

- [54] **FLUID TURBOMOTOR**
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[51] Int. Cl. **F01d 17/00, F01d 1/08**
[58] Field of Search..... **415/149, 150, 151, 415/152, 164, 143, 158, 163, 165**

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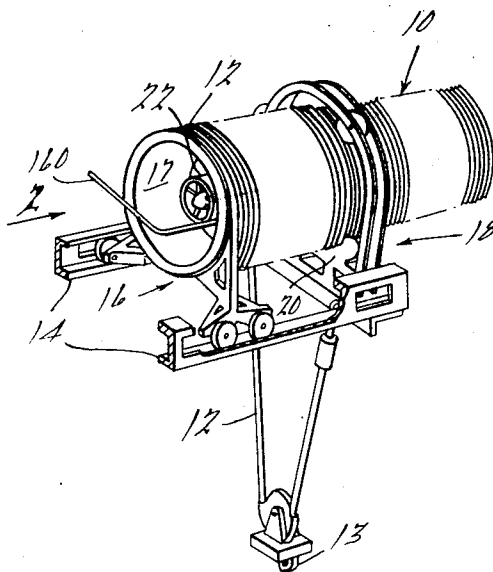
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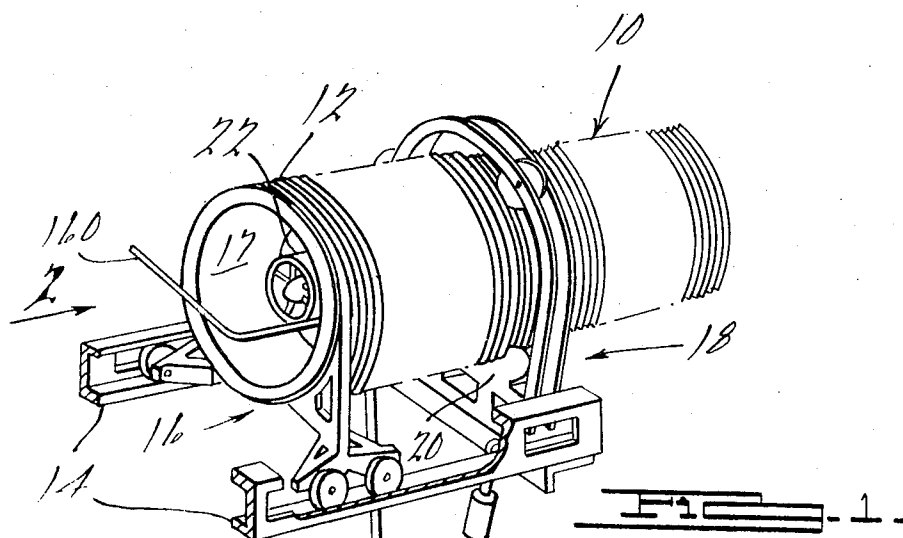
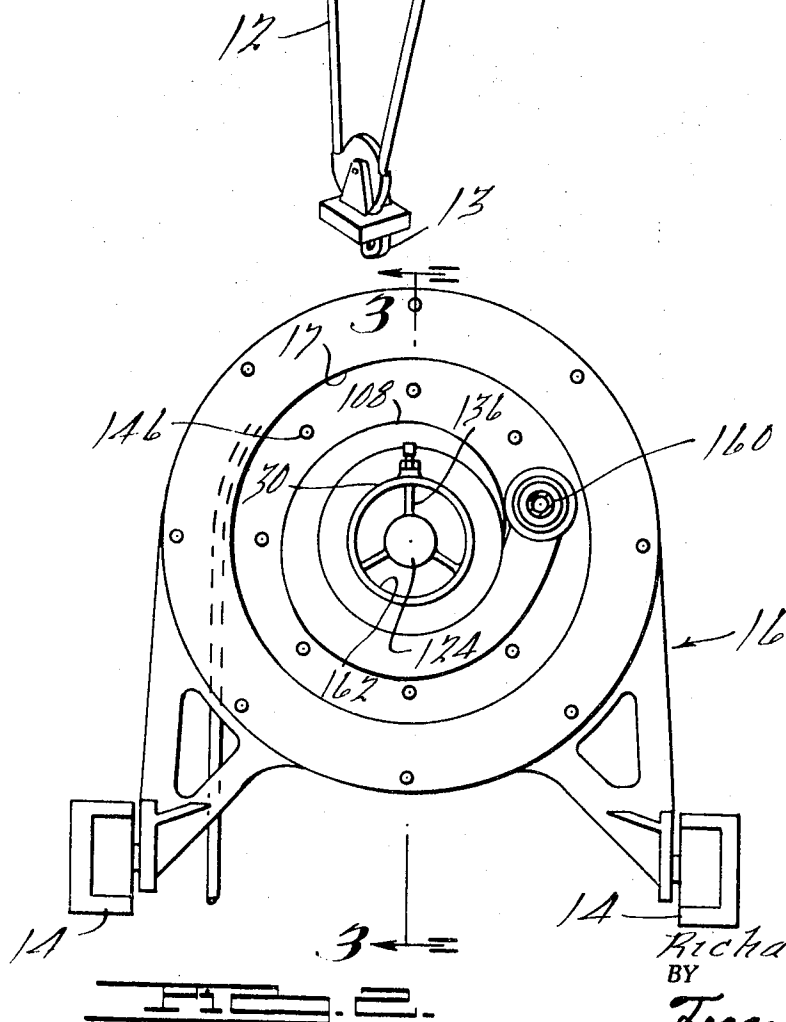
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[57] **ABSTRACT**

A fluid turbomotor comprising a radial turbine wheel of the radial inflow type, a single stage axial turbine wheel on the same shaft as the radial wheel, and a blade type nozzle structure interposed between the radial and axial wheels, wherein the working fluid is introduced at the radial periphery of the radial wheel, is discharged at an axial face of the radial wheel into the nozzle structure, and is directed by the blades of the nozzle structure into the axial turbine wheel; also disclosed is a novel nozzle ring structure for the radial turbine wheel comprising two side-by-side blade sets of different angular orientation which are selectively moved into working relationship to the inflow of the radial wheel to vary the direction at which the working fluid is delivered to the radial wheel. The invention turbomotor is disclosed as the power source in an overhead cable hoist structure.

6 Claims, 14 Drawing Figures





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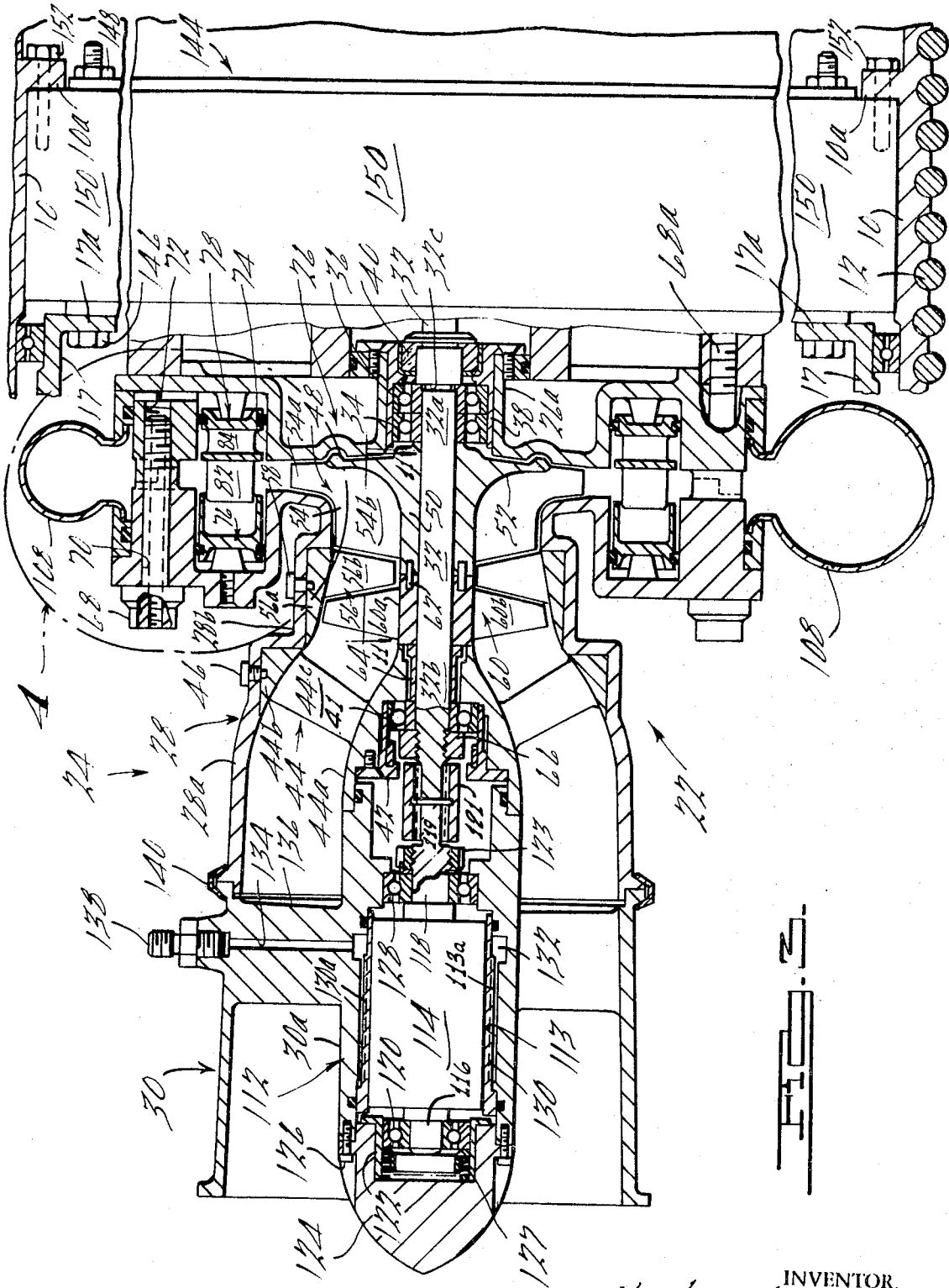
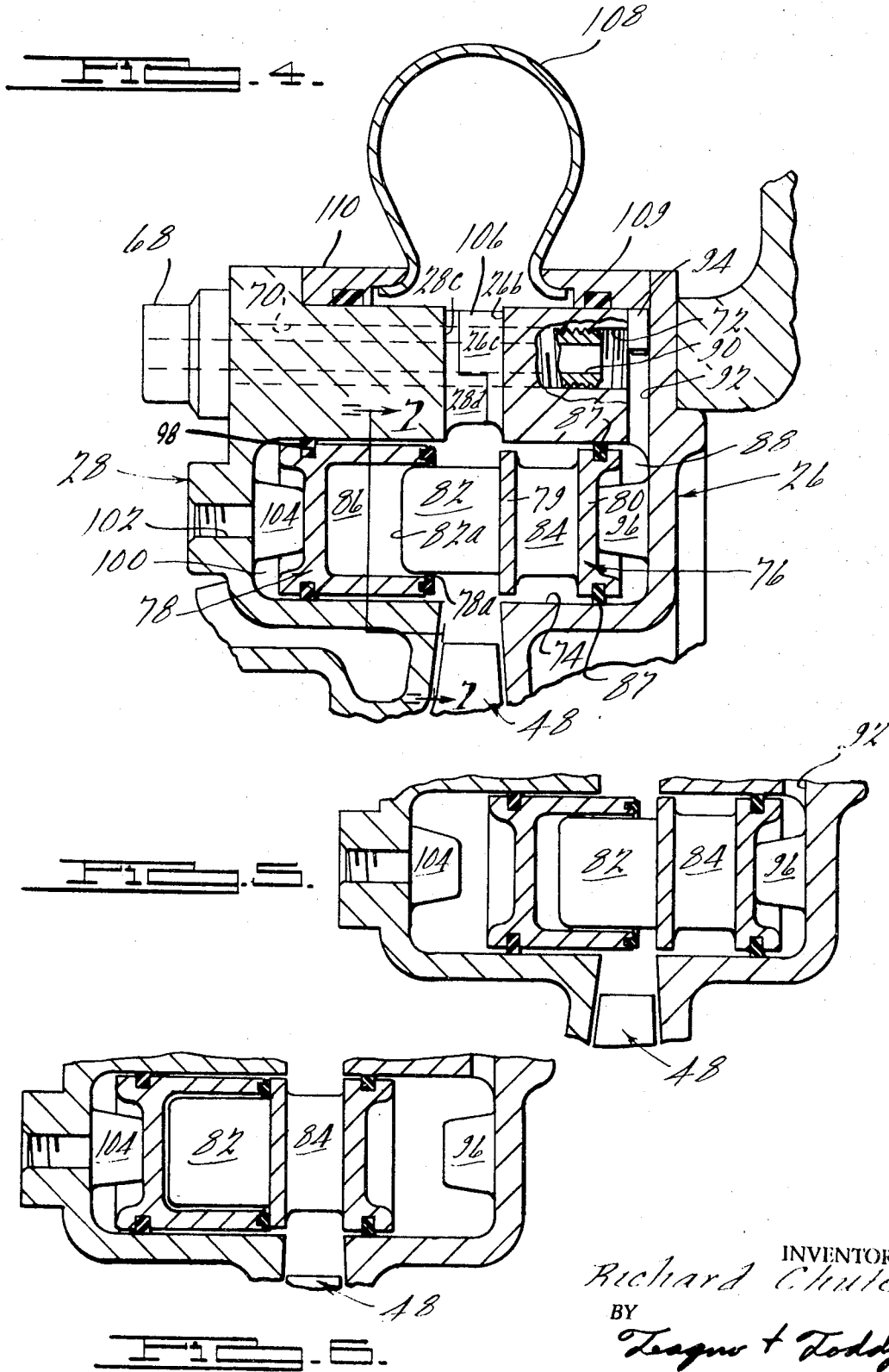


FIG. 2

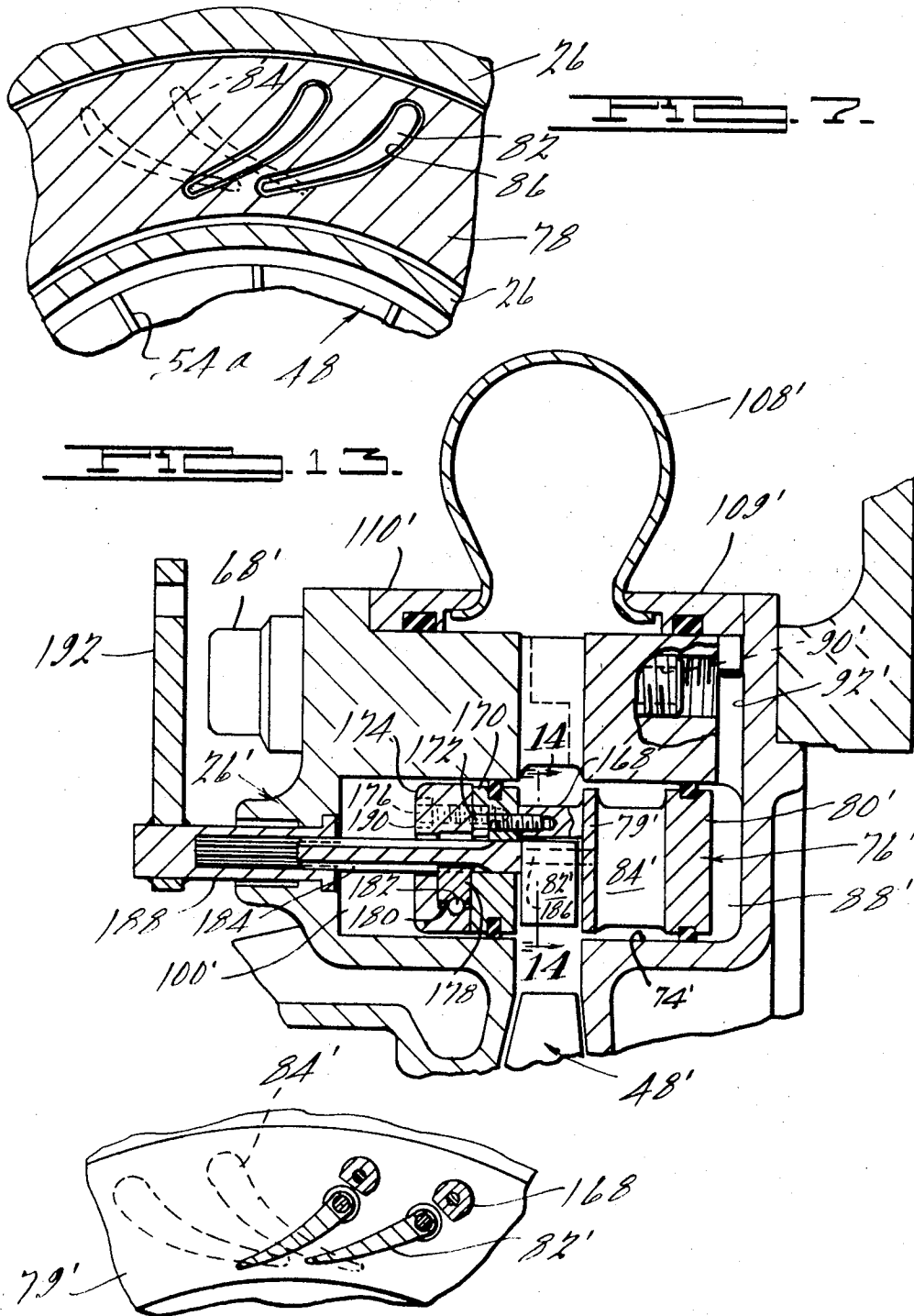
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FIG. 11.

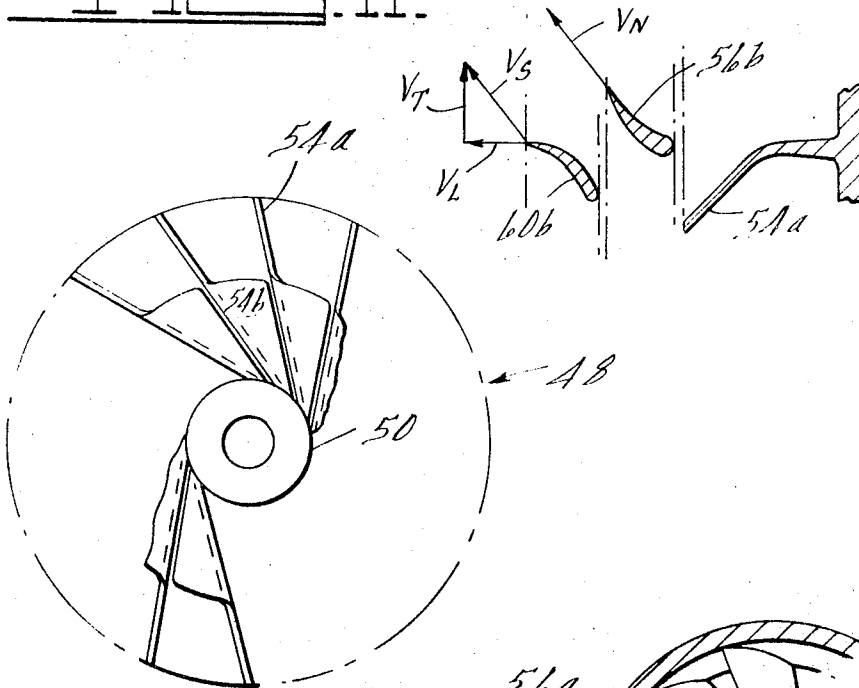


FIG. 8.

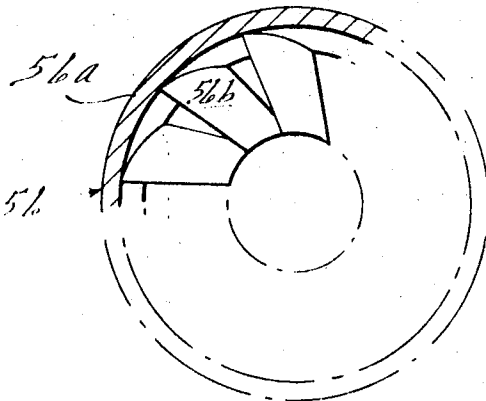


FIG. 9.

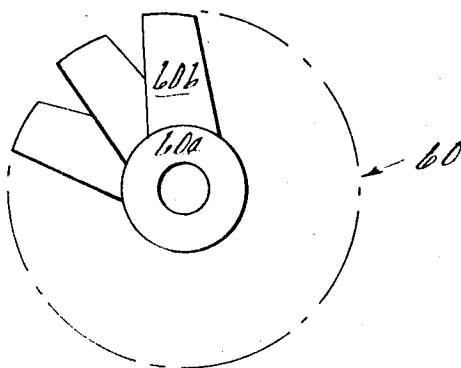
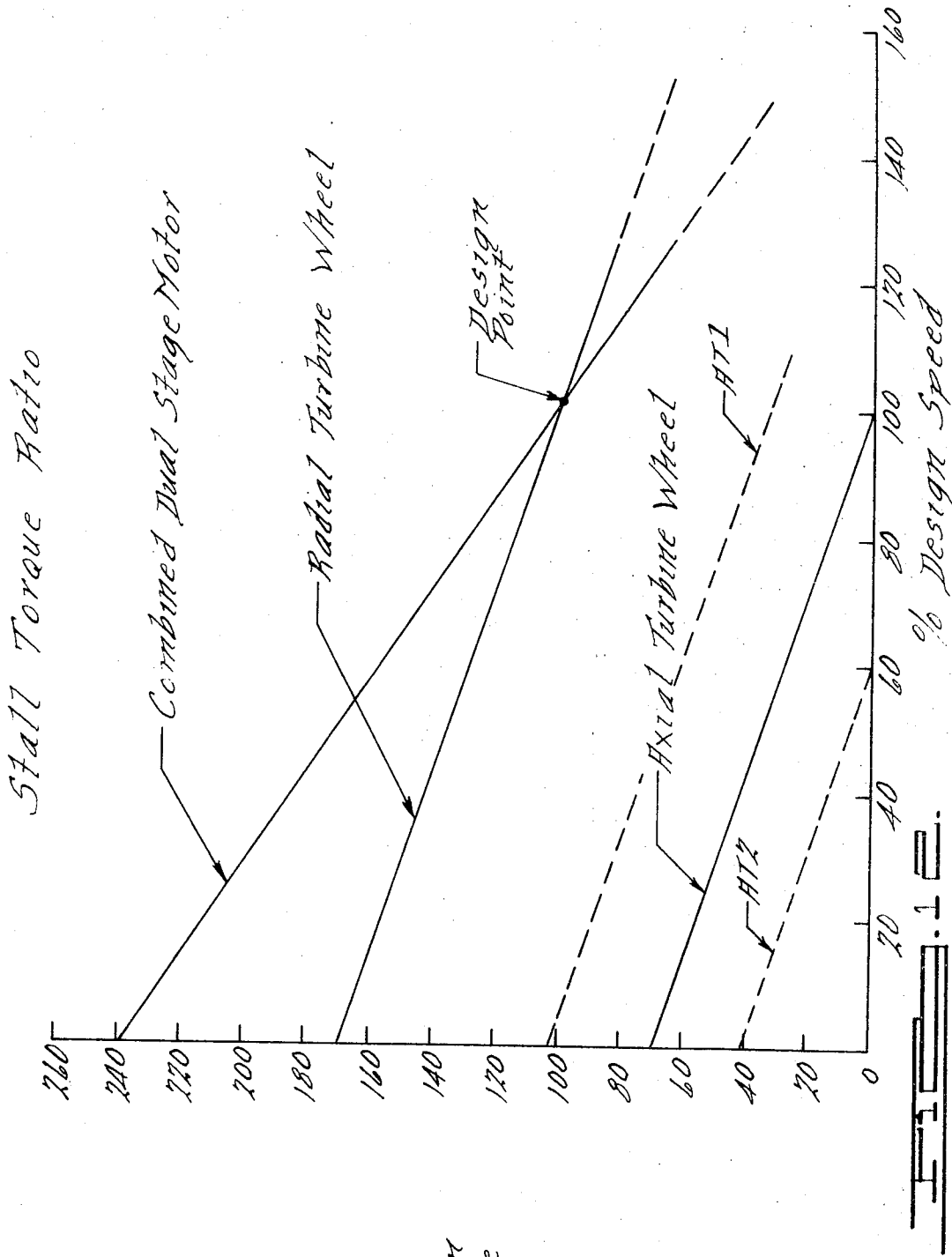


FIG. 10.

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FLUID TURBOMOTOR

BACKGROUND OF INVENTION

This invention relates to improvements in fluid turbomotors. Various forms of turbomotors have been proposed and/or made commercially available. While these turbomotors, considered in each case with respect to their intended purpose, have been generally satisfactory, none has offered the combination of a relatively high stall torque ratio and reasonably efficient operation in a reverse mode. Furthermore, none of the prior art turbomotors has offered a completely satisfactory mechanism for modulating the working fluid inflow to the turbomotor. Also, none of the prior art turbomotors is capable of satisfying all of the unique power and controllability requirements of a cable type hoist.

SUMMARY OF THE INVENTION

It is an object of the present invention to provide an improved fluid turbomotor.

A more specific object is to provide a fluid turbomotor having a relatively high stall torque ratio coupled with an ability to operate with reasonable efficiency in a reverse mode.

Another specific object is to provide an improved fluid modulating mechanism for controlling the delivery of the working fluid to a radial turbine wheel.

Another specific object is to provide a turbomotor which is uniquely suited for use as the power source of a cable type hoist.

According to an important feature of the invention, the turbomotor comprises a radial turbine wheel and an axial turbine wheel coupled to a common shaft with a first nozzle assembly surrounding the radially outer periphery of the radial wheel and a second nozzle assembly interposed between the turbines; the working fluid is directed by the first nozzle assembly into the radial periphery of the radial wheel, is discharged at an axial face of the radial wheel into the second nozzle assembly, and is directed by the second nozzle assembly into the blades of the axial turbine. In the disclosed embodiment, the second nozzle assembly is of the blade type and the blades of the axial turbine wheel are positioned at an angle such that, at the rated speed of the turbomotor, the vectorial sum of the relative fluid leaving velocity of the blades of the axial turbine and the blade tangential velocity of the axial turbine blades is substantially equal to the vector of the fluid leaving the blades of the second nozzle assembly. With this vectorial relationship, at rated speed of the turbomotor, the axial turbine will windmill and the radial wheel will supply substantially all of the torque output of the motor.

According to another important feature of the invention, the nozzle assembly for the radial turbine wheel comprises two side-by-side blade sets with different blade angles, and the turbomotor includes means for shifting the nozzle assembly between a first position in which one of the blade sets is aligned with the outer periphery of the turbine wheel and the other blade set is displaced with respect to the wheel periphery, and a second position in which the other blade set is aligned with the outer periphery of the wheel and the first blade set is displaced with respect to the wheel periphery. In the disclosed embodiments, the one blade set is arranged to rotate the turbine wheel in one direction and

the other blade set is arranged to rotate the wheel in the reverse direction.

Other objects and features of the invention will be apparent from the detailed description of the preferred embodiments and from the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. 1 is a perspective view of a cable hoist mechanism embodying a turbomotor according to the invention;

FIG. 2 is an end view of the hoist mechanism of FIG. 1 looking in the direction of the arrow 2 in FIG. 1;

FIG. 3 is a longitudinal, fragmentary cross-sectional view taken on line 3—3 of FIG. 2;

FIG. 4 is an enlarged view of the nozzle structure seen in circle 4 of FIG. 3;

FIGS. 5 and 6 are similar to FIG. 4 but showing moved positions of the nozzle structure;

FIG. 7 is a cross-sectional view taken on line 7—7 of FIG. 4;

FIGS. 8, 9 and 10 are respective isolated end views, looking in the direction of the arrows 8, 9 and 10 of FIG. 3, of a radial turbine wheel, axial turbine wheel, and axial turbine nozzle assembly embodied in the invention turbomotor;

FIG. 11 is a fragmentary cross-sectional view taken on line 11—11 of FIG. 3;

FIG. 12 is a stall torque ratio graph for the invention turbomotor;

FIG. 13 illustrates a modified form of nozzle structure; and

FIG. 14 is a cross-sectional view taken on line 14—14 of FIG. 13.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

The invention turbomotor is disclosed as the power source for a cable type hoist. The hoist seen in FIG. 1 includes a drum 10, a cable 12 wrapped in helical fashion around drum 10, a load hook 13 carried in reeved fashion by cable 12, a pair of tracks 14 for supporting the drum, a carriage 16 rollably guiding on tracks 14 and including a journal portion 17 rotatably supporting the drum, and a nut structure 18 including a partial ball nut 20 guiding in a vacant portion of the helical drum groove to thread the drum leftwardly or rightwardly along the track structure in response to clockwise or counterclockwise rotation of the drum so that the pay-out or pay-in location of cable 12 remains fixed as the cable is unwound from or wound on the drum. Details of the construction and operation of the hoist of FIG. 1 are shown and described in copending U.S. Pat. application Ser. No. 76,514 filed on Sept. 29, 1970 and assigned to the assignee of the present invention.

A turbomotor seen generally at 22 in FIG. 1 is positioned within the hollow end of drum 10 and is arranged to selectively drive the drum in forward and reverse directions to pay-in or pay-out the cable. Turbomotor 22 is seen in end view in FIG. 2 and in longitudinal cross section in FIG. 3, to which reference is now made.

Turbomotor 22 includes a housing seen generally at 24. Housing 24 is a three-part structure including an end housing 26, a central housing 28, and a brake housing 30. A central output shaft 32 is journaled in housings 26 and 28; end 32a of shaft 32 is journaled in dou-

ble ball bearings 34, 36 positioned in a sleeve 38 received in the central hub portion 26a of end housing 26. An oil seal 40 is also received in sleeve 38 and sealingly engages shaft end 32a. The other shaft end 32b is journaled in a ball bearing 41 positioned in a sleeve 42. Sleeve 42 is received in the hub portion 44a of a wheel member 44 rigidly positioned in bell-mouth portion 28a of central housing 28. Wheel member 44 includes an outer annulus or rim portion 44b interconnected to hub portion 44a by spokes 44c; a set screw 46 carried by housing portion 28a engages rim portion 44b to rigidly position wheel member 44 within the turbomotor housing.

A radial turbine wheel 48 is keyed to shaft 32 adjacent shaft end 32a. Radial wheel 48 is of the radial inflow type. As seen in FIGS. 3, 8 and 11, wheel 48 includes a central hub portion 50, an end wall portion 52 at the right-hand axial face of the wheel as viewed in FIG. 3, and a plurality of vanes or blades 54 formed integrally with portions 50 and 52. Each vane 54 includes a radial inflow portion 54a extending from the radially outer periphery of the wheel inwardly at an angle generally tangent to the outer periphery of hub portion 50, and an axial discharge portion 54b which is bent over with respect to radial inflow portion 54a and defines, in cooperation with an adjacent discharge portion 54b, an axial discharge passage for discharge of the working fluid at the left-hand axial face of wheel 48 as viewed in FIG. 3.

Fluid leaving radial wheel 48 enters an axial turbine wheel nozzle structure seen generally at 56. Nozzle structure 56, as best seen in FIGS. 3, 10 and 11, includes an annulus portion 56a secured in neck portion 28b of housing 28 by set screw 58, and a plurality of inwardly directed blades 56b spaced circumferentially about the lengthwise axis of shaft 32 but defining a central opening for passage of the pilot end of radial wheel hub portion 50. Blades 56b may number, for example, between 15 and 20 and may, for example, be inclined, as seen in FIG. 10, at an angle of approximately 15° with respect to a radius of annulus 56a.

Working fluid is directed by nozzle assembly 56 into an axial turbine wheel seen generally at 60. Axial turbine wheel 60, as best seen in FIGS. 3, 9, and 11, includes a hub portion 60a keyed to shaft 32 and a plurality of blades or vanes 60b radiating outwardly from hub portion 60a. Blades 60b may number, for example, between 15 and 20 and may, for example, be inclined, as seen in FIG. 9, at an angle of approximately 15° with respect to a radius of hub 60a. The rightward pilot end (FIG. 3) of hub portion 60a is secured to the left-hand pilot end of hub portion 50 by a plurality of pins 62. A spacer 64 is positioned on shaft 32 between the inner race of bearing 41 and the left face of hub portion 60a and a nut 66 coacts with threads on shaft end 32b to snug ball bearing 41, spacer 64, hub 60a, hub 50, and ball bearings 34, 36 up against a shoulder 32c on shaft 32.

Housings 26 and 28 are rigidly secured together by a plurality of bolts 68 which pass through smooth bores 70 in housing 28 for threaded receipt in tapped bores 72 in housing 26. Thus secured together, and with particular reference now to FIGS. 4-7, housings 26 and 28 coact to define an annular chamber 74 slideably receiving, in piston fashion, an annular nozzle assembly 76 and a throttling ring 78.

Nozzle assembly 76 includes a support ring 79, a support ring 80, a first nozzle blade set comprising a plurality of blades 82 circumferentially spaced about the central turbomotor axis and each disposed at a first angle (FIG. 7) with respect to the orbit of radial wheel 48, and a second nozzle blade set comprising a plurality of blades 84 circumferentially spaced about the central turbomotor axis and each disposed at a second angle with respect to the radial wheel orbit. The blades 82 of the first blade set are each integral at one end with support ring 79 and project axially therefrom in cantilever fashion to present a free edge 82a; and the blades 84 of the second blade set extend in bridge fashion between support rings 79 and 80.

Throttling ring 78 is in the form of an annular block and presents an annular face 78a in confronting relation to the free blade edges 82a; ring 78 has a plurality of axially extending pockets 86 formed therein each of a shape to snugly receive a blade 82 and each opening in annular face 78a.

Sealing rings or gaskets 87 carried by support ring 80 serve to define a sealed annular subchamber 88 to the right (FIG. 4) of support ring 80, and pressurized control fluid may be supplied to subchamber 88 through a bore 90 provided in one or more bolts 68 communicating with a cross bore 92 opening at its radially inner end in chamber 88 and plugged at its radially outer end by a plug 94. Stop means 96 constituted by one or more bosses formed integral with housing 26 coacts with support ring 80 to define the extreme rightward position (FIG. 4) of nozzle assembly 76.

Sealing rings or gaskets 98 carried by throttling ring 78 serve to define a sealed annular subchamber 100 to the left of ring 78, and pressurized control fluid may be supplied to chamber 100 through a bore 102 in the adjacent wall portion of housing 28. Stop means 104 constituted by one or more bosses integral with housing 28 coacts with the left face of throttling ring 78 to define the extreme leftward position of the throttling ring.

Working fluid is delivered to nozzle assembly 76 through an annular opening 106 defined between axially spaced annular walls 26b and 28c. Space 106 is defined and maintained by interfitting bosses 26c and 28d formed integral with housings 26 and 28 and passing bolts 68. Working fluid is in turn delivered to space 106 through a scroll 108 encircling space 106 and secured to housings 26, 28 by a pair of annuluses 109, 110 fitted over the radially outer periphery of housings 26, 28 respectively.

The invention turbomotor includes a brake assembly seen generally at 112 in FIG. 3. Brake assembly 112 includes an enlarged drum 114 having trunnions 116 and 118, journalling the drum in a central chamber 113 defined within hub portion 30a of brake housing 30. Trunnion 116 is journaled in a ball bearing 120 received in a sleeve 122 positioned in an end cap 124 secured by screws 126 to the left end of the hub portion 30a. Bearing 120 is maintained in a snug, preloaded condition by spring means 127. Trunnion 118 is journaled in a ball bearing 128 received in hub portion 30a and includes a splined extension 119 which is coupled to the splined left end of turbomotor shaft 32 by a splined sleeve coupling 121. Nut 123 on a threaded portion of trunnion 118 maintains bearing 128 in a snug condition.

A sleeve 130 of phenolic-asbestos material (e.g., "Pyrotex" made by Raybestos-Manhattan Inc. of Pas-

saic, New Jersey) loosely surrounds drum 114 and is adapted to be contracted to brakingly grasp the drum by hydraulic pressure introduced into the sealed annular chamber 132 between the exterior surface of the sleeve and the surface of housing chamber 113. Hydraulic fluid is introduced into chamber 132 through a bore 134 extending through a spoke 136 of brake housing 30 for connection to a suitable hose fitting 138. The exterior surface of sleeve 130 has grounding splines 130a which coact with mating splines 113a on the cylindrical wall surface of chamber 113 to ground the sleeve through the brake housing in both the relaxed and the contracted condition of sleeve 130. Further details of the construction and operation of contracting sleeve 130 may be found in U.S. Pat. application Ser. No. 763,784 filed on Sept. 30, 1968 and assigned to the assignee of the present invention. The entire brake assembly is a modular unit and may be disconnected from the turbomotor by removal of a clamp ring 140 coupling housing units 28 and 30.

The output shaft 32 of the turbomotor drives hoist drum 10 through a suitable gear reducer seen generally at 144. Gear reducer 144 is secured to the journal portion 17 of carriage 16 by a plurality of bolts 146 (see also FIG. 2) passing through an inturned flange 17a of carriage journal portion 17 and through gear reducer 144 for coaction with nuts 148 and turbomotor 22 is in turn secured to gear reducer 144 by the extended portion 68a of those screw bolts 68 that are not hollowed out for transmission of pressure fluid to subchamber 88. Turbomotor 22 is thus, in effect, carried by carriage 16. Gear reducer 144, for example, may comprise a Breeze Cyclocentric Drive available in various reduction ratios from Breeze Corporation Inc., Union, New Jersey. Cyclocentric Drives are available either with or without a rotational direction change between the input shaft and the output ring gear (150); the present disclosure contemplates a Cyclocentric Drive in which the output ring gear 150 rotates in a reverse direction from input shaft 32. Output ring gear 150 is drivingly coupled to drum 10 by a plurality of screw bolts 152 passing through an internal flange 10a on drum 10 for threaded engagement with suitable tapped bores in output ring gear 150. Drum 10 is thus driven through gear reducer 144 at a rotational speed which is reduced with respect to that of turbomotor output shaft 32 by a degree determined by the reduction ratio provided by reducer 144.

OPERATION OF PREFERRED EMBODIMENT

The operation of the invention turbomotor will now be described with reference to its use in the disclosed hoist apparatus. When the hoist drum is not being rotated to pay-in or pay-out the cable, braking sleeve 130 is maintained in a contracted condition by hydraulic fluid introduced into chamber 132 through fitting 138. Sleeve 130 thus acts as a static brake for the hoist. When it is desired to rotate drum 10, hydraulic pressure is relieved in chamber 132 to release the static brake and pressurized working fluid (e.g. air) is supplied to scroll 108 through a suitable hose 160 (FIG. 1); scroll 108 distributes the working fluid around the periphery of the motor and the fluid passes selectively through annular space 106 into chamber 74 where it coacts with nozzle assembly 76. In the absence of pressurized control fluid (e.g. air) in chambers 88 and 100, nozzle assembly 76 and throttling ring 78 assume their

splayed apart positions of FIG. 4 in which each is pushed up against its respective stop means by the pressure of the working fluid. Blades 82 are thus the working blades at this time and determine the angle at which the working fluid will be delivered to radial turbine wheel 48. In the disclosed hoist embodiment, blades 82 are placed at an angle such that it causes the drum to be rotated in a paying-in direction (clockwise in FIG. 2) to lift a load on hook 13. The angle of blades 82 is best seen in FIG. 7; it will be seen that blades 82 function to direct working fluid against the radial inflow portions 54a of radial turbine wheel blades 54 at an angle to cause wheel 48 to rotate in counterclockwise direction as viewed in FIG. 7.

When it is desired to pay-out the cable, i.e., reverse the direction of rotation of drum 10, turbomotor 22 is reversed by directing pressurized control fluid into chamber 88 through bores 90 and 92 to move nozzle assembly 76 to the left to the position shown in FIG. 6 in which blades 84 are now in working position with respect to the radial turbine wheel blades. Blades 84, as best seen in FIG. 7, are angled in an opposite sense to blades 82 and function to direct the working fluid against turbine blades 54 in a sense to produce clockwise rotation of turbine wheel 48 as viewed in FIG. 7. The leftward movement of nozzle assembly 76 in chamber 74 is made possible by the receipt of blades 82 in pockets 86 in blocking ring 78. When it is desired to pay-in the cable again, control pressure in chamber 88 is relieved and control pressure is established in chamber 100; blocking ring 78 is thus moved rightward in chamber 74 to push nozzle assembly 76 back to its FIG. 5 position up against stop 96. When nozzle assembly 76 has been pushed to its stop position, control pressure is relieved in chamber 100 and the pressure of the working fluid passing through the nozzle structure acts to push blocking ring leftward to its stop position so that parts 76, 78 again assume the positions of FIG. 4.

If it is desired, while operating in the paying-in position of FIG. 4, to modulate the nozzle structure to vary the flow of working fluid to the radial turbine, a control fluid pressure is established in chamber 100 of a magnitude to overcome the pressure of the working fluid and slide throttling ring 78 rightward in chamber 74; as ring 78 moves rightward, blades 82 are selectively and progressively received or "swallowed up" in pockets 86 so that the effective working area of blades 82 is selectively and progressively reduced. In the FIG. 5 configuration, for example, the working area of blades 82 has been reduced to a fraction of their working area in FIG. 4 by the rightward "swallowing" movement of blocking ring 78. Blocking ring 78 may be selectively positioned anywhere along the width of blades 82 to provide an infinite number of modulated positions with such selective positioning being accomplished by selective control of the pressure in chamber 100 relative to the pressure of the working fluid passing through the nozzle structure.

Whether operating in the paying-in mode of FIG. 4 or the paying-out mode of FIG. 6, the working fluid directed by the nozzle blades into radial turbine wheel 48 flows through radial inflow portions 54a and thence through the axial outflow passages defined between adjacent axial discharge blade portions 54b for discharge at the left axial face (FIG. 3) of the radial turbine wheel into axial turbine wheel nozzle structure 56. The blades 56b of nozzle structure 56 in turn directs the working

fluid against the blades 60b of axial turbine wheel 60 and the working fluid leaving turbine wheel 60 flows around the hub 44 a of wheel 44 and around the hub 30a of housing 30 for discharge through annular opening 162.

According to an important feature of the invention, the blades of the axial turbine wheel and the axial turbine wheel nozzle are relatively positioned such that the axial turbine "windmills" at the design point of the turbomotor. Specifically, and with particular reference to FIG. 11, the blades 60b of the axial turbine are positioned at an angle such that, at the rated speed of the turbomotor, the vectorial sum V_s of the relative fluid leaving velocity V_L of the axial turbine blades and the blade tangential velocity V_T of the axial turbine blades is substantially equal to the vector V_N of the fluid leaving nozzle blades 56b. Thus, at rated or design speed of the turbometer, the working fluid in passing through the axial turbine wheel undergoes no change in direction and, disregarding friction losses, undergoes no change in pressure so that the working fluid gives up substantially no work to the axial turbine and the axial turbine simply "windmills," contributing no torque to the torque output of the turbomotor. In the disclosed embodiment, wherein the turbomotor has a design speed of approximately 35,000 RPM, the axial turbine blades 60b are arranged such that the relative fluid leaving velocity V_L has a direction substantially parallel to the axis of the turbomotor and the axial turbine nozzle blades are arranged such that the nozzle blade tangential velocity V_N has a direction at a significant angle with respect to the turbomotor axis; however, when the turbine blade tangential velocity V_T for the turbomotor running at its design speed of e.g. 35,000 RPM is vectorially summed with the leaving velocity V_L , the resultant velocity vector V_s corresponds in both magnitude and direction to the vector V_N .

These relationships are also illustrated in the stall torque ratio graph of FIG. 12. Stall torque ratio is defined as the ratio of the torque at which the turbomotor will stall to the design torque of the turbomotor. FIG. 12 has percentage of design torque as the ordinate and percentage of design speed as the abscissa. From the graph, it will be seen that the axial turbine contributes maximum torque at zero speed and ceases to contribute any torque at one hundred percent of design speed (i.e., at design speed). The radial turbine also contributes maximum torque at zero speed but is still contributing substantial torque at design speed, at which point it is contributing all of the turbomotor torque since the axial wheel is now windmilling. The described axial-radial combination produces a turbomotor having a higher stall torque ratio than either the radial turbine alone or the axial turbine alone, and yet has the ability to operate with moderate efficiency in reverse. If the axial turbine were arranged to produce significant torque at design speed, it would seriously hamper or even eliminate operation of the turbomotor in reverse. Thus, as represented on the graph by the dash line curve labelled AT1, if the axial turbine were designed to have a higher torque curve than the invention curve, e.g. a torque curve representing sixty percent of the torque curve for the radial unit, the axial turbine would contribute to turbomotor torque at design speed but would eliminate the reverse operational mode since, assuming sixty percent efficiency for the radial wheel in reverse, the radial wheel reverse torque would be

cancelled at all speeds by the drag of the axial turbine. Similarly, as represented on the graph by the dash line curve labelled AT2, if the axial turbine were designed to have a lower torque curve than the invention curve, the axial turbine would have less of a drag effect in the reverse mode but would also generate a considerable drag at design speed in the forward mode. The described arrangement, wherein the axial stage is designed to windmill at design speed, is thus seen to present a happy marriage of the radial and axial turbine characteristics and is seen to be an ideal turbomotor for use in applications, such as the disclosed hoist application, where good torque is required in the "forward" (pay-in) mode but yet an operational and usable "reverse" (pay-out) mode is also required.

MODIFIED NOZZLE CONSTRUCTION

A modification of the FIG. 4-7 radial turbine wheel nozzle construction is shown in FIGS. 13 and 14 in which parts which are in common with or similar to the FIG. 4-7 construction are given corresponding reference numerals but with a prime added. In the FIG. 13 and 14 modification, the blocking ring 78 is replaced with a structure whereby the blades 82' may be selectively pivoted about axes generally parallel to the turbomotor axis. The nozzle assembly 76' of FIGS. 13 and 14, in addition to support rings 79' and 80' and bridging blades 84', includes a plurality of strut members 168 integral with support ring 79' and extending therefrom in cantilever fashion to overlie each blade 82', a first ring member 170 secured to assembly 76' by screw bolts 172 received in threaded bores in strut members 168, and a second ring member 174 secured to first ring member 170 by screw bolts 176. An externally toothed ring gear 178 is journaled in an annular cavity 180 in ring member 174 by a ball race 182. Ring gear 178 engages a plurality of splined shafts 184; each shaft 184 terminates at one end in a pin portion 186 journaled in support ring 79'. Each blade 82' is keyed to a respective pin portion 186 so that the blade is rotated by rotation of the corresponding shaft 184. The illustrated shaft 184 extends leftwardly for splined receipt in an internally splined sleeve 188 positioned in the adjacent wall portion of housing 26'; the remaining shafts 184 are stub shafts which are journaled at their left ends in the left or inner wall portion 190 of ring member 174 with their splines engaged with the teeth of ring gear 178. A handle 192 is secured to the left end of sleeve 188. When handle 192 is turned, the illustrated blade 82' is rotated directly; ring gear 178 is also rotated by rotation of the illustrated shaft 184 and ring gear 178 in turn rotates stub shafts 184 so that all blades 82' are rotated in unison. Nozzle assembly 76' is shown in the forward or pay-in mode in FIG. 13 in which incoming working fluid is directed into radial wheel 48' by blades 82'. The angle and effective working area of blades 82' (see FIG. 14) may be selectively modulated by selective rotation of handle 192. Reversal of nozzle assembly 76' is accomplished as in the embodiment of FIG. 4-7 by admission of pressurized control fluid into annular subchamber 88' through bolt passage 90' and cross bore 92'. Leftward movement of the nozzle assembly to bring blades 84' into working position is allowed and guided by sliding movement of the illustrated spline shaft 184 in the splined bore of sleeve 188. With blades 84' in working position, the working fluid is directed into radial wheel 48' at an

angle (see FIG. 14) to cause rotation of wheel 48' in a reverse or pay-out direction. Nozzle 76' may be returned to the forward mode position of FIG. 13 by release of control pressure in subchamber 88' and admission of pressurized control fluid into subchamber 100' 5 through a suitable port (not shown) in housing 26'.

While the invention has been illustrated and described in detail with reference to the disclosed embodiments, it will be understood that various changes and modifications may be made in the disclosed embodiments without departing from the scope or spirit of the invention. 10

I claim:

1. A fluid turbomotor comprising:

- A. means defining a housing; 15
- B. a radial turbine wheel of the radial inflow type;
- C. means mounting said turbine wheel for rotation within said housing about its central axis;
- D. a nozzle ring structure positioned in said housing concentric with said axis and encircling said turbine wheel, said nozzle ring structure including 20

1. a support ring,

2. a first blade set comprising a plurality of blades circumferentially spaced about said axis and disposed at a first angle with respect to the orbit of said wheel, the blades of said first blade set are integral at one end with said support ring and project therefrom in cantilever fashion to present a free edge, and 25

3. a second blade set spaced axially from said first blade set and comprising a plurality of blades circumferentially spaced about said axis and disposed at a second angle with respect to said orbit; 30

E. means mounting said ring structure in said housing for axial movement relative to said wheel between 35

1. a first position axially aligning said first blade set with the outer periphery of said wheel and axially displacing said second blade set with respect to said outer periphery, whereby working fluid may be directed through said first blade set against the vanes of said turbine wheel at said first angle, and 40

2. a second position axially aligning said second blade set with said outer periphery and axially displacing said first blade set with respect to said outer periphery, whereby fluid may be directed through said second blade set against the vanes of said turbine wheel at said second angle; 45

F. a ring member disposed within said housing in side-by-side relation to said first blade set and presenting an annular face in confronting relation to said free edges, said ring member having a plurality of axially extending pockets formed therein each of a shape to snugly receive a respective blade of said first blade set and each opening in said confronting annular face; and 50

G. means mounting said ring member for axial movement within said housing relative to said nozzle ring structure to selectively receive the blades of said first blade set within said pockets and thereby selectively vary the effective, exposed working area of the blades of said first blade set. 55

2. A fluid turbomotor according to claim 1 wherein

H. said mounting means comprises means defining an annular chamber within said housing concentric with said axis and encircling said radial turbine wheel; 60

I. said nozzle ring structure comprises an annular first piston slideably received in said chamber for axial reciprocation to selectively move said nozzle ring structure between said first and second positions; and

J. said ring member comprises an annular second piston slideably received in said chamber in tandem relation to said first piston and arranged for axial reciprocation relative to said first piston to selectively vary the effective working area of said first set blades.

3. A fluid turbomotor according to claim 2 and further including

K. first sealing means defining a first sealed subchamber within said chamber between an annular end wall of said chamber and the adjacent annular end wall of said first piston; 15

L. first conduit means opening in said first subchamber for introducing pressurized control fluid into said first subchamber to reciprocate said first piston in said chamber between said first and second positions;

M. second sealing means defining a second sealed subchamber within said chamber between the other annular end of said chamber and the adjacent annular end wall of said second piston; and

N. second conduit means opening in said second subchamber for introducing pressurized control fluid into said second subchamber to reciprocate said second piston in said chamber relative to said first piston to selectively vary the effective working area of said first set blades.

4. A fluid turbomotor comprising

A. means defining a housing;

B. a radial turbine wheel of the radial inflow type;

C. means mounting said turbine wheel for rotation within said housing about its central axis;

D. a nozzle ring structure positioned in said housing concentric with said axis and encircling said turbine wheel, said nozzle ring structure including

1. a support ring portion, and

2. a plurality of blades circumferentially spaced about said axis and integral at one end with said support ring and projecting axially therefrom in cantilever fashion to present a free edge; 40

E. a ring member disposed within said housing in side-by-side relation to said blades and presenting an annular face in confronting relation to said free blade edges, 45

said ring member having a plurality of axially extending pockets formed therein each of a shape to snugly receive a respective blade and each opening in said confronting annular face; and

F. means mounting said ring member for axial movement within said housing relative to said nozzle ring structure to selectively receive said blades within said pockets and thereby selectively vary the effective, exposed working area of said blades. 50

5. A fluid turbomotor according to claim 4 wherein

G. said mounting means comprises means defining an annular chamber within said housing concentric with said axis and encircling said radial turbine wheel; and

H. said ring member comprises an annular piston slideably received in said chamber for axial reciprocation in said chamber to selectively vary the effective working area of said first set blades. 55

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6. A fluid turbomotor comprising:
- A. means defining a housing;
 - B. a radial turbine wheel of the radial inflow type;
 - C. means mounting said turbine wheel for rotation within said housing about its central axis;
 - D. a nozzle ring structure positioned in said housing concentric with said axis and encircling said turbine wheel, said nozzle ring structure including
 - 1. a first blade set comprising a plurality of blades circumferentially spaced about said axis and each disposed at an angle variable with respect to the orbit of said wheel,
 - 2. means mounting the blades of said first blade set for pivotal movement relative to said ring structure about axes parallel to said axis,
 - 3. means for pivoting the blades of said first blade set in unison about said axes, and
 - 4. a second blade set spaced axially from said first set and comprising a plurality of blades circum-

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- ferentially spaced about said axis and each disposed at a fixed angle with respect to said orbit;
- E. means mounting said ring structure in said housing for axial movement relative to said wheel between
 - 1. a first position axially aligning said first blade set with the outer periphery of said wheel and axially displacing said second with respect to said outer periphery, whereby working fluid may be directed through said first blade set against the vanes of said turbine wheel at said variable angle, and
 - 2. a second position axially aligning said second blade set with said outer periphery of said wheel and axially displacing said first blade set with respect to said outer periphery, whereby fluid may be directed through said second blade set against the vanes of said turbine wheel at said fixed angle.

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