axial extent than the other, each of said passages being proportioned so that if laid out as an equivalent conical tube the angle of the conical wall to the cone axis will be relatively small as between limits ranging from approximately 3° to 10° thereby to effect a gradual rate of deceleration of discharge flow during pump or turbine operation.

14. The combination in a hydraulic reversible pump-turbine comprising a high specific speed propeller type runner in which the vane passages have an angle of equivalent conical flare falling preferably below approximately 5°, and entrance and discharge passages each having an angle of equivalent conical flare falling below approximately 10° thereby to effect a gradual rate of change in the velocity of flow therethrough.

15. A hydraulic reversible pump-turbine comprising a mixed flow Francis type runner having vane passages proportioned so that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as between limits ranging from approximately 3° to 10° thereby to give at the most a gradual rate of change in the velocity of flow therethrough, and passages adapted to effect gradual deceleration of and efficient regain of energy from fluid discharged from said runner in either direction of runner rotation.

16. The combination set forth in claim 14 further characterized by having said discharge passage comprising a spreading annular passage.

17. The combination set forth in claim 14 further characterized by having said entrance passage in the form of a volute.

18. A hydraulic reversible pump-turbine comprising a low specific speed Francis runner having vane passages of such cross-sectional area that a relatively small rate of change in the velocity of flow therethrough is obtained, and means forming a discharge passage for said runner in either direction of runner rotation and adapted to effect a relatively low rate of deceleration of the discharged fluid in each case, all of said passages being so proportioned that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as between limits ranging from approximately 3° to 10°.

19. A hydraulic reversible pump-turbine comprising a runner and flow passages leading to and away from the runner each adapted to effect relatively gradual deceleration of flow received from the runner, said runner having vane passages in which the opposing walls at any section normal to the flow diverge from each other at angle not substantially greater than 10° with respect to the axis of the passage, thereby said runner is adapted to operate efficiently with flow in either direction relatively thereto.

20. A hydraulic reversible pump-turbine comprising a runner and flow passages leading to and away from the runner each adapted to effect relatively gradual deceleration of flow received from the runner, said runner having a passage between successive vanes such that in an equivalent rectangular tube having a rectified centerline of length equal to the center flow line of said passage and having the same transverse areas at corresponding points of the centerline, the opposing walls at any section diverge from each other at an angle not substantially greater than 10° with respect to the axis of the passage, whereby said runner is adapted to operate efficiency with flow in either direction relatively thereto.

21. A hydraulic reversible pump-turbine adapted to operate as either a turbine or pump, having a runner the passages between the vanes of which gradually enlarge in the direction of flow when pumping, and a discharge passage from said runner when operating as a turbine, said discharge passage being formed with gradually enlarging cross-sections receding from the runner and adapted for efficient deceleration of flow received from the runner, all of said passages being so proportioned that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as between limits ranging from approximately 3° to 10°.

22. A reversible pump-turbine installation comprising a headrace and a tailrace, and a reversible hydraulic machine, an electrical generator-motor directly connected thereto, operable at times as a turbine-generator unit and at other times as a pump-motor unit, said hydraulic machine having a runner and a draft tube receiving the flow therefrom when operating as a turbine, and said draft tube being formed to provide gradual deceleration of the flow therethrough and efficiently to regain velocity head of the runner discharge, said runner being placed at an elevation above the level in the tailrace which is less than that permissible in a non-reversible turbine of the same specific speed by an amount at least equal to the loss of head in the draft tube during turbine operation.

23. A high specific speed reversible pump-turbine comprising, in combination, means forming entrance and discharge passages having a gradual rate of flare falling within

a limit of approximately 7° based on an equivalent conical passage, and a runner having a hub carrying a small number of adjustable blades forming passages therebetween with an equivalent conical flare falling within approximately 5°.

24. A high specific speed reversible pump-turbine comprising, in combination, means forming entrance and discharge passages having a gradual rate of flare falling within a limit of approximately 7° based on an equivalent conical passage, a propeller type runner having a hub and blades carried thereby for adjustment about axes intersecting the hub axis, said blades forming passages therebetween with an equivalent conical flare falling within approximately 5° and means for simultaneously adjusting said blades while the runner is rotating.

25. A hydraulic reversible pump-turbine comprising entrance and discharge passages, and a propeller type runner having adjustable blades with flow passages therebetween, all of said passages having their flow areas so proportioned that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as by being within a range of values with the maximum limit of approximately 10° whereby during normal operation of the apparatus either as a pump or turbine the flow throughout is accelerated or decelerated at a relatively small rate.

26. The combination in a reversible pump-turbine, of a runner provided with vanes which are everywhere forwardly inclined to the radial when operating as a turbine and rearwardly inclined when operating as a pump, and entrance and discharge passages for said runner, each of said latter passages and the passages formed between the runner vanes so proportioned that if laid out as equivalent conical tubes the angle of the wall to the cone axis will be relatively small as between limits ranging from 3° to 10°.

27. The combination set forth in claim 26 further characterized in that the working face of said runner vanes is nowhere concave in sections taken in the direction of flow.

28. The combination set forth in claim 6 further characterized in that the guide vanes are relatively thin and gradually tapered in both directions of discharge therefrom as during either pump or turbine operation.

29. The combination set forth in claim 1 further characterized by the provision of an electrical machine connected to the runner shaft and adapted to operate as a motor during pump operation at a speed in excess of that at which it operates as a generator during turbine operation.

30. The combination set forth in claim 1 further characterized by the provision of an electrical machine connected to the runner shaft and adapted to operate as a motor during pump operation at a speed in excess of that at which it operates as a generator during turbine operation, said electrical machine being adapted to obtain its change in speed by pole-changing provisions.

31. The combination set forth in claim 1 further characterized by the provision of an electrical machine connected to the runner shaft and adapted to operate as a motor during pump operation at a speed in excess of that at which it operates as a generator during turbine operation, said electrical machine being adapted to obtain its change in speed by pole-changing provisions, said speed for turbine operation bearing a ratio to that for pump operation approximately equal to the square root of the product of the overall efficiencies of turbine and pump operation.

32. The combination set forth in claim 1 further characterized by the provision of an electrical machine connected to the runner shaft and adapted to operate as a motor during pump operation at a speed in excess of that at which it operates as a generator during turbine operation, said electrical machine being adapted to obtain its change in speed by pole-changing provisions, said speed for turbine operation bearing a ratio to that for pump operation of not less than approximately 85 per cent. as nearly as obtainable by said pole-changing provisions.

33. A hydraulic reversible pump-turbine comprising entrance and discharge passages provided with a throat portion, and a runner disposed adjacent said throat and provided with blade passages, all of said passages being so proportioned that if laid out as equivalent conical tubes the angle of the wall to the cone axis will be relatively small as between limits ranging from 3 to 10° while the outer surface of the entrance and discharge passages adjacent the runner has a relatively large radius of curvature when viewed in meridian section, said surface curving from an axial direction toward radial and having a radius of curvature not less than substantially two-thirds of the throat radius.

34. The combination in a reversible pump-turbine, of a low specific speed reaction type runner provided with a relatively small number of blades in comparison to a number employed in a runner used only for turbine operation and of the same specific speed, said blades being everywhere forwardly inclined to the radial when operating as a turbine and rearwardly inclined when operating as a pump, and entrance and discharge passages for said runner, each of said latter passages and passages formed between the runner blades so proportioned that if laid out as equivalent conical tubes the angle of the wall to the cone axis will be relatively small as between limits ranging from 3 to 10°.

35. The combination in a reversible pump-turbine comprising a runner provided with

blades which are nowhere concave on their faces in sections taken substantially normal to the runner axis, and entrance and discharge passages for said runner, both of said latter passages and the passages formed between the runner blades so proportioned that if laid out as equivalent conical tubes the angle of the wall to the cone axis will be relatively small as between limits ranging from 3 to 10°.

36. A hydraulic reversible pump-turbine comprising entrance and discharge passages and a runner having blades forming passages therebetween, the walls of all of said passages being arranged whereby the flow areas thereof are so proportioned that if laid out as equivalent conical tubes the angle of the conical wall to the cone axis will be relatively small as by being within a range of values with a maximum limit of approximately 10° whereby in either direction of runner rotation the fluid flow throughout will be accelerated or decelerated at a relatively small rate, and governor controlled guide vanes for variably controlling the flow of fluid to said runner during turbine operation.

37. The combination in a reversible pump-turbine comprising entrance and discharge passages, and a runner having blade passages, all of said passages being so proportioned as to effect a relatively gradual rate of acceleration or deceleration during pump or turbine operation and said runner passages expanding in one plane and contracting in perpendicular section with divergence in either drection of flow not exceeding substantially 10° based on an equivalent conical tube.

38. The combination in a hydraulic reversible pump-turbine of the high specific speed propeller type comprising flow passages and a propeller runner having a relatively small number of overlapping blades, said flow passages and runner passages being so proportioned that if laid out as equivalent conical tubes the angle of the wall to the cone axis will be relatively small as between limits ranging from 3 to 10°.

39. A reversible hydraulic pump-turbine comprising a runner provided with a hub and blades carried thereby, and flow passages leading to and away from the runner each adapted to effect relatively gradual deceleration of flow received from the runner, said runner having blade passages in which the opposing blade surfaces diverge from each other at an angle not greater than substantially 20° during pump operation, while the opposing walls formed by the runner hub and outer wall of the flow passage gradually converge in the direction of flow.

40. A reversible hydraulic pump-turbine comprising a runner and flow passages leading to and away from the runner each adapted to effect relatively gradual deceleration of flow received from the runner, said runner having vane passages which in meridian section have opposing walls gradually converging in the dircetion of pump flow while in a flow section perpendicular to said first section the opposing walls diverge in the direction of pump flow, said divergence not being substantially greater than 20°.

41. A reversible hydraulic pump-turbine comprising a runner and flow passages leading to and away from the runner each adapted to effect relatively gradual deceleration of flow received from the runner, said runner having a passage between successive vanes such that in an equivalent rectangular tube having a rectified centerline of length equal to the center flow line of said passage and having the same transverse sections at corresponding points of the centerline, the walls corresponding to the runner hub and opposing wall gradually converge in the direction of pump flow, while the walls corresponding to the vane surfaces diverge from each other in the same direction of flow at an angle not substantially greater than 20°.

42. A reversible hydraulic pump-turbine comprising, in combination, entrance and discharge passages, and a propeller type runner having adjustable blades, said entrance and discharge passages and the runner blade passages all being so proportioned that if laid out as equivalent conical tubes the angle of the wall to the cone axis will be relatively small as between limits ranging from 3 to 10° whereby the flow throughout will be accelerated or decelerated at a relatively gradual rate for either pump or turbine operation.

43. A reversible hydraulic pump-turbine comprising, in combination, entrance and discharge passages, a propeller type runner having adjustable blades, said entrance and discharge passages and the runner blade passages all being so proportioned that if laid out as equivalent conical tubes the angle of the wall to the cone axis will be relatively small as between limits ranging from 3 to 10° whereby the flow throughout will be accelerated or decelerated at a relatively gradual rate for either pump or turbine operation, and fixed guide vanes disposed in the passage which constitutes the entrance passage during turbine operation, said guide vanes being inclined so as to impart whirl to the fluid flowing to said runner.

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