June 18, 1963 V. S. WAGNER 3,094,041

SERVOMECHANISM

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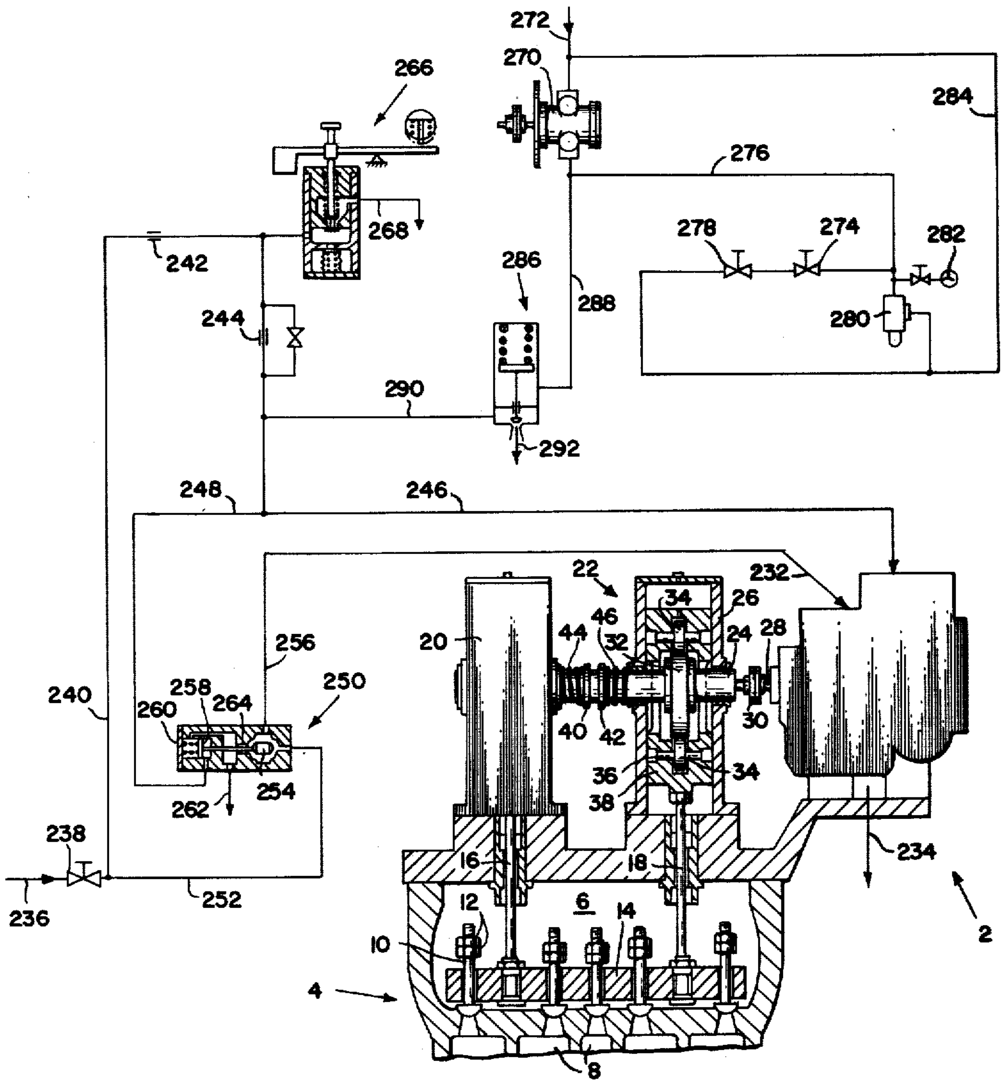

FIG. I.

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SERVOMECHANISM

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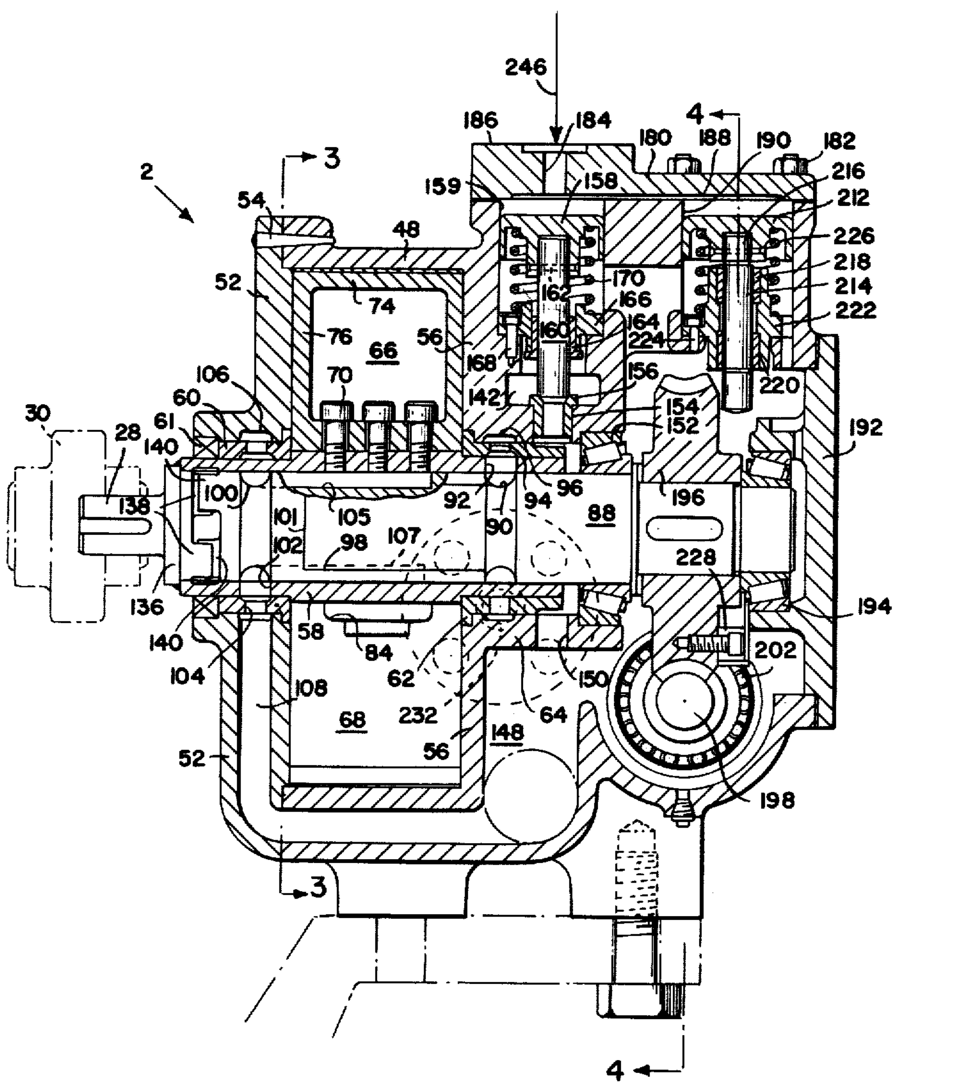

FIG. 2.

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SERVOMECHANISM

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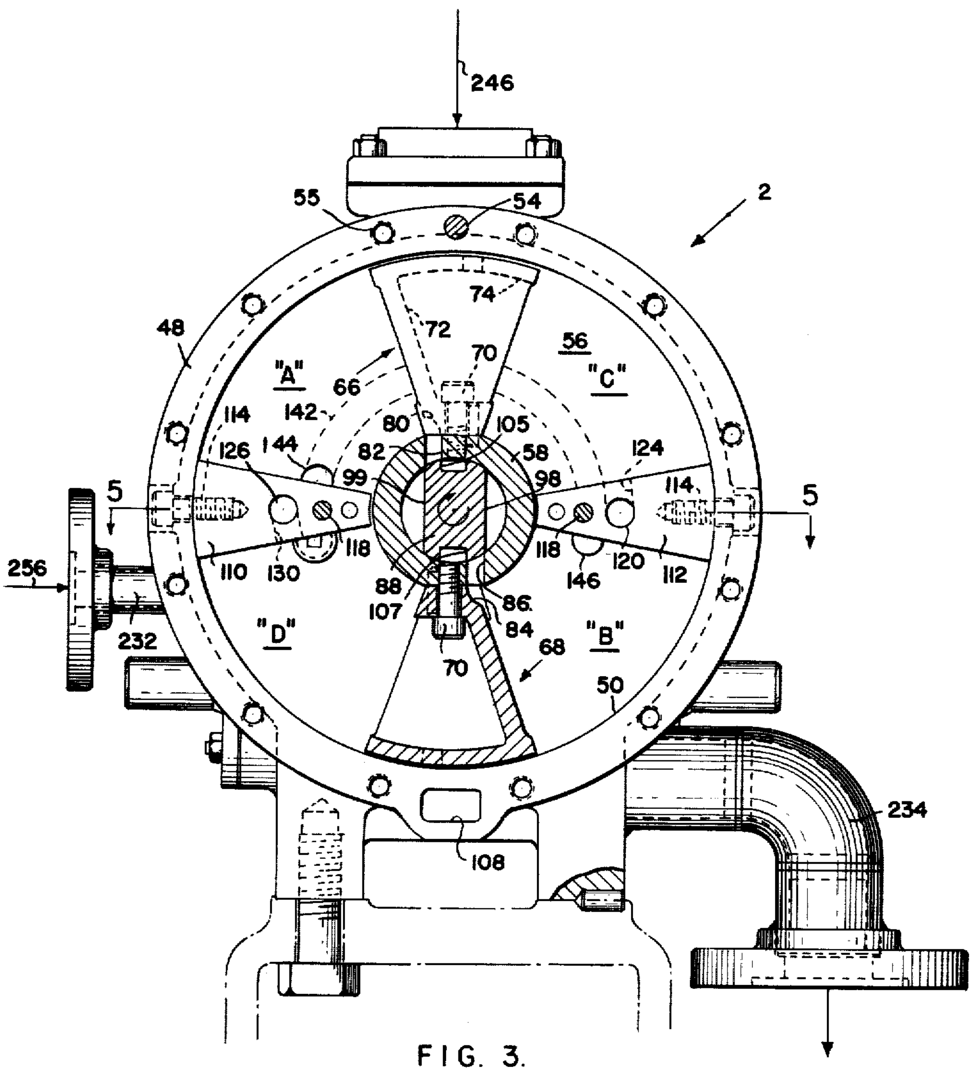

FIG. 3.

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SERVOMECHANISM

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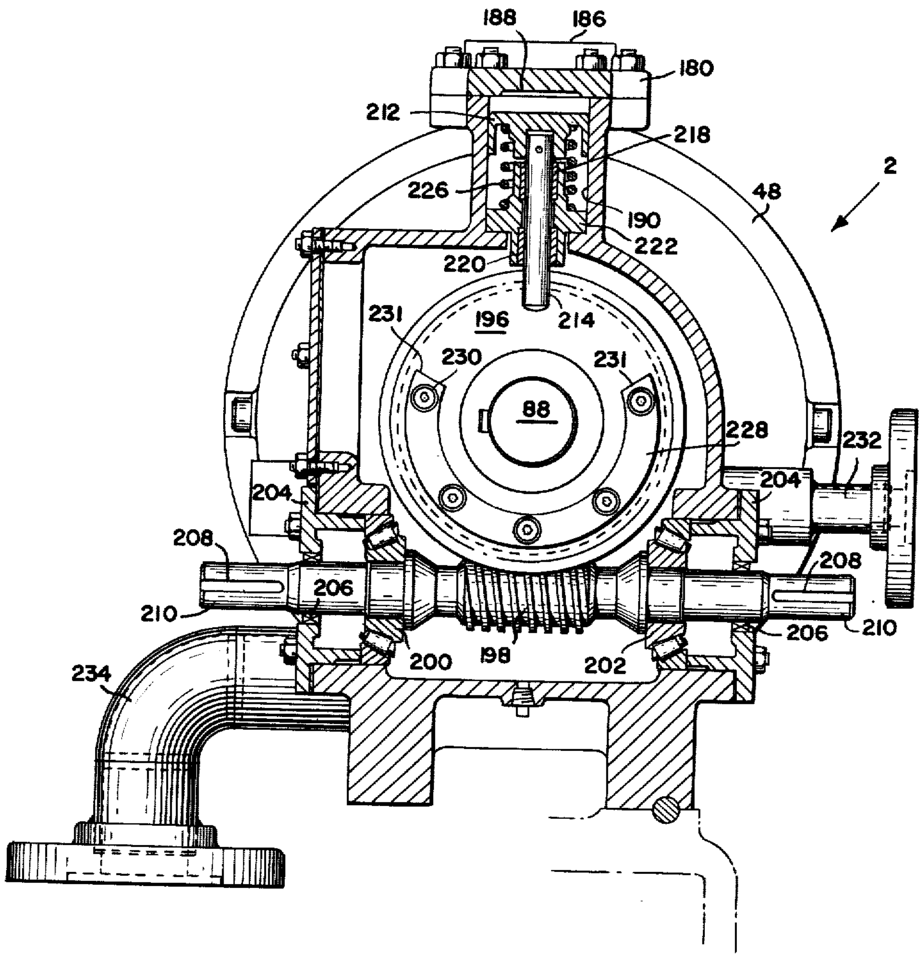

FIG. 4.

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SERVOMECHANISM

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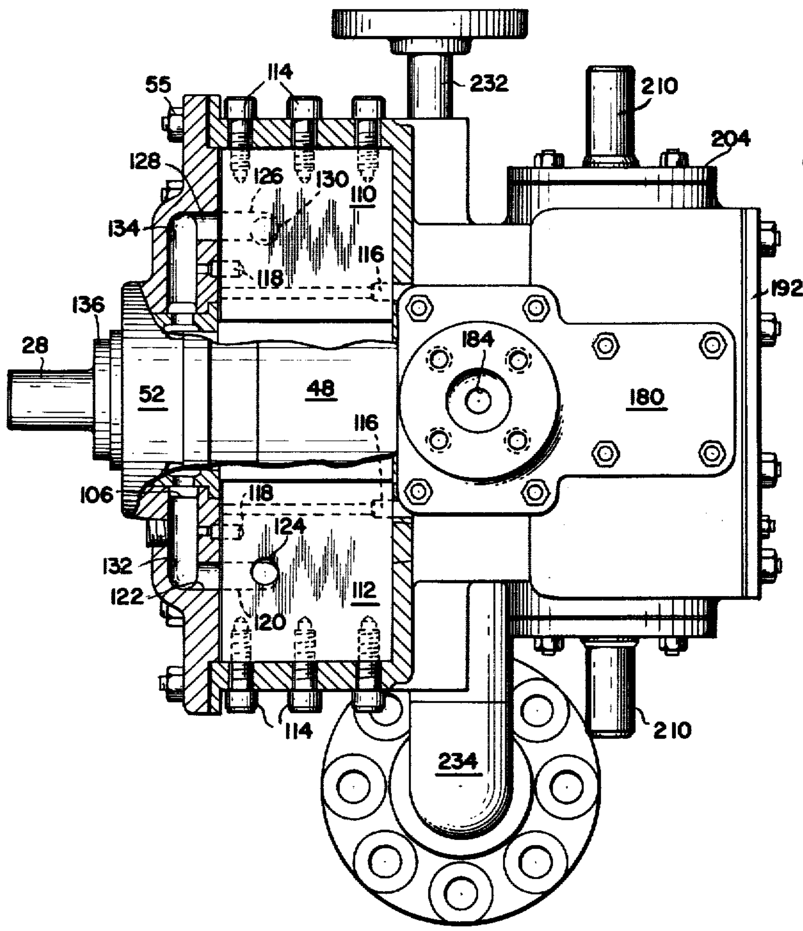

FIG. 5.

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Filed Oct. 7, 1960 6 Sheets-Sheet 6

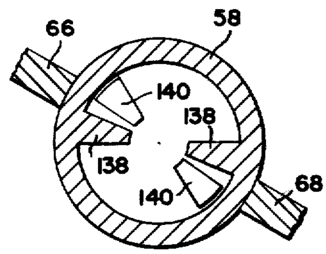

FIG. 6A.

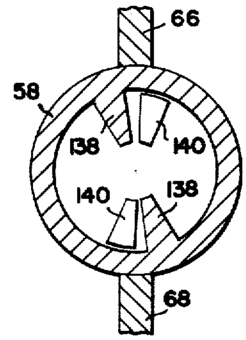

FIG. 6B.

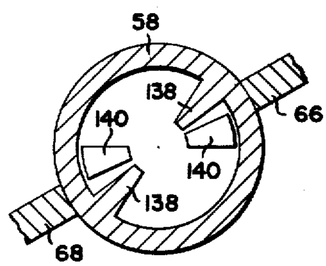

FIG. 6C.

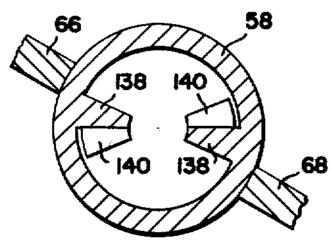

FIG. 6D.

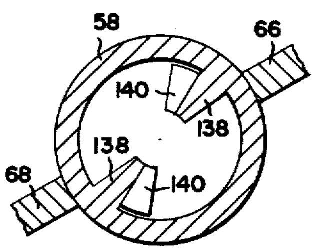

FIG. 6E.

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3,094,041
SERVOMECHANISM

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Filed Oct. 7, 1960, Ser. No. 61,128
8 Claims. (Cl. 91—375)

This invention relates to a servomechanism for operating, for example, the valves of a steam turbine, and more particularly to such a mechanism which automatically closes the valves under certain conditions while nevertheless permitting manual operation thereof when necessary.

While the invention described hereafter is applicable to many situations wherein servomechanisms are employed to operate various devices, it will be described particularly in relation to the operation of the large steam valves generally employed in marine steam turbine propulsion plants. These valves regulate the admission of steam to the turbine nozzles, and thereby control the power output of the turbine. Usually, the turbine has its own oil system, and it obviously is imperative that when the oil pressure drops due to a failure of the system the turbine should be shut down, which is effected by closing of the steam valves. Also, upon an abnormal increase in the turbine speed, the steam valves are closed to the extent necessary to bring the turbine back to its proper speed or to shut the turbine down if it were to be in a "runaway" condition. Such controlling operations preferably are effected automatically. Since the valves are large, direct manual operation thereof is difficult and some form of power operation is desirable. Such power operation may be hydraulical, as specifically disclosed hereafter, using oil from the ship's oil system, but in the event of a failure in such system and consequent deactivation of the hydraulic power operator, it should be possible nevertheless to manually operate the valves if necessary.

Accordingly, it is an object of the invention to provide a servomechanism with a power operator for the manual operation of steam valves or the like.

It is a further object to provide such mechanism which is responsive to certain conditions such as a failure in an oil system and/or an unsafe speed to close or adjust the valves automatically.

It is a further object to provide such mechanism which permits manual operation of valves or the like despite inoperability of the power operator, for example due to a failure in its operating oil system.

Further objects and advantages will become apparent from the following disclosure read in conjunction with the drawings, in which:

FIGURE 1 is a flow diagram of a typical hydraulic circuit and a schematic illustration of the servomechanism operably connected to the turbine steam valves;

FIGURE 2 is a vertical axial cross-section of the servomechanism;

FIGURE 3 is a transverse cross-section taken on line 3—3 of FIGURE 2;

FIGURE 4 is a cross-section taken on line 4—4 of FIGURE 2;

FIGURE 5 is a plan view of the servomechanism, with a portion thereof shown as a cross-section taken on line 5—5 of FIGURE 3; and

FIGURES 6A to 6E inclusive schematically illustrate the relative positions of the pilot valve and vane during various stages of operation.

Referring to FIGURE 1, the servomechanism for operating the turbine steam valves is indicated at **2**, and hereafter for purposes of description this mechanism will be referred to briefly as the "operator." A bank of valves **4** is provided within the steam chest **6** to individually regulate the admission of steam to chests as indicated at

2

8 associated respectively with groups of the turbine nozzles. Each of the valve members **10** has a threaded stem on which are locked a pair of nuts **12** adapted for engagement by a valve lifting bar **14**. The bar **14** is adapted to be raised and lowered by a pair of rods **16** and **18** operated, respectively, by cam assemblies **20** and **22**. It will be noted that the nuts **12** of the respective valves are positioned at various heights with respect to their valve stems. The purpose of this arrangement is that it is desired not to open all of the valves simultaneously, but to open the valves in sequence, one at a time, and as the bar **14** is lifted it will engage the fixed nuts of certain valves to lift and open them prior to the opening of other valves. The cam assembly **22**, which is identical with cam assembly **20**, is shown in cross-section to illustrate the operating parts thereof. A shaft **24** (which is common to assemblies **20** and **22**) is journalled in the housing **26** of assembly **22** and connected to an output shaft **28** of the operator **2** by a coupling **30**. An eccentric cam **32** is keyed to shaft **24**, and a cam follower mechanism therefor comprises a plurality of rollers **34** in circumferentially spaced engagement with the cam **32**. Rollers **34** are rotatably mounted by pins **36** in a cage **38** connected to the rod **18**. Thus it will be evident that rotation of cam **32** effects reciprocation of rod **18** to raise and lower bar **14**. It will be understood that the above described structure of cam assembly **22** is identical with that of assembly **20**, and the two assemblies are jointly acting on the bar **14**. A pair of collars **40** and **42** are integral with shaft **24**, and between these collars and the respective adjacent housings of assemblies **20** and **22** a pair of torsion coil springs **44** and **46** are concentrically mounted on the shaft. Spring **44** is a right hand helix and spring **46** is left hand helix, and the effect of this arrangement is that rotation of shaft **24** in a clockwise direction, as viewed from the left end in FIGURE 1, is resisted by the coiling of springs **44** and **46**.

Passing now to description of the specific operator structure **2**, reference will be made to FIGURES 2 to 5. The main body of the operator housing, generally indicated at **48**, may consist of a single steel casting. The left end (FIGURE 2) of housing **48** has the form of a drum or cylinder **50** which is closed at one end by a head **52** secured to the housing by a positioning taper pin **54** and bolts **55**, and is closed at its other end by a cast-in wall **56**. A sleeve **58** passes axially through cylinder **50** and is rotatably mounted at its left end (FIGURE 2) by a bushing **60** and oil seal **61** in head **52**, and at its right end by a bushing **62** mounted in an annular flange **64** extending from wall **56**. A pair of vanes **66** and **68** extend radially from sleeve **58** and are mounted thereon by bolts **70**. The vanes **66** and **68** are identical, the vane **66**, for example, being substantially cup-shaped and comprising a back wall **72**, an arcuate end **74** closely fitting the cylinder **50**, and side walls **76** and **78** closely fitting, respectively, the head **52** and wall **56**. As will be seen hereafter the vanes **66** and **68** are, in effect, similar to pistons movable within cylinder **50**. Vanes **66** and **68** are milled away or cast at **80** and **84** to expose, respectively, ports **82** and **86** in the wall of sleeve **58**.

A member **88**, hereafter referred to as the "pilot valve," is closely fitted and rotatably received within sleeve **58**. Pilot valve **88** has a circumferential groove **90** which is axially aligned with ports **92** and **94** in the sleeve **58** and bushing **62**, respectively, and with a continuous circular channel **96** in wall **56**. Pilot valve **88** has a second circumferential groove **100** axially aligned with ports **102** and **104** in sleeve **58** and bushing **60**, respectively, and with a channel **106** in head **52** extending around bushing **60** and communicating with a passage **108**.

Pilot valve **88** has flat portions or lands **98** and **99**

(FIGURE 3) extending axially from groove **90** to the extent indicated at **101**. Extending rightwardly in (FIGURE 2) from groove **100** pilot valve **88** has a pair of channels **105** and **107** which terminate short of the groove **90**.

A pair of opposed tapered abutments **110** and **112** extend horizontally from the cylinder **50** (FIGURE 3), from which they are supported by bolts **114**. Pins **116** and **118** anchor the abutments to the wall **56** and head **52**, respectively. A passage **120** in abutment **112** is aligned with an opening **122** in head **52**, and terminates in the upwardly opening passage **124**.

Vane **110** has a similar passage **126** which is aligned with an opening **128** in head **52** and terminates in the downwardly opening passage **130** (FIGURE 3). Passages **132** and **134** (FIGURE 5) provide communication between openings **122** and **128**, respectively, and channel **106**, which in turn communicates with passage **108** as previously mentioned.

The exposed end of sleeve **58** is welded to a flange **136** on the operator output shaft **28**, and a jaw clutch or coupling between shaft **28** and pilot valve **88** is constituted by lugs **138** on the former and lugs **140** on the latter. (This coupling will be described in greater particularity hereafter with reference to FIGURES 6A to 6E.)

A U-shaped chamber **142** is cast into the housing in the general area of wall **56**. An opening **144** located just above the abutment **110** (FIGURE 3) passes through wall **56** and communicates with chamber **142**. Similarly, an opening **146** located just below abutment **112** passes through wall **56** and communicates with chamber **142**. The space **148** is in effect a sump which, as described hereafter, is connected to an internal drain of the hydraulic system. A hole **150** (FIGURE 2) provides communication between sump **148** and the gap between bushing **62** and a roller bearing assembly **152** rotatably mounting pilot valve **88**. Through a bore **154**, having a valve seat **156** therein, communication thereby may be established between chamber **142** and sump **148**. A piston or plunger **158** is reciprocable within a cylindrical bore **159** cast in the housing, and is connected to a valve stem **160** by pin **162**. Stem **160** is reciprocable in a bearing **164** mounted in a ring **166** that is retained in place by a cap screw **168**. A compression coil spring **170** bottomed against ring **166** urges plunger **158** upwardly. It will be noted that the lower end of stem **160** is adapted to mate with the valve seat **156** to thereby prevent the interchange of a liquid between chamber **142** and sump **148**.

The open right upper portion of housing **48** (FIGURE 2) is closed by a head **180** secured by bolts **182**. A line connection is indicated at **184**, the head **180** having an annular flange **186** for connection as a flanged line coupling. A channel **188** in head **180** provides communication between line connection **184** and a further cylindrical bore **190** in the housing.

The open right end of housing **48** (FIGURE 2) is closed by a plate **192** mounting a further roller bearing assembly **194** for supporting pilot valve **88**. A worm gear **196** is keyed on the end shaft portion of pilot valve **88**, and a meshing worm **198** is rotatably supported in roller bearing assemblies **200** and **202** (FIGURE 4). The bearing assemblies are retained by cover plates **204** and oil seals **206**. The end shaft portion **210** of the worm **198** has splines **208** for accommodating a manually operable crank, handwheel or the like.

A piston or plunger **212** is reciprocable in the cylindrical bore **190** and is connected to a rod **214** by a pin **216**. Rod **214** is reciprocable in bearings **218** and **220** mounted in a ring **222** which is retained in place by a cap screw **224**. The plunger **212** is urged upwardly by a compression coil spring **226** bottomed against ring **222**. An arcuate segment **228** secured to the worm gear **196** by cap screws **230** has ends **231** which comprise abutments directly engageable by the rod **214** in the position shown upon a predetermined rotation of the gear. However, it will be evident that if rod **214** moves to the limit of its upward movement, rotation of worm gear **196** will be unlimited.

A fitting **232** adapted to be flange coupled to a hydraulic line is tapped into the housing **48** and communicates with the channel **96**, ports **94** and **92**, groove **90** and flats **98** and **99**. A further fitting **234** is tapped into the housing **48** in communication with sump **148** and is adapted to be flange coupled to a hydraulic line.

Referring now back to FIGURE 1, the hydraulic system which operates the servomechanism or operator **2** will now be described, it being understood that the hydraulic circuit elements per se are conventional and are described merely as an example of one application of the invention. Oil enters the system through line **236**, which is connected to the high pressure lubricating oil system of the steam turbine. After a hand valve **238** the line **236** branches off into a line **240**. After two flow restricting orifices **242** and **244**, line **240** branches off into a first line **246** leading to the line connection **184** (previously described with reference to FIGURE 2), and a second line **248** leading to a valve **250**. A second branch **252** of line **236** leads to valve **250**, and when a spool **254** of this valve is in its extreme left-hand position (FIGURE 1) the flow of oil is permitted from branch **252** to a line **256** which is connected to the previously mentioned fitting **232**. A piston **258** operatively connected to spool **254** normally is urged to the left against the force of a spring **260** by oil pressure in the line **248**, but when, due to a drop in said pressure, spool **254** is caused to move to its right-hand position (FIGURE 1) communication is established between line **256** and a drain connection **262** via a clearance **264**. It will be evident that the pressure of oil in line **246** normally will act on pistons **158** and **212** (FIGURE 2) to force them downwardly against the opposition of springs **170** and **226**, respectively. However, upon a failure of such pressure in line **246** piston **158** will be allowed to move upwardly under the action of spring **170**, lifting valve stem **160** from seat **156**. Likewise, plunger **212** will be allowed to move upwardly against the action of spring **226**, lifting rod **214** out of the path of the arcuate abutment **228**.

An overspeed tripping device is indicated generally at **266**. Devices of this type are conventional, a detailed description therefore being unnecessary, and one may comprise valve means for opening the line **240** to internal drain **268** when the speed of a turbine or the like exceeds a predetermined safe maximum speed. Device **266** is of the type which must be manually reset after it has been tripped to "dump" the line **240** to drain **268**.

A positive displacement pump **270** is driven directly from the shaft of the steam turbine, and draws oil from a line **272** connected to the lubricating oil system. Pump **270** discharges against an adjustable locked orifice **274**, the discharge pressure in line **276** having an approximate squared relationship with respect to the turbine speed, and against a manual test valve **278**. A pressure release valve **280** and a pressure gauge **282** also are provided. The discharge of pump **270** is recirculated back to the pump inlet through a line **284**. A conventional valve **286** is responsive to pressure in a line **288**, which is connected to the outlet of pump **270**. Upon the attainment of a predetermined pressure in line **288**, indicating an excessive speed of the turbine, valve **286** operates to connect a branch **290** of line **240** to a drain line **292**. It will be evident that upon the operation of valve **286** to open line **290** to drain **292**, a drop in oil pressure will occur in line **240** and the branches thereof **246** and **248**.

The operation of the aforedescribed mechanism will now be described, and in connection therewith supplemental reference will be made to FIGURES 6A to 6E, inclusive. These figures, which are intended to be schematic, represent the various relationships between the pilot valve and the vanes during different stages of operation and as viewed in FIGURE 3. Reference hereafter to clockwise and counterclockwise rotation of the pilot valve, vanes and other elements should be understood as when viewed according to FIGURE 3 and FIGURES 6A to 6E. When the steam valves 4 are closed, vanes 66 and 68 are in the position shown in FIGURES 6A and 6D; in other words, they are at the limit of counterclockwise rotation as viewed in FIGURE 3. Rotation of vanes 66 and 68 in a clockwise direction, i.e. toward their respective positions shown in FIGURE 6B, tends to open valves 4, it being understood that such clockwise rotation of the vanes and of the sleeve 58 is resisted by springs 44 and 46 acting through shaft 24. When vanes 66 and 68 are rotated to the limit of their clockwise rotation, as shown in FIGURES 6C and 6E, the valves 4 are completely open. As mentioned, such clockwise rotation of the vanes and of shaft 24 can be effected only by overcoming the resisting action of torsion springs 44 and 46. It will be noted that the jaw coupling or clutch comprising the lugs 138 rigid with sleeve 58 and the lugs 140 on pilot valve 88 also is illustrated in FIGURES 6A to 6E.

Assume now that the steam valves 4 are closed, the vanes and pilot valve being in their respective positions indicated by FIGURE 6A, and that before it is desired to open these valves the oil system of the turbine is operating properly and high pressure oil is standing in line 236 (FIGURE 1). With valve 238 open, oil under pressure flows through lines 240 and 246 to the plungers 158 and 216 (FIGURE 2); oil also flows through line 248 to valve 250, effecting the movement of spool 254 to the left and permitting the flow of oil through lines 252 and 256 to the connection 232. The valve 286 constitutes a primary overspeed control and device 266 constitutes a secondary overspeed control, but since the turbine shaft is not rotating when the valves 4 are closed, neither of the drains 268 and 292 will be open. By virtue of the high pressure oil acting thereon, plungers 158 and 216 will be in their lowermost positions as shown in FIGURE 2.

Referring to FIGURES 6A, 6B and 6C (FIGURE 6B corresponds to FIGURE 3) when the lugs 140 are in the positions shown therein with respect to lugs 138, the channels 105, 107 are in the positions shown in FIGURE 3 with respect to the ports 82, 86.

For any setting of the steam valves 4, during normal hydraulic operation of the servomechanism, the positions of lugs 140 with respect to lugs 138 are as shown in FIGURES 6A, 6B and 6C, i.e. the lugs 140 are positioned slightly clockwise from or ahead of lugs 138. Now, if pilot valve 88 is rotated clockwise by a manual cranking of worm 198, flat 99 will uncover port 82, permitting oil under pressure to flow from the channel 96, ports 92 and 94 and groove 90 past the flat 99 and into a chamber designated "A." Similarly, flat 98 will uncover port 86, permitting high pressure oil to flow past flat 98 and through port 86 into a chamber designated "B." The liquid pressure in chambers A and B, acting against vanes 66 and 68 in a clockwise direction, causes the vanes to "follow" the pilot valve 88 in a clockwise direction. The sleeve 58 and shaft 24 are driven in a clockwise direction by such movement of the vanes, which movement is opposed by the torsion springs 44 and 46. Such clockwise rotation of shaft 24, of course, results in opening of the valves 4 by the cam assemblies 20 and 22. The driving of vanes 66 and 68 continues until they have "caught up with" the pilot valve 88, that is, until the ports 82 and 86 no longer are uncovered by the flats 99 and 98, respectively. As an example, assume that pilot valve 88 was rotated to a position shown in FIGURE 3. The vanes 66 and 68 would be driven clockwise until they reach the condition shown in FIGURE 3 and FIGURE 6B, and in which the ports 82 and 86 are covered by the pilot valve 88.

Two other chambers, "C" and "D," are identified in FIGURE 3, and these chambers are connected, via passages 120, 124 and 126, 130 to the passage 108 and sump 148, the sump being connected to an internal drain in the hydraulic system by connection 234. Thus, in the course of clockwise rotation of vanes 66 and 68 oil may be displaced from chambers C and D to the drain.

It should be noted that the opening of the steam valves is in no way dependent upon the driving of lugs 138 by lugs 140, but rather that the lugs 140 stay ahead of (clockwise of) the lugs 138. The driving of sleeve 58 and shaft 24 to open the steam valves during normal hydraulic operation is effected solely by oil pressure in chambers A and B acting on vanes 66 and 68 which act in much the same manner as pistons. It will be evident that the extent of rotation of vanes 66 and 68 and sleeve 58 is directly proportionate to the amount of rotation of pilot valve 88 initially effected by a manual cranking of worm 198. Thus, if by manual cranking of worm 198 the pilot valve 88 is rotated clockwise to an extent where its lugs 140 are positioned as shown in FIGURE 6C, vanes 66 and 68 and sleeve 58 will be driven to their extreme clockwise position wherein the ports 80 and 86 again will be closed and the steam valves 4 will be opened.

In a balanced, stable condition of the servomechanism as, for example, in FIGURE 3, the torsion springs 44 and 46 continuously exert a counterclockwise force on sleeve 58, but this force is equally opposed by oil pressure in chambers A and B. Ordinarily, due to the leakage of oil from chambers A and B springs 44 and 46 would be able gradually to move vanes 66 and 68 in a counterclockwise direction. However, such counterclockwise movement would move ports 80 and 86 relative to the flats 98 and 99 of the pilot valve 88, and these ports would be uncovered to admit more high pressure oil to chambers A and B to maintain the stabilized condition. Consequently, in a stabilized condition the ports 82 and 86 may not be completely covered, but are open by perhaps a very small amount to permit a small flow of oil to chambers A and B. Compensation is thus made for the aforementioned leakage of oil.

With the steam valve 4 now fully opened and the servomechanism in the condition shown in FIGURE 6C, the steam valve may be closed in one of three ways: (*a*) by manual operation of worm 198, (*b*) by operation of either the primary or secondary overspeed tripping means, or (*c*) upon failure of the turbine oil system.

The steam valves may be closed by rotation of worm 198 in a direction to move pilot valve 88 counterclockwise (FIGURE 3). Such counterclockwise rotation causes channels 105 and 107 to uncover ports 82 and 86, respectively. Since channels 105 and 107 communicate with the sump 148 via groove 100, ports 102 and 104, and passage 108, chambers A and B will be connected to the drain. The drop in oil pressure in chambers A and B will permit springs 44 and 46 to rotate shaft 24, sleeve 58 and vanes 66 and 68 in a counterclockwise direction, thereby also effecting the closing of steam valves 4. Again, the steam valves will be closed only to an extent proportionate to the amount of rotation of pilot valve 88, and if the pilot valve is rotated counterclockwise from the FIGURE 6C to the FIGURE 6B or FIGURE 3 position the vanes 66 and 68 will follow counterclockwise until the ports 82 and 86 are again covered. If the vanes 66 and 68 continue counterclockwise past the position wherein the ports 82 and 86 are closed, these ports will be uncovered by the flats 98 and 99 to admit high pressure oil to chambers A and B, with the consequences previously described. Complete closing of the steam valve 4, of course, is effected by movement of pilot valve 88 to the limit of its counterclockwise rotation.

At this point it should be noted that under normal operating conditions, when high pressure oil is acting downwardly on plunger 212, rod 214 is disposed in the path of abutment 228 and thereby limits the rotation of worm gear 196 and pilot valve 88. Thus, when worm gear 196 is rotated counterclockwise to bring the abutment 228 into engagement with rod 214 the pilot valve

88 will effect the full opening of steam valves 4, and clockwise rotation of worm gear 196 to bring the other end of abutment 228 into engagement with rod 214 will effect full closing of the steam valves. During normal hydraulic operation, by virtue of the rotation-limiting rod 214 the limit of counterclockwise rotation of lugs 140 is shown in FIGURE 6A and the limit of clockwise rotation is shown in FIGURE 6C. Springs 44 and 46 effect the urging of shaft 24 and lugs 138 in a counterclockwise direction (FIGURES 6A, 6B and FIGURE 6C) to close the steam valves, but lugs 140 are positioned ahead of, rather than behind, lugs 138 in a clockwise direction. It will be evident, therefore, that ordinarily manual rotation of pilot valve 88 cannot be effective to drive shaft 24 through lugs 138 and 140 to open the steam valves in opposition to springs 44 and 46. In other words, lowered rod 214 prevents the rotation of lugs 140 to the positions thereof shown in FIGURE 6D, where lugs 140 are behind lugs 138 and further rotation would drive sleeve 58 to open the steam valve. This latter aspect, however, will be discussed more fully hereafter.

Closing of the steam valves also may be effected automatically upon the attainment of an excessive turbine speed. The valve 286 constitutes the primary overspeed tripping means for closing the steam valve when the speed of the turbine reaches, say, 110% of the design full power speed. The device 266 constitutes a secondary overspeed tripping device for closing the steam valves when the speed of the turbine reaches, say, 115% of the design full power speed. Both the primary and secondary overspeed tripping means effect the draining of line 246, either through drain 292 or drain 268. As previously described, under normal hydraulic operation oil pressure in line 246 acts downwardly on plunger 158 to position the stem 160 against valve seat 156. Upon a failure of such pressure, however, spring 170 is permitted to lift valve stem 160 from seat 156. Assuming now that the steam valves 4 are open, with high pressure oil existing in chambers A and B, if the pressure in line 246 drops the stem 160 will be lifted to drain chambers A and B via openings 144 and 146, chamber 142, valve seat 156, hole 150, and sump 148. With the drainage of chambers A and B, high pressure oil no longer acts on vanes 66 and 68 in opposition to springs 44 and 46 which will, therefore, effect closing of the steam valves.

In the case of the primary overspeed control, when pump 270 is driven by the turbine at a predetermined high speed the back pressure exerted on the pump by orifices 274 will act through line 288 to open the valves 286, thereby connecting line 290 to drain 292. In the case of the secondary overspeed control device 266, when the turbine speed reaches, say, 115% of rated maximum speed, the line 246 will be connected to drain 268. In either case the effect is to lower the pressure in line 246 and cause the draining of chambers A and B through chamber 142. When the secondary overspeed tripping device 266 has been actuated, it must be manually reset to close the drain 268. However, when only the valve 286 has been opened, upon a resumption of normal turbine speed the drain 292 will be closed automatically. In the latter case, remembering that in the meantime the pilot valve 88 has not been changed from its original setting, the resumption of pressure in chambers A and B will effect a reopening of the steam valve 4 in a manner previously described.

Closing of the steam valves 4 automatically takes place if the turbine oil system fails or is operating unsatisfactorily. If the oil pressure in line 236 drops the pressure will drop also in the line 256 leading to connection 232. Line 256, it will be recalled, feeds chambers A and B via the groove 90 and flats 98, 99, etc. The oil pressure in lines 240, 246 and 248 also will drop. Therefore, first, spring 260 is permitted to move spool 254 to the right (FIGURE 1), connecting line 256 to drain 262

and, second, plungers 158 and valve stem 160 move upwardly to connect chambers A and B to drain line 234 via chamber 142 and sump 148. By the draining of chambers A and B torsion springs 44 and 46 are permitted to close steam valves 4.

In certain cases, even through the pressure in the turbine oil system is not sufficient to operate the servomechanism or operator 2, it may be desired nevertheless to open the steam valves 4. At such time, in the absence of substantial oil pressure in line 246, rod 214 will be raised out of the path of abutment 228 and will not limit the manual rotation of pilot valve 88. By the clockwise rotation of pilot valve 88 (by manual cranking of worm 198) lugs 140 will be brought to their positions indicated in FIGURE 6D. Upon further rotation of pilot valve 88 lugs 144 will drive lugs 138 in the manner of a jaw clutch, sleeve 58 and shaft 24 thereby being driven in the direction to open steam valves 4. This latter condition of the elements is illustrated in FIGURE 6E. Manual closing of the steam valves by springs 44 and 46 is permitted by the manual rotation of pilot valve 88 in a counterclockwise direction, i.e. from the FIGURE 6E to the FIGURE 6D position.

It will be understood that various departures from the above described embodiment of the invention may be made without departing from the scope thereof as defined by the following claims.

What is claimed is:

1. A servomechanism comprising a cylinder, a displaceable member within said cylinder, a source of hydraulic pressure for driving said member, an output member connected to said displaceable member for transmitting movement thereof, a pilot valve member having means for the hydraulic control of the operation of said displaceable member by said hydraulic pressure, said displaceable member constituting a follower driven in direct relation to the movement of said valve member, intermeshing lugs on said output member and valve member, the lugs on said output member normally being disengaged from and following the lugs on said valve member, means for operating said valve member, means normally preventing movement of said valve member to an extent that the lugs on the valve member engage and drive the lugs on the output member, and means operable to render inoperative the last-mentioned means, whereby the said output member may be driven directly by said valve member.

2. A servomechanism comprising a cylinder, a displaceable member within said cylinder, a source of hydraulic pressure for driving said member, an output member connected to said displaceable member for transmitting movement thereof, a pilot valve member having means for the hydraulic control of the operation of said displaceable member by said hydraulic pressure, said displaceable member constituting a follower driven in direct relation to the movement of said valve member, intermeshing lugs on said output member and valve member, the lugs on said output member normally being disengaged from and following the lugs on said valve member, means for operating said valve member, means responsive to and operable by said hydraulic pressure to prevent movement of said valve member to an extent that the lugs on the valve member engage and drive the lugs on the output member, the last-mentioned means being rendered inoperable upon a failure of said hydraulic pressure, whereby the said output member may be driven directly by said valve member.

3. A servomechanism comprising a cylinder having a displaceable member therein and defining therewith a chamber, a source of hydraulic pressure for driving said displaceable member, means providing a drain, a valve having a pair of elements, said elements being relatively movable in one direction to an open position wherein said chamber is connected to said source and in the opposite direction to an open position wherein said chamber is connected to said drain, the elements having a

closed position intermediate the aforementioned open positions, said displaceable member constituting and providing the mechanical output of the servomechanism and being connected to one of said elements to operate the same, and the other of said elements constituting mechanical input means for controlling the operation of the servomechanism, and means automatically responsive to to a predetermined drop in pressure at said hydraulic source to drain said chamber regardless of the position of said elements.

4. A servomechanism comprising a cylinder having a displaceable member therein and defining therewith a chamber, a source of hydraulic pressure for driving said displaceable member, means providing a drain, a valve having a pair of elements, said elements being relatively movable in one direction to an open position wherein said chamber is connected to said source and in the opposite direction to an open position wherein said chamber is connected to said drain, the elements having a closed position intermediate the aforementioned open positions, said displaceable member constituting and providing the mechanical output of the servomechanism and being connected to one of said elements to operate the same, and the other of said elements constituting mechanical input means for controlling the operation of the servomechanism, a clutch adapted to connect said other element to said displaceable member to drive the same, and means normally preventing engagement of said clutch but responsive to a predetermined drop in pressure at said source to permit such engagement, whereby said displaceable member is normally drive by hydraulic pressure but may be directly driven by operation of said other element upon a drop in said pressure.

5. A servomechanism comprising a cylinder, a vane within said cylinder, means defining with said vane and cylinder a chamber, a sleeve mounting said vane for displacement in a first direction by hydraulic pressure in the chamber, means resiliently urging said vane in a second and opposite direction in opposition to said pressure, motion transmitting means transmitting movement of said vane to perform mechanical work, a port in said sleeve for the ingress and egress of liquid in said chamber, a pilot

valve member within said sleeve and movable relative thereto, a source of hydraulic pressure, means providing a drain, said member normally occupying a position with respect to said sleeve to close said port, and said member being movable from said position in said first direction to open said port to said source and movable in said opposite direction to open said port to said drain, whereby said member constitutes the input means of the servomechanism, with movement of the member resulting in proportionate displacement of said vane.

6. Servomechanism according to claim 5, including means for transmitting motion of said pilot valve member to drive the first-mentioned motion transmitting means, and means responsive to the pressure at said source preventing operation of the second-mentioned motion transmitting means when said pressure exceeds a predetermined minimum.

7. Servomechanism according to claim 5, including a normally disengaged clutch connecting said pilot valve member and said motion transmitting means, and means responsive to a predetermined drop in the pressure at said source to engage said clutch to permit direct driving of said motion transmitting means by said pilot valve member.

8. Servomechanism according to claim 5, including means automatically responsive to a predetermined drop in the pressure at said source for draining said chamber regardless of the position of said pilot valve member relative to said port.

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