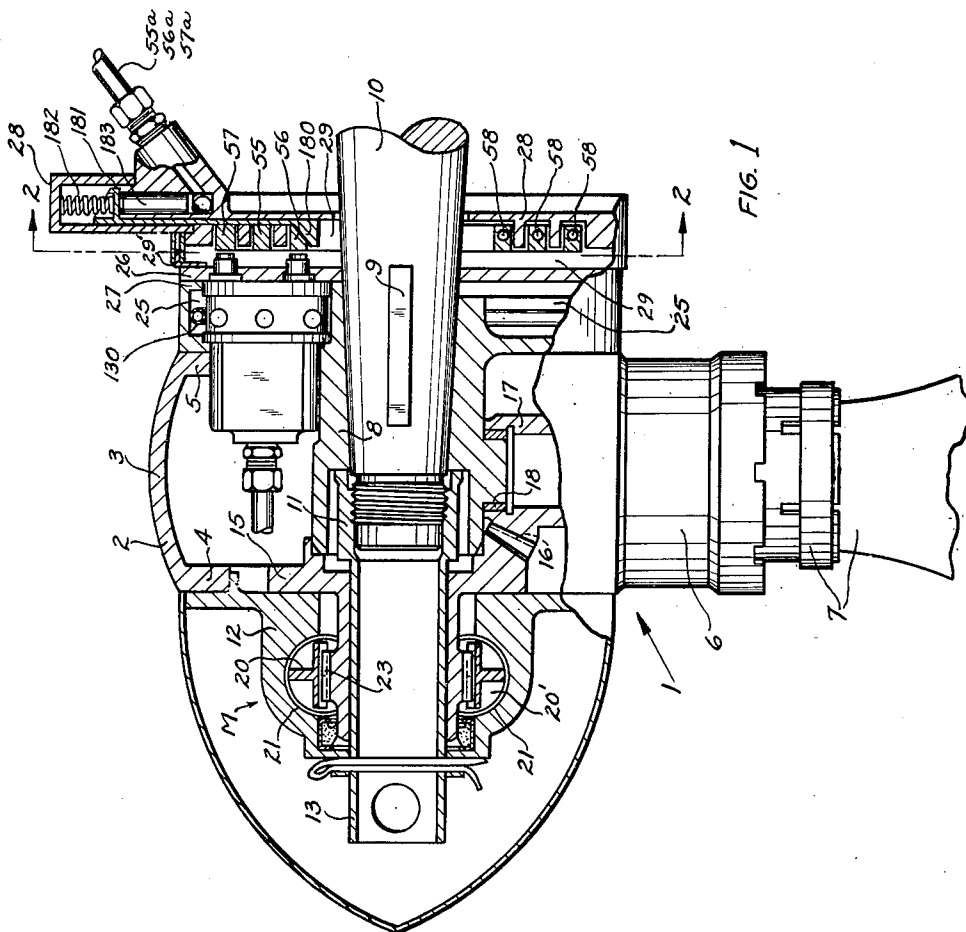


Filed March 1, 1946

VARIABLE PITCH PROPELLER

7 Sheets-Sheet 1



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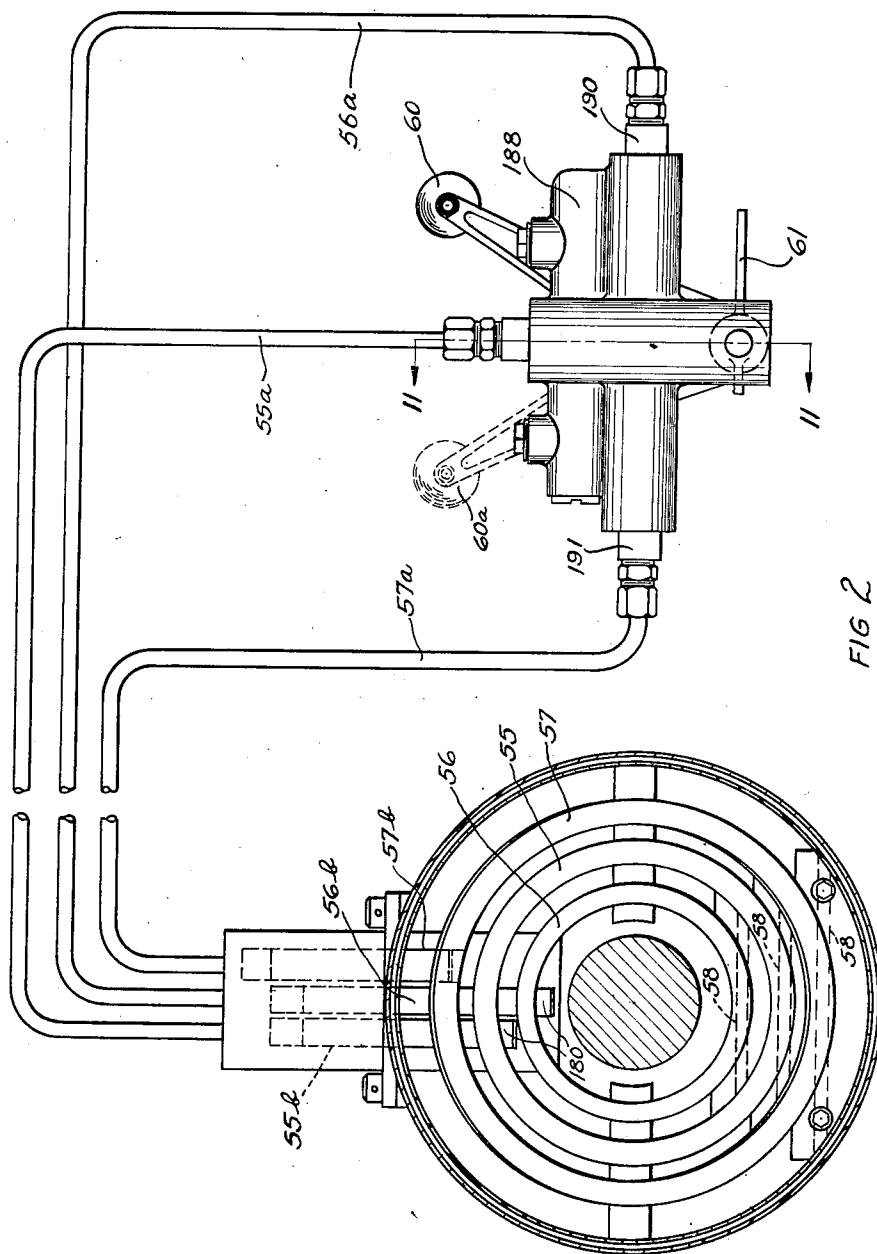
March 4, 1952

C. R. SACCHINI
VARIABLE PITCH PROPELLER

2,588,166

Filed March 1, 1946

7 Sheets-Sheet 2



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2,588,166

VARIABLE PITCH PROPELLER

Filed March 1, 1946

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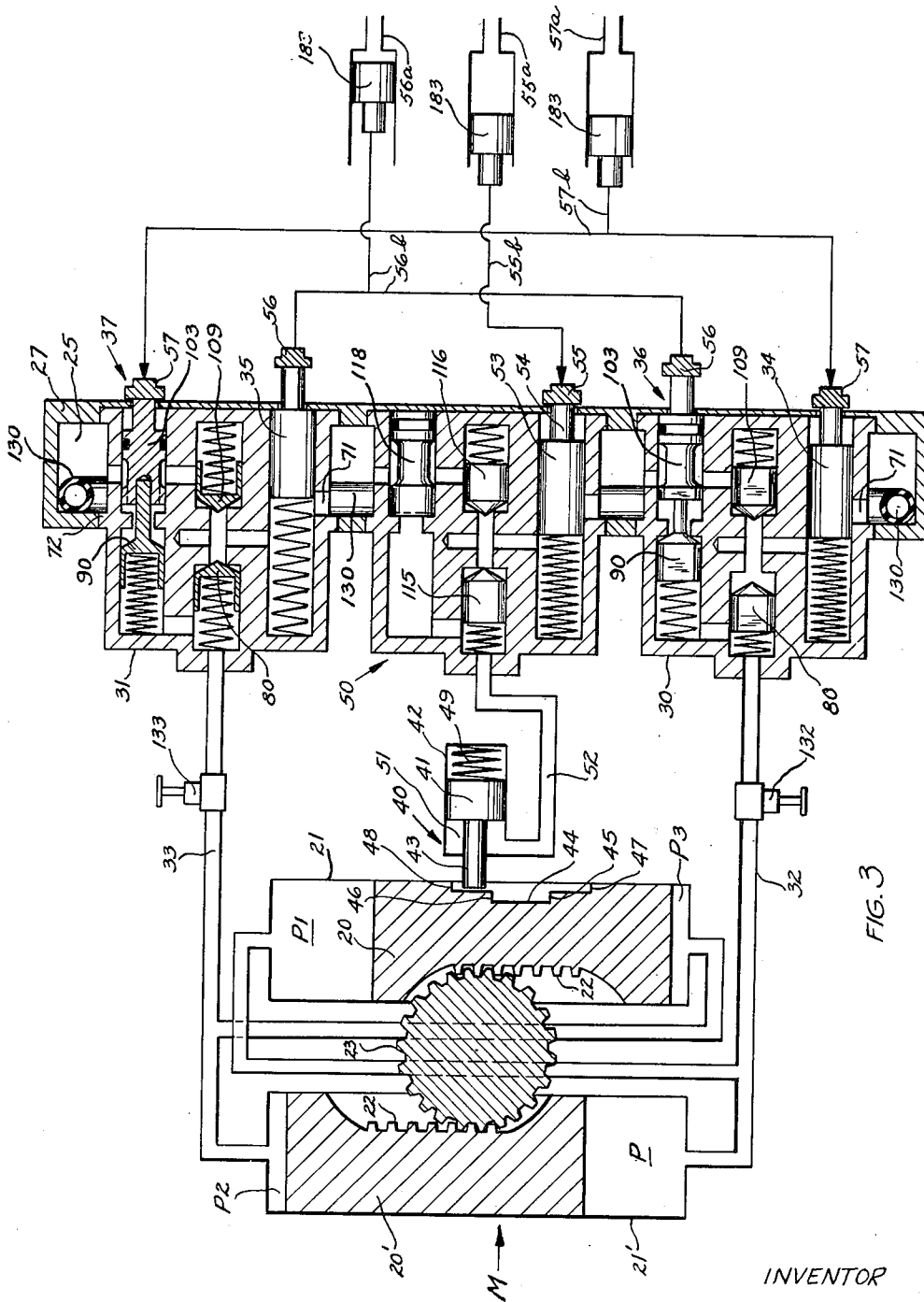


FIG. 3

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VARIABLE PITCH PROPELLER

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7 Sheets-Sheet 4

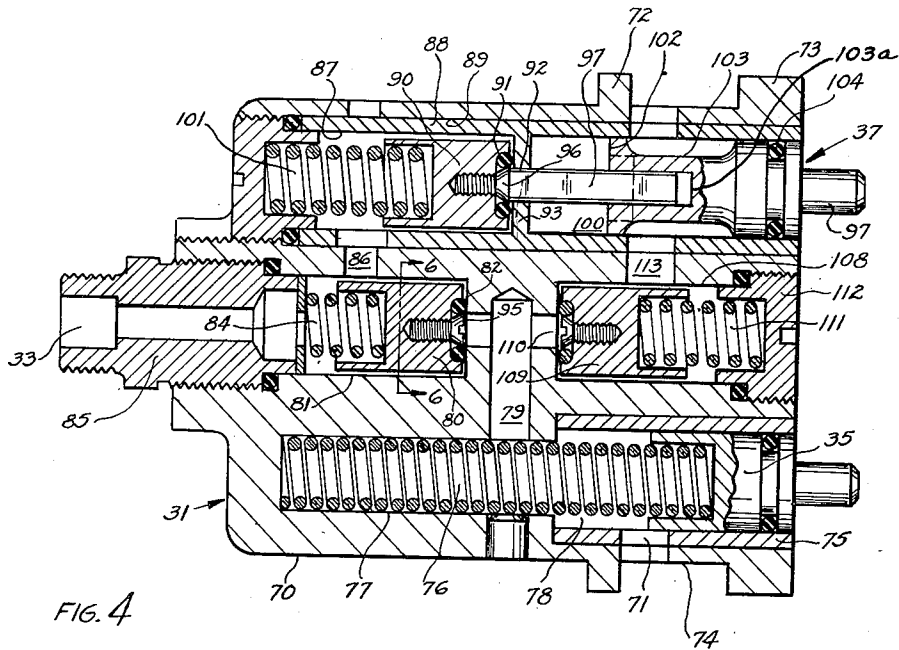


FIG. 4

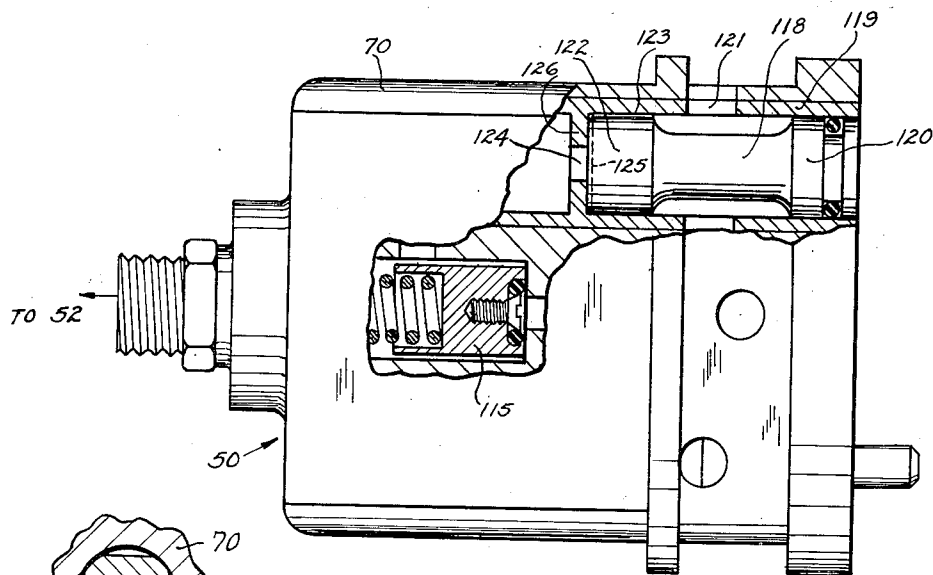


FIG. 5

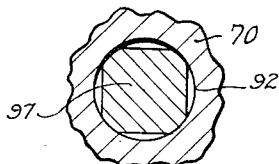


FIG. 6

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2,588,166

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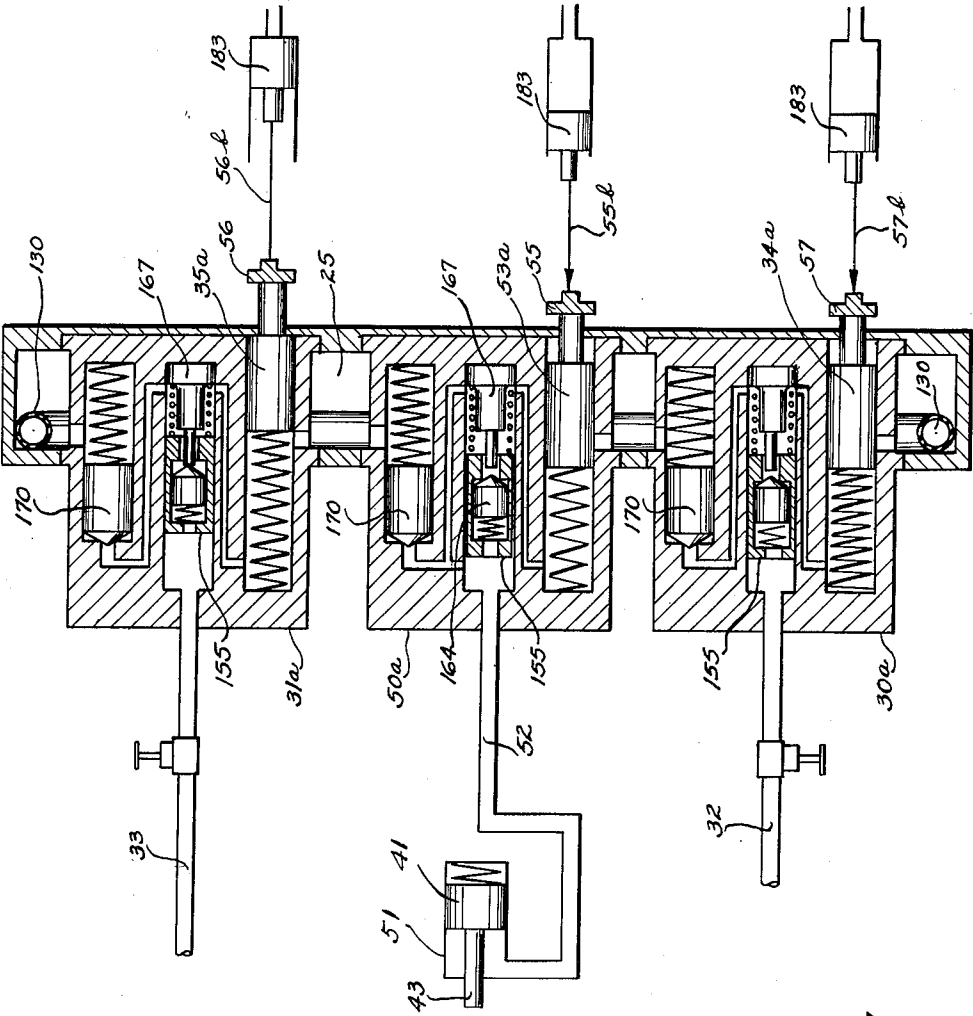


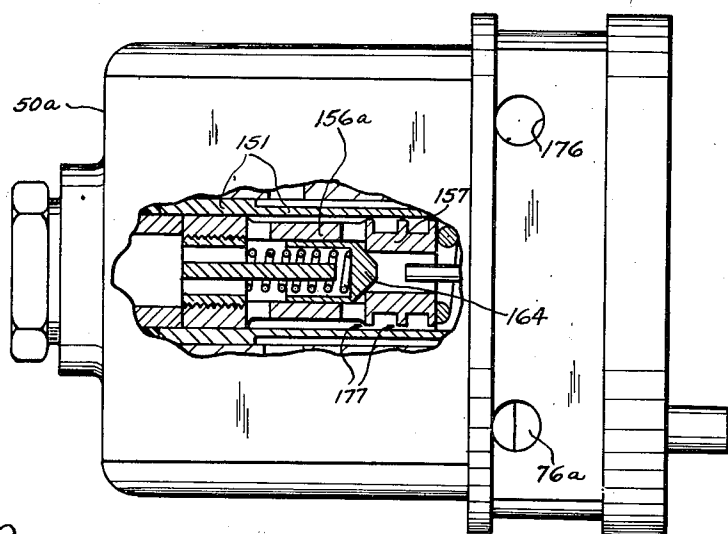
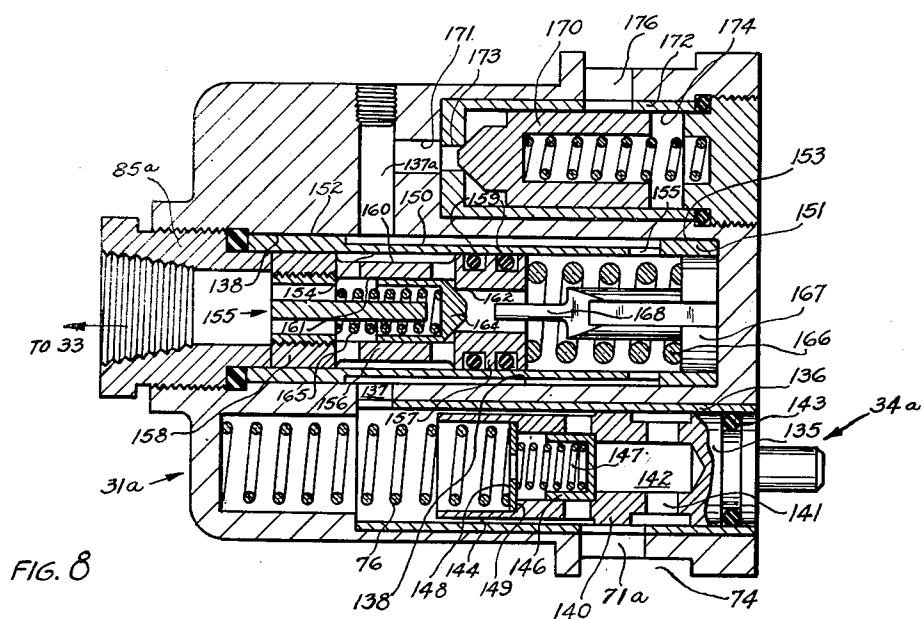
FIG. 7

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Filed March 1, 1946

VARIABLE PITCH PROPELLER

7 Sheets-Sheet 6

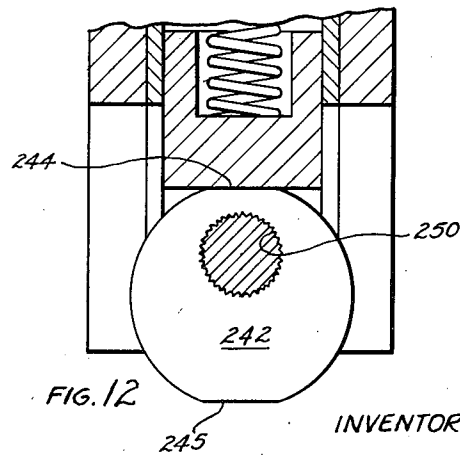
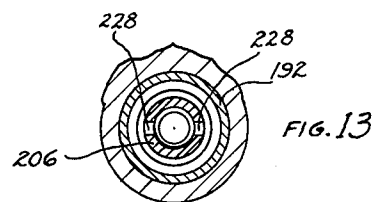
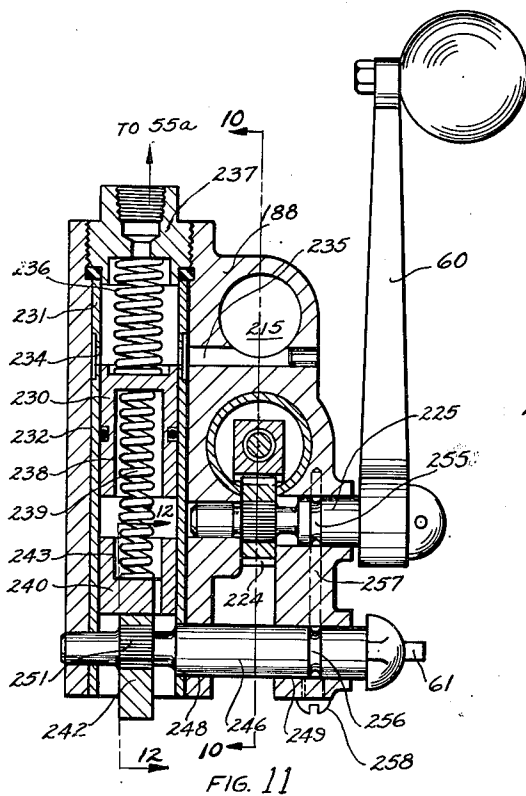
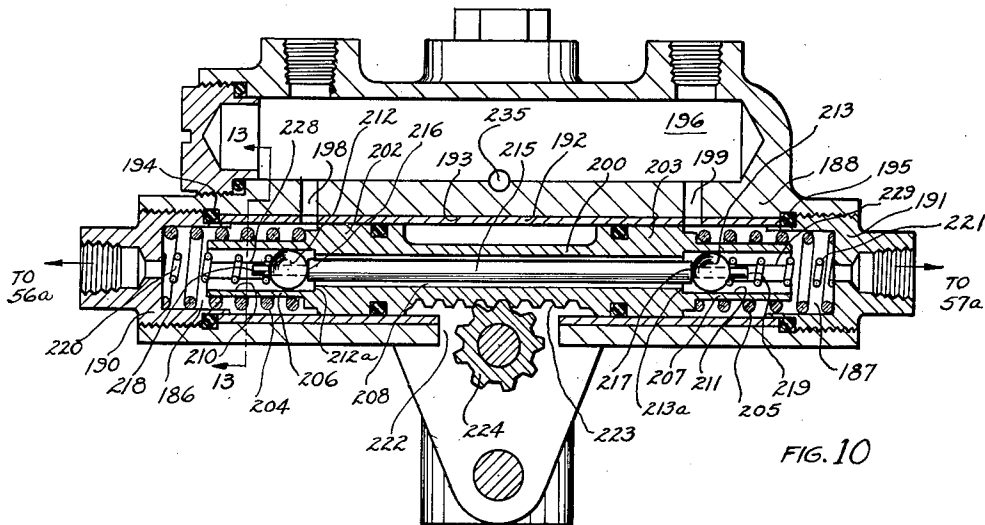


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7 Sheets-Sheet 7



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UNITED STATES PATENT OFFICE

2,588,166

VARIABLE PITCH PROPELLER

Columbus R. Sacchini, Willoughby, Ohio, assignor to The Marquette Metal Products Company, Cleveland, Ohio, a corporation of Ohio

Application March 1, 1946, Serial No. 651,278

5 Claims. (Cl. 170—160.31)

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This invention relates to a variable pitch propeller mechanism and particularly to a mechanism provided with fluid pumping and servomotor means connected with the blade or blades to be adjusted, which pumping means is operated as a consequence of rotation of the propeller.

The general object is to provide an improved hydraulically operated controllable pitch propeller.

Another object is to provide an improved hydraulically operated controllable pitch propeller in which the operating fluid is contained in a sealed system rotatable with the propeller, characterized in that no matter how full of fluid the system is, no part of the fluid control mechanism can thereby be restricted against free operation.

Another object is to provide, in a hydraulically actuated controllable pitch propeller, an improved means for preventing the blade or blades from being moved out of adjusted position by forces opposing the pitch adjusting mechanism of the propeller, and specifically to provide an improved hydraulically-acting anti-creep mechanism for the blade or blades.

A further object is to provide, in a hydraulically operated controllable pitch propeller, an improved means for controlling, independently of the speed of rotation of the propeller, the rate at which the blades are adjusted.

A further specific object is to provide a simplified fluid actuated mechanism for initiating the operation of a blade pitch adjusting means.

A further object is to provide an improved hydraulically actuated selectively operable mechanism for causing initiation of operation of a blade pitch changing mechanism of a propeller.

A further object is to provide, in a hydraulically operated propeller, a blade pitch control system which includes a releasable pitch limiting latch rendered operative in part by fluid in a hydraulically operated actuating system separate from the pitch adjusting mechanism, means to enable the actuating mechanism always to be operated instantly to a full functioning position where it is retained until the pitch limiting latch has been operated in a manner to permit blade adjustment to an abnormal pitch setting.

A concomitant object is to provide an improved means for enabling a latch-release-actuating hydraulic mechanism to be moved quickly manually into a self retaining position so as to store energy for subsequent hydraulic operation.

A further object is to provide an improved manually operable hydraulic actuator for an oppositely acting blade pitch adjusting mechanism of a propeller, which actuator is self centering to neutral position upon release of manually applied force.

Other objects and features of the invention will

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become apparent from the following description of the illustrative forms shown in the drawings, wherein:

Fig. 1 is a longitudinal assembly sectional view in a vertical plane of a propeller mechanism of the three-blade type;

Fig. 2 is a view partly in section as indicated by the line 2—2 on Fig. 1 and showing, in elevation, the preferred hydraulic actuator mechanism;

Fig. 3 is a schematic assembly view showing the fluid supply, pumping mechanism and part of the hydraulic actuator system in one form;

Fig. 4 is a central longitudinal sectional view of one of two fluid control units shown schematically on Fig. 3;

Fig. 5 is a fragmentary central sectional view of a fluid control unit similar to that shown by Fig. 4, provided for control of the pitch limiting latch mechanism;

Fig. 6 is a detail sectional view as indicated at 6—6 on Fig. 4, showing a typical shape of valve plungers used in the fluid control units;

Fig. 7 is a schematic partial assembly view similar to Fig. 3, but showing a somewhat modified arrangement of fluid control mechanism;

Fig. 8 is a view similar to Fig. 4, showing more in detail part of the modified construction according to Fig. 7;

Fig. 9 is an elevational view of the unit similar to Fig. 8, shown partly in central section;

Fig. 10 is a vertical sectional view of the main hand pump or actuator portion of the mechanism shown at the right in Fig. 2;

Fig. 11 is a sectional detail view taken as indicated at 11—11 on Fig. 2;

Fig. 12 is a detail sectional view taken along the plane indicated at 12—12 on Fig. 11, and

Fig. 13 is a detail sectional view taken at 13—13 on Fig. 10.

Referring further to the drawings, Figs. 1 to 5 inclusive, the assembly shown by Fig. 1 shows a propeller mechanism 1 including a central hub 2, formed as a hollow steel forging with a spheroidal central portion 3 and parallel annular front and rear end flanges 4 and 5. The forging includes equally spaced radially extending barrel portions 6, one being shown, for supporting blade assemblies such as 7. Each barrel portion 6 contains suitable bearings enabling the blade assembly to be rotatably adjusted in opposite directions for forward pitch in a normal pitch range, further adjusted forwardly for feathering and reversely adjusted beyond normal low pitch as for braking.

The hub 2 is mounted on a central hub carrier 8, suitably secured as by a key 9 to a tapered portion of an impeller shaft 10. A sleeve-like nut 11 engaging threads on the forward end of the shaft further secures the shaft to the hub carrier. A pitch adjusting servomotor M has a

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housing or block 12, partly surrounding the sleeve portion 13 of the nut 11, and appropriately secured to the hub as against the forward flange 4 thereof.

Journalled between the forward portion of the hub carrier 8 and the servomotor block 12 (partly supported by both) is a master pitch adjusting gear element 15, having gear teeth 16 in mesh with teeth of respective tubular portions 17 of the blade assemblies. Each tubular portion 17 has a pilot bearing at 18 on the hub carrier.

The mechanical parts of the servomotor M comprise identical but relatively oppositely acting pistons 20 and 20' in cylinders 21 and 21' respectively carried in the motor block 12. Each piston 20 has rack teeth 22 in constant mesh with gear teeth 23 on the forward sleeve portion of the gear 15, so that as the pistons are reciprocated the blades are adjusted through the master gear 15 and the tubular portions 17 of the blade assemblies.

The servomotor operating means includes a rear end portion of the hub carrier 8 which is formed to provide the main walls of a sealed annular sump or hydraulic fluid supply chamber 25, closed rearwardly as by a closure plate 26 attached to a rear flange 27 of the annular portion of the hub carrier. The sump has a suitable filler opening, not shown, and is suitably sealed off from the hollow interior of the hub forwardly from the sump and also sealed around the central portion of the hub carrier and impeller shaft. The walls forming the sump chamber thus rotate with the hub whenever the propeller is turned by the shaft 10.

Behind the closure plate 26 of the sump is a fixed, that is nonrotatable, framework 28 providing a generally closed chamber 29 peripherally sealed as by the telescoping joint assembly 29' (top, Fig. 1) so that the chamber 29 may retain suitable lubricant for a fluid control actuator cam mechanism to be described and contained in the chamber 29. Fluid supplied from the sump chamber 25 is selectively pumped to oppositely operating portions of the servomotor M by one or the other of two hydraulic control (pumping and valving) units of substantially identical form indicated at 30 and 31, Fig. 3. A pump plunger 34 of the unit 30, through suitable communicating lines partly indicated at 32, supplies the pressure chambers P and P1 of the servomotor and the pump plunger 35 of the unit 31, through a communicating line diagrammatically indicated at 33, supplies the pressure chambers P2 and P3 of the servomotor.

The hydraulic control units 30 and 31 also contain exhaust valve mechanisms generally indicated at 36 and 37. The exhaust valve mechanisms are operated in timed relation to the operation of the pumps 34 and 35 in such a manner that when, for example, the pump 34 supplies fluid to the servomotor chambers P and P1, the exhaust valve mechanism 37 is operated to vent used fluid from the servomotor spaces P2 and P3 to the sump; and when the pump 35 of unit 31 is operated, the exhaust valve mechanism 36 of unit 30 operates to vent fluid from chambers P and P1.

A pitch limiting mechanism generally indicated at 40, Fig. 3 only, cooperates with an element of the servomotor mechanism normally to limit pitch adjustment of the blades in either direction within a range suitable for forward driving and, abnormally, to limit pitch adjustment beyond that range in opposite directions. As shown, a

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servomotor or plunger 41 of the pitch limiting mechanism operates in a suitable cylinder 42 and has a latch portion 43 normally seated in a recess 44 in, for example, the piston 20.

In the actual design (not shown) the latch 43 cooperates with a stepped depression in the forward face of the gear 15, the plunger 41 being mounted in the pitch adjusting servomotor housing 12.

The normal pitch limits are established by abutments 45 and 46 defining parts of the depression 44; and beyond those abutments, in respective directions of movement of the piston 20, are spaces defined in part by limiting abutments 47 and 48. The plunger 41 is withdrawn to the illustrated position, against the force of a return spring 49, by fluid forced by a pumping unit generally indicated at 50, mounted on the hub carrier and operated essentially the same as are the units 30 and 31. The pump plunger 53 of the unit 50 is connected with a pressure chamber 51 of the latch plunger 41 by a suitable conduit represented in simplified form at 52, Fig. 3. The fluid forcing unit 50 is controlled at the will of the operator to withdraw the latch plunger member 43 from the depression 44 whenever it is desired to effect reverse pitch as by movement of the piston 20 beyond the latch abutment 46 in the direction toward the abutment 48 and also whenever it is desired to move the piston in the opposite direction past the abutment 45 for feathering.

Reference may be had to an application of Gordon W. Hardy, 629,539, filed November 19, 1945, now Patent Number 2,515,037, issued July 11, 1950, and owned by the assignee hereof, for constructional details of a mechanism corresponding essentially to that thus far described and which details are not referred to herein.

The pump plunger 53 of the latch control unit 50 and the pump plungers 34 and 35 each has an operating stem projecting out of the sump rearwardly for operative contact with a respective manually settable cam in the chamber 29. The exhaust valves 36 and 37 have operating stems similarly arranged, all the stems terminating rearwardly in the same plane.

The stem 54 of the plunger 53 makes contact with the intermediate one of three concentric ring cams 55, 56 and 57. The cams are hinged for movement toward and away from the propeller hub on respective pivot pins 58 near the bottom of the chamber 29 and supported by the framework 28. With that arrangement, the upper portion of each of the cam rings can be thrust forwardly at the will of the operator to cause energization of the pumping systems and operation of the exhaust valves. In Fig. 3 the intermediate cam member 55 has been thrust forwardly to cause release of the pitch limiting latch plunger 41—43 of pitch adjusting mechanism and the outer ring 57 has been thrust forwardly to cause reciprocations of the pumping element 34 of the unit 30 and cooperating opening movements of the exhaust valve 37 of the unit 31. Those operations have resulted in movement of the piston 20 toward reverse pitch and beyond the normal pitch limiting abutment 46 with respect to the latch 43.

If, instead of the ring cam 57, the ring cam 56 is thrust forwardly, assuming the other two cams 55 and 57 are retained in their rearward non-operating positions (latch 43 seated in depression 44), the operation would be an actuation of the pump plunger 35 of unit 31 and valve mechanism

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36 of unit 30. The blades would thereby move toward forward pitch, being limited in that movement by the abutment 45 of the latch mechanism. If, during such operation of the ring cam 56, the ring cam 55 were thrust forwardly prior to movement of the piston 20, then the piston 20 would be free to make the necessary traverse of the latch 43 to enable feathering and the feathering operation would terminate when the latch element 43 engaged the abutment 47.

The selective operation of the ring cams 55, 56 and 57 will be more fully described later, but for the moment it may be noted in reference to Fig. 2 that, with an actuator handle 60 in the position in which it is shown, hydraulic fluid is forced through a line 56a to a servomotor of a vertically sliding cam 56b, causing operation of the innermost ring cam 56 on the pump 35 of the unit 31 and exhaust valve 36 of the unit 30, thus causing forward pitch adjustment of the blades. With the handle 60 in a reversed position as indicated by broken lines at 60a, Fig. 2, actuator fluid is supplied through a line 57a to the servomotor of a cam 57b, thus causing the operation of the pump plunger 34 and the exhaust valve mechanism 37 of units 30 and 31 respectively so that the blades are moved toward low pitch or reverse positions. Actuation of the intermediate cam ring 55 is accomplished by a movement of a separate actuator handle 61, Fig. 2, from the position in which shown through 180°, causing actuating fluid to be fed through a line 55a to a servomotor of a cam 55b, thus to move the latch element 43 from between the abutments 45 and 46.

Referring particularly to Fig. 4, wherein one of the fluid control units 30 and 31 is illustrated in detail, it will be noted that said unit comprises a generally cylindrical body 70 adapted to fit and sealingly close an opening in a forward wall of the sump chamber at a peripheral flange 72 of the body. Another flange 73 of the body makes peripheral sealing contact with the rearward wall 27 of the sump chamber so that only the reduced diameter space 74 between the two flanges can communicate with the sump through suitable ports in the body. The pump plunger 35 is mounted in a sleeve 75 having an inlet port 71 communicating with an aligned port in the wall of the body 70 leading to the sump. A spring 76 in a bore 77 of the body forces the plunger 35 toward the ring cam 56 to the inoperative position shown on both Figs. 3 and 4. Reciprocation of the pump by the ring cam 56, Fig. 3, causes movement of fluid trapped in the pump chamber space 78 through a cross passage 79 to a check valve plug 80 disposed centrally of the body 70 and slidable in an axial bore 81 of said body. The preferred cross sectional shape of the check valve plug is as shown by Fig. 6, i. e. non-circular to allow fluid to pass the plug. The valve 80 preferably has a ring seal 82 closing against a flat rearward face defining part of the bore 81, which face is centrally ported axially of the body for communication with the cross passage 79. A fairly light weight return spring 84 operates the valve plug 80 to cause the ring seal to close said central port whenever the pump plunger 35 ceases to move to the left as illustrated on Fig. 4.

The fluid pumped past the check valve 80 passes through the central bore of a fitting 85 to the delivery line 33, formed partly in the fitting, but said fluid is also free to flow through a cross passage 86 to the central bore 87 of an

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exhaust valve sleeve 88, sealingly fitting an eccentric bore 89 in the body 70. Exit of fluid from the bore 87 is normally prevented by a valve plug 90 similar to the plug 80 and guided by the bore 87. The plug 90 has a ring seal 91 closing a port 92 formed in a wall 93 of the sleeve 88 intermediate the ends of said sleeve. The ring seals 82 and 91 are of similar construction, being secured to the respective plug members 80 and 90 by respective screws 95 and 96 having tapered or flared heads. The flared head of the screw 96 has a non-circular stem or extension 97 integral therewith which projects slidably through the port 92 but does not seal said port. The port 92 communicates with a chamber 100 in the rearward end of the sleeve 88 and when once the ring seal 91 is moved off the valve seat around the port 92 against the pressure of a return spring 101 in the sleeve, fluid is free to flow from port 92 past a non-circular land 102 of a valve operating plunger 103. The plug element 90, seal 91, stem 96 and the valve operating plunger 103 constitute the moving parts of the exhaust valve mechanism 37. The plunger 103 is sealed as at 104 against exit of fluid from the rearwardly open bore of the sleeve 88, but rearwardly from the non-circular land 102 of the plunger 103 the sleeve and body are formed to provide an outlet port at 105 leading to the sump. The non-circular stem 97 of the exhaust valve plug 90 extends into a socket 103a in the plunger 103 a sufficient distance so that reciprocation of the plunger operates the plug 90 to open the latter.

The length of the stem 97 may be different in unit 30 from the corresponding stem in unit 31, since the exhaust valve of one unit 30 or 31 operates in advance of the cooperating pump plunger 34 or 35 of the other unit; while the other exhaust valve operates in trailing relation to its cooperating pump. When the two valve stems 97 are of proper length for abutment by the bottom of the socket 103a of the associated plunger 103, the exhaust valves and pumps are in phase. Hence, it is merely necessary to select the proper length of stems 97. Alternatively adjustable abutments (not shown) can be provided between the stems 97 and sockets 103a, which abutments would be set and locked prior to assembly of the propeller mechanism.

Additionally, for protecting the hydraulic mechanism against damage by overpressure after the latch 43 has engaged one of the abutments 45-48, the body 70 is provided with a relief valve communicating with the cross passage 79 and preferably located coaxially of the check valve 80 in a rearwardly open bore 108 of the body 70. The plug 109 of the relief valve is formed similarly to the plug 80, having a ring seal closing against a flat surface around the central relief port 110. The relief valve differs from the check valve 80 principally by having a much stronger return spring shown at 111 and seated at its opposite ends in a recess of the plug 109 and a similar recess in a fixed plug 112 closing the bore 108. The bore 108 has an outlet passage 113 leading to the sump chamber, past the smaller diameter central portion of the exhaust valve plunger 103.

In operation, the pulsations of the pump plunger 35 cause movement of fluid through the conduit 32 to operate the servomotor pistons 20 and 20' to the desired positions or until the pistons are stopped by relative abutment of the pitch limiting latch mechanism elements. After such relative abutment any further operation of

the pump plunger causes movement of the relief valve plunger 109 to the right, Fig. 4, dumping excess pressure fluid to the sump chamber 25. To exhaust fluid from the servomotor M, the plunger 103 of the unit 30 is operated by the appropriate ring cam, said plunger 103 of unit 30 abutting the stem 97 of the associated valve plug assembly (90, 97), thus opening the valve and allowing fluid to move through the line 33 to the sump chamber. Due to the non-circular cross section of the stem 97 of the exhaust valve plug 90, there is no pocketing of fluid within the central bore of the actuating plunger 103 in which the stem slides.

The pump portion 53 of the latch release control unit 50 is the same as that already described in connection with Fig. 4 and it will be noted in Fig. 3 that a counterpart of the check valve 80 is provided as at 115 ahead of the delivery line 52 leading to the latch release plunger chamber 51. Also, the unit 50 has a relief valve 116 to prevent damage to the latch mechanism upon over-delivery of fluid by the pump 53. The relief valve 116 is identical with relief valve 109, 111, Fig. 4.

Venting of operating pressure from the latch plunger chamber 51 (Fig. 3) after the release of the latch has been accomplished requires only a bleed-off operation. As shown at the upper right in Fig. 5, a fixed plug 118 corresponding in position to the exhaust valve plunger 103, is seated in the rearward portion of a sleeve 119. The plug 118 is formed with a central necked portion terminating rearwardly at an enlarged head 120, sealing the rearward end of the sleeve 119 behind an exhaust port 121 leading to the sump chamber. Forwardly from the port 121 a relatively smaller diameter enlargement 122 of the plug 118 occupies the central bore of the sleeve 119 and provides a peripheral gap 123 around the enlargement 122 and within the sleeve. The gap may be on the order of a few thousandths of an inch radial width to provide a restricted orifice for passage of fluid from the port 124 which is open to communication with the delivery line 52 as shown by Fig. 3. The forward end of the plug 118 has a groove 125 across the face of the plug which rests against the partition 126. Thus, whenever the pump 53 of the latch release control unit 50 is operated, the latch 43 is withdrawn from position necessary to make contact with either latch abutment 45 or 46, and the restriction at 123, Fig. 5, blocks return movement of the latch to initial position temporarily only. When the pump 53 is caused (by the operation of the mechanism through manipulation of the actuator handle 61, Fig. 2) to cease operation, then the operating fluid for the latch plunger is bled from the supply conduit 52 and the connecting passages to the sump chamber through the restricted orifice 123 and the latch is returned to its initial forwardmost position by the spring 49. Said spring forces the actuating fluid past the restricted orifice.

It will be seen from comparison of Figs. 3 and 4 that, when either exhaust valve assembly is operated by an appropriate ring cam 56 or 57, the operation of moving the valve plungers 103 effects diminishment of the space within the right hand or rearward portions of the valve sleeves 88. If, at such time as an exhaust valve is operated, the sump or supply chamber 25 and the communicating conduits leading from the servomotor M are filled with fluid, the operation of opening the exhaust valve

will be blocked because hydraulic fluid is substantially incompressible. Also, after either pump plunger 34 or 35 on its pumping stroke has closed its associated inlet port 71 and assuming the cooperating exhaust valve has been opened, a complete charge of fluid in the sump chamber 25 and communicating passages leading to the then exhausting cylinder chambers of the servo motor M will block further movement of such pump plunger. In actual practice, with a similar system it was found that it was impossible to fill the sump chamber entirely with operating fluid because, until such time as some of the charge had been forced or bled out of the system, the cam rings could not be moved to exhaust valve and pump plunger operating positions. To overcome that difficulty, a readily compressible element 130 is placed in the sump chamber 25. In the mechanism shown by Figs. 1 and 2, the compressible element is preferably in the form of a tubular elastic ring formed either as a single length of tube sealed at both ends or as an endless ring. Said element could be of any other desired shape. A suitable material for the ring 130 is synthetic rubber. The ring lies outwardly from all the pump and valve units 30, 31 and 50 as against the inner surface of the radially outward wall of the sump chamber.

In order, independently of the speed of rotation of the propeller, to govern the rate at which pitch adjustment is effected by operation of the pump plungers 34 and 35, each of the feeding and exhausting conduits 32 and 33 is provided with a suitable needle valve with a restricted orifice (not shown).

Conventional needle valves are represented at 132 and 133 respectively in the lines 32 and 33. Those valves are not adapted or intended to be adjusted during flight. Since the strokes of the plungers 34 and 35 are of a length determined by the set or adjusted positions of the cam rings 56 and 57 and that may be varied by the operator and the speed of reciprocation of the pump varies according to propeller speed, the rate of fluid delivery to the servomotor chambers P, P1, etc. would ordinarily vary greatly. By restricting the passage of fluid from the pumps to the servomotor, the rate can be controlled at a value considerably less than the maximum capacity of the pumping mechanisms. In event the propeller is rotating above the speed required for cruising speed, for example, the pumps will tend to deliver fluid at greater than the desired rate and, in that event, the over-delivery will be returned to the sump by opening of the relief valves 109 in case of the control units 30 and 31.

In the particular arrangement above described, no anti-creep valve mechanism is necessary, since the exhaust valve operations are timed with reference to the pumping operations and whenever the exhaust valves are closed no amount of force exerted on the blades can force exhaust fluid from either conduit 32 or 33. Return flow to sump past the pump plungers, while the plungers are in position to open the fluid inlet ports 76, is prevented by the check valves 80-82, Fig. 4.

Referring to the arrangement according to Figs. 7, 8 and 9, these differ from the construction described above principally in that anti-creep valve mechanisms are provided and the mechanically operating exhaust valve mechanisms are omitted. The cam ring 56, Fig. 7, operates only the pump plunger 35a correspond-

ing to the plunger 35 of Fig. 3 and the cam ring 57 operates only the pump plunger 34a which corresponds to the plunger 34 of Fig. 4. The cam ring 55 has exactly the same function as described in connection with Fig. 3, namely, to operate the pump plunger 53a of the pitch limit latch release mechanism. The fluid delivery and exhaust lines 32 and 33 are identical with the previously described construction. The servo portion of the latch release mechanism is the same as previously described and the cam rings are or may be adjusted as previously described above, through the servomotor portions of the manual actuating mechanism (right, Figs. 1 and 2).

Referring especially to Fig. 8, attention is called to the two-part plunger arrangement 135 which replaces the single piece pump plungers 34 and 35 of Figs. 3 and 4. The plunger member 135 slidably seals the cylinder sleeve 136 at the rearward end of the sleeve and the forward end of said sleeve is open at 137 for communication with the interior of a central bore 138 of the body 31a, corresponding to the body 31 of Fig. 4. The body 31a is generally the same as the bodies 30, 31 and 50 already described. An inlet port 71a leads from the annular space 74 of the body 31a to the interior of the sleeve 136 and is initially open at both sides of a sealing land or relatively enlarged portion 140 of the plunger member 135. Behind the land, the plunger member has cross passages at 141 communicating with a central forwardly open socket 142 of the plunger. Rearwardly from the cross passages 141 the plunger member 135 is slidably sealed against the inner wall of the sleeve 136 as by a ring seal 143. In the fully retracted or rearmost position of the plunger 135, fluid is free to pass from the sump chamber 25, through the cross passages 71a to an annular space 144 around the forward end of the plunger member 135 and within the sleeve 136, thus to charge the pump. As the plunger member is thrust forwardly by the cam ring 56, the land 140 seals off the passage 71a so that the fluid occupying the pump sleeve is forced through the passage 137. The return stroke of the plunger member 135 may be accomplished by a fairly light spring 76a as compared to the high rate return spring 76 of Fig. 4 because return movement of the plunger member 135 is, in the Fig. 8 arrangement, unrestricted by vacuum after the check valve (to be described) at the delivery end of the unit 31a closes.

The vacuum breaker arrangement includes a check valve plug 146 which acts to admit low pressure fluid freely to the annular space 144 through the rearward inlet passages 141 and 142 of the member 135 on the return stroke of the pump plunger. The plug 146 is held in sealing position by a light spring 147 bearing at one end against an end wall portion of the plug 146 and at the opposite end against a perforated disc 148 within the pump plunger 135. The disc is held in position against a shoulder 149 of the pump plunger by the force of the spring 76a which is materially greater than the force of the return spring 147 of the vacuum breaker valve.

It will be seen that during return movement of the pump plunger member 135, the vacuum breaker valve 146 is free to move forwardly to open position (to the left) within the plunger member; wherefore, until the forward edge of the land 140 has moved rearwardly to open the inlet passage 71a to the pressure chamber ahead of the pump plunger assembly, fluid is free to

move from the sump through the pump inlet passage 71a, cross passages 141 and socket 142 to the pressure chamber of the pump.

The fluid delivered by the pump to the passage 137 travels by way of the central bore 138 in a space provided by a reduced diameter portion 150 of a valve sleeve 151. The sleeve 151 is held in fixed position by the fitting 85a to which the delivery duct 33 is connected. Said sleeve has land portions at 152 and 153 tightly fitting the bore 138. The sleeve 151 has a central smooth bore 154 from end to end and a rearward portion of the sleeve ahead of the land portion 153 is opened by way of cross passages 155 to said smooth bore 154. Within the bore 154 is slidably mounted a piston assembly 155 which forms the main portion of the anti-creep valve mechanism. The same valve mechanism functions for exhaust of fluid from the pitch adjusting servomotor.

The anti-creep and exhaust valve piston assembly 155 includes a spool shaped body member 156 having lands 157 and 158 at its rearward and forward portions respectively, slidably fitting the central bore 154 of the sleeve 151. It is only necessary that the land 157 be especially provided for sealing said bore 154 and ring seals 159 are provided for that purpose. Between the two lands of the piston body member 156, the latter has a reduced diameter portion 160 with cross passages at both ends of said portion. The annular space thus provided around the piston body member communicates with the interior axial main bore 161 of said member. The bore 161 terminates rearwardly in a reduced diameter bore portion 162 so as to form a check valve seat for a valve plug 164 which is closed against the seat by a light spring 165. The fluid forced by the pump to the interior of the sleeve 151 through the cross passages 155 forces the check valve plug 164 off its seat and then passes through the main bore 161 to the outlet fitting 85a. The check valve plug 164 normally retains the fluid forced by the pump.

The piston body 158 is held in the illustrated position against the fitting 85a by a relatively strong spring 166 seated in the bottom of the bore 138 against a plug element 167. The plug has a stem with a reduced diameter end portion 168 extending into the bore 162 of the plunger 156 in position to move the relatively small diameter check valve plug 164 off its seat whenever return pressure in the delivery line leading to the servomotor exerts sufficient pressure on the piston to overcome the spring 166 and move the piston assembly rearwardly or to the right as illustrated in Fig. 8. Then the stem portion 168 engages the rearward end of the check valve plug 164 and unseats said plug. Thereupon, return pressure in the delivery line 33 bypasses the check valve 164 and escapes from the valve body 31a by way of the pump plunger cylinder sleeve 136 and the then retracted pump plunger member 135. The pressure maintained by the spring 166 is predetermined in accordance with whatever force it takes to hold the blades from being moved out of adjusted position by counter pressure on the blades.

The operation of the anti-creep valve mechanism for servomotor exhausting purposes is the same as described above. The pump pressure of unit 30a overcomes the pressure of the spring 166 of unit 31a and vice versa for exhaust.

In the event operation of either pump 34a or 35a, Fig. 7, is continued until the latch plunger

43 engages one of the pitch limiting stops described in connection with Fig. 3, the over-delivery of the pump is vented from the unit 30a or 31a to the sump through a respective relief valve assembly 170, Fig. 8, the construction and operation of which is essentially the same as that of the relief valve 109—111 of Fig. 4. To communicate the annular space around the sleeve 151 with the relief valve, a portion of the cross passage 137 designated 137a leads to an outlet bore 171 in a valve sleeve 172. The relief valve plug closes a small forward port 173 in the sleeve 172 and the shank or guiding portion of the plug is non-circular so as not to seal the bore 174 of the sleeve 172 in which the plug slides. The main body of the unit 31a has an outlet passage at 176 for communicating the interior of the sleeve 172 with the sump. The return spring for the relief valve plug has usually nearly twice the rate of the anti-creep valve spring 166.

The anti-creep valve mechanism of Fig. 8 is not needed for the control unit 50a of Figs. 7 and 9 which controls the pitch limiting latch operating servomotor. A restricted orifice for exhausting fluid from the latch release control system (line 52 etc., Fig. 7) is needed and that can easily be provided by removing the sealing rings 159 from the rear land of the piston member 156a as shown by Fig. 9. The annular gap 177 (Fig. 9) between the piston land 157 and the inner bore of the sleeve 151 serves as the bleed-off aperture for the latch operating servo system.

The valve construction according to Fig. 8 may be modified so as to employ the compressible ring seal closures and flat valve seats such as provided for the valve plugs 80, 99 and 109 of Fig. 4. In other words, in Fig. 8 the anti-creep valve plug 164 and the relief valve plug 170 may be made with appropriate recesses for ring type seals, held in place as by flared head screws and closing against flat valve seats. Such construction greatly reduces the time required to machine the valve seats, since the ports closed by the compressible ring seals do not have to be aligned accurately with the guide bores for the plugs. Moreover, centrifugal force acting on the flat-seat-engaging valve plugs of Fig. 4 never affects the sealing operation, even though the valve plugs have considerable lateral clearance in the guide bores. In the type of valve plug employed as at 164 and 170 in Fig. 8, the lateral clearance in the guides must be reduced substantially to a minimum consistent with easy sliding of the plugs. A further advantage of the flat seat and compressible ring seal closure is that foreign particles in the hydraulic fluid are less apt to cause faulty valving operation.

Attention is called to the fact that it is unnecessary in the fluid control units according to Figs. 4, 5, 8 and 9 to seal the rear end faces of the main unit bodies against the back closure plate 26, Fig. 1. That feature is of special importance in connection with the relief valves 109, Fig. 4, and corresponding relief valve 170 of Fig. 8. The rearwardly open bores in the unit bodies which receive the valve plug assemblies are plugged by positively acting compressed linear seals so that when the valve assemblies are subjected to sufficiently high pressure to open the relief valve plugs there is never any likelihood of loss of fluid from the sealed hydraulic system as along the operating stems of the ring cam actuated parts.

Referring further to the ring cam actuating mechanisms shown by Figs. 1 and 10 to 13 inclusive, the sliding cams 55b, 56b and 57b are mounted on the framework 28 in suitable guides and each has a bevelled lower end portion 183 (one shown in Fig. 1) which, through contact with an adjacent bevelled rear surface of the associated ring cam, thrusts the selected ring cam forwardly into pump or valve actuating position with respect to the control units 30, 31, 50, etc. as already described. Each of the sliding cams has an arm portion at its upper end as at 181, Fig. 1, which is normally forced downwardly by a spring 182 to hold the sliding cam in inoperative position. Adjustable limit stops for the sliding cams may be provided on the framework 28 in position to engage the upper ends of the sliding cams. To operate the sliding cams, each arm 181 overhangs a respective servomotor plunger or piston 183 in a suitable cylinder bore of the framework 28. The actuator feed lines 55a, 56a and 57a communicate with the cylinders of respective pistons 183 in sealed relation thereto.

Referring to the actuator pump mechanism of Figs. 10 to 13 inclusive, the lines 55a and 57a are supplied by oppositely disposed pump pressure chambers 186 and 187 in a common housing 188 having fittings 190 and 191 connected to the respective feed lines 56a and 57a, Fig. 2, and also acting as sealing plugs for the outer ends of the chambers 186 and 187. A common cylinder sleeve 192, Fig. 10, is fitted tightly into a horizontal bore 193 of the housing 188. The end fittings 190 and 191 have ring seals 194 and 195 pressed against the respective ends of the cylinder sleeve. The pressure chambers 186 and 187 are each open to communicate with a common supply chamber 196 for reserve fluid preferably formed in the housing 188 above the cylinder sleeve 192. Vertical ports 198 and 199 formed in the cylinder sleeve and housing 188 communicate the respective pressure chambers with the reservoir.

Slidable in the sleeve 192 is a double ended piston 200 having identical enlarged land or head portions 202 and 203 suitably sealed against the wall of the cylinder sleeve. The land portions, in the illustrated neutral or inactive position of the piston, are so disposed that both supply passages 198 and 199 are open. The piston is held in the centered neutral position by fairly strong springs 204 and 205 of equal length and scale. The springs surround reduced diameter extensions 206 and 207 of the piston and bear oppositely upon shoulders adjacent the piston heads and may be seated at their opposite ends in respective socket portions of the plugs 190 and 191.

In order to use a single piston for supplying fluid selectively to the feed lines 56a and 57a, it is necessary to provide a fluid bypass from one pressure chamber to the other in order that return of the piston to centered position will not be blocked by vacuum in the last active pressure chamber. The bypass is formed as a central axial bore 208 in the piston 200, said bore terminating in respective counterbores 210 and 211 within the reduced diameter extensions 206 and 207. The counterbores provide check valve guides for valve plugs, each in the form of a metal ball as indicated at 212 and 213, slidably fitting the guides. The piston extensions 206 and 207, Figs. 10 and 13, have slots as at 228 and 229 leading from the outer ends of the extensions past the balls so that fluid from the

pressure chambers can move freely past the balls to the central bore 208. The balls are held off their associated valve seats 212a and 213a by a rigid push rod 215 which extends axially through the bore 208 out of contact therewith for its entire length. Opposite reduced ends 218 and 219 of the rod provide shoulders 216 and 217 which abut the valve balls in such a position that both balls are held out of contact with their seats in the position of the rod 215 illustrated in Fig. 10. The reduced diameter ends of the rod 215 extend through diametral openings in the balls, partly to provide supports for respective ends of rod and ball centering springs 220 and 221. The centering springs are very light in comparison to the centering springs for the piston, but the light springs are also of substantially equal length and scale in order to center the ball and rod assembly.

To operate the piston 200, the lower side of both the cylinder sleeve 192 and the housing 188 are open as at 222 and adjacent the opening a reduced diameter central portion of the piston between the lands is provided with rack teeth 223. The rack teeth continuously mesh with a pinion 224 supported on a horizontal shaft 225, suitably journaled in the housing 188 as shown in Fig. 11. The shaft is suitably keyed as by flutes or serrations to the pinion 224 and the outwardly projecting end of the shaft carries the actuator handle 60. The handle 60 is vertical in the neutral position of the actuator piston 200.

In operation, the handle 60 is normally held in its vertical position by the relatively heavy centering springs 204 and 205. Movement of the handle in either direction away from said central position as to the full line or broken line positions shown on Fig. 2 effects movement of the piston 200 in the selected direction for operation of the ring cams 56 and 57 of the propeller control mechanism. Movement of the piston 200 to the left, Fig. 2 (right, Fig. 10), as by turning of the arm 60 toward the broken line illustrated position, Fig. 2, traps fluid in the pressure chamber 187 when the piston land 203 cuts off the supply passage 199. The operation also causes closing movement of the check valve 213 due to the fact that the springs 221 and 221 continue to center the ball and rod assembly 212, 213 and 215. Thereupon, the trapped charge of fluid in the chamber 187 moves through the supply line 57a to the associated servomotor 183 to move the ring cam 56 to operating position. An opposite movement of the piston 200 (right, Fig. 10) obviously operates the ring cam 57 as will be readily apparent.

In order to release the operating ring cam and allow it to be returned to initial position as by contact with the actuating stems of the operating pump and/or valve assemblies of units 30 and 31, the operator merely releases the handle 60 and the compressed high rate centering spring returns the piston and the arm to the centered position illustrated in Fig. 10. Since the centering springs 220 and 221 of the ball check valves 212 and 213 are relatively weak, the ball 212 or 213 which sealed its center port leading through the piston will be opened by inertia so that fluid is free to pass from one pressure chamber to the other as soon as the operator releases the handle 60.

The separate means for operating the pitch limit latch releasing ring cam 55 is a vertical

piston 230 slidable in a sleeve 231 in a vertical bore 232 of the housing 188. The piston 230 has its upper end normally held in position to unseal lateral inlet ports 234 of the sleeve 231 leading to the supply chamber 215 as through a cross passage 235 in the housing. The piston 230 is held in the illustrated position by a spring 236 acting directly against the top end of the piston 230 at its lower end and seated against a socket in an outlet fitting 237 of the housing to which the fluid line 55a is connected. Normally balancing the spring 236 to hold the piston 230 normally in its illustrated position, the lower end of the piston 230 has an axial closed socket 238 which receives the upper end of a coil spring 239. The lower end of the coil spring 239 bears against a cross head 240 slidable in the lower end portion of the cylinder sleeve 231 and resting against a rotatably mounted operating cam 242. The cross head 240 has an axially vented socket 243 which receives and supports the lower end of the coil spring 239. The cam 242 preferably comprises a metal disc of generally circular form but provided with opposite flat surfaces at 244 and 245, 180° apart about an eccentric supporting shaft 246 keyed to the cam as by serrations at 251. The shaft is suitably journaled as at 248 and 249 in spaced portions of the housing 188.

When the operating handle 61 of the pitch limit latch release actuator mechanism is in the position illustrated in Fig. 2, the flat surface 244 (Fig. 12) is in contact with the lower end of the cross head 240, thus retaining the cam and handle 61 in the positions illustrated in Figs. 11 and 12. When the handle 61 is moved through 180° in either direction, the cross head 240 is raised until finally the flat surface 245 is moved in contact with the lower end of the cross head. That operation results in compressing the coil spring 239, overcoming the counter-pressure of the spring 236 and raising of the piston 230 to cut off the inlet ports 234 and move the trapped fluid in the upper end of the cylinder 231 to the servomotor of the intermediate ring cam 55. That causes initiation of operation of the pitch limit release latch 43 of Figs. 3 and 7.

The cooperating flat surface 245 of the cam 242 and the lower flat face of the cross head 240 hold the cross head 240 in its raised position and maintain pressure on the piston operating spring 239. Because of the yielding connection between the cross head and the piston 230, the piston cannot block the cam 242 in its movement toward piston actuating position, even though the pressure chamber spaces and connecting delivery line 55a are of insufficient capacity to receive the volume of fluid attempted to be forced thereto. Because of the possibility of leakage in the pressure chambers of the pitch limit latch release actuator system, the mechanism is so designed as to provide some over-delivery of fluid to the servomotor of the sliding cam 55b, but no matter how much provision for over-delivery is made in order to insure immediate response of the pitch limit latch release servomotor, the cross head 240 can always be moved to its acting position where it is retained by the cooperating flat surfaces 245 of the cam and the flat bottom end of the cross head 240.

A further feature of the construction shown by Fig. 11 is the provision of a common means for holding the actuator piston operating shafts 225 and 246 in their respective journal bores. As

shown, the shaft 225 has a peripheral groove at 255 and vertically aligned therewith the shaft 246 has a peripheral groove 256. Formed adjacent and intersecting the two grooves the housing 188 has a vertical bore 257 which receives a securing key tangent to the peripheral grooves of the two shafts and occupying said grooves. The key may be formed simply as a short section of wire (not shown) held in position by a cap screw 258 in the lower end of the bore 257.

I claim:

1. In a controllable pitch propeller, a pitch adjustable blade, two simultaneously and oppositely acting blades adjusting power mechanisms and means turned by said mechanisms to change the pitch of the blade, all rotatable with the propeller, a fluid operated actuating mechanism including servomotors respectively operable to cause operation of the blade adjusting means, said actuating mechanism including a double ended piston and cooperating cylinder means forming pressure chambers connected with respective servomotors, said piston being movable manually in opposite directions from a neutral position, spring means arranged to return the piston to neutral position after movement in either direction, and a by-pass valve system capable of communicating said pressure chambers, said system including spring means maintaining the by-pass open in the neutral position of the piston but closing the by-pass whenever the piston is moved in either direction from that position.

2. In a controllable pitch propeller, a pitch adjustable blade, simultaneously and oppositely acting blade adjusting power mechanisms and means turned by said mechanisms to change the pitch of the blade, all rotatable with the propeller, a fluid operated actuating mechanism including servomotors respectively operable to cause operation of the blade adjusting means, said actuating mechanism including a double ended piston and cooperating cylinder means forming pressure chambers connected with respective servomotors, said piston being movable manually in opposite directions from a neutral position, spring means arranged to return the piston to neutral position after movement in either direction, and a by-pass valve system in a through passage of the piston capable of communicating said pressure chambers through the piston, said system including spring means maintaining the through passage open in the neutral position of the piston but closing the passage whenever the piston is moved in either direction from that position.

3. In a rotary assembly, a reversible hydraulic motor, two alternately acting pumps each supplying a respective pressure chamber of the motor, non-rotating means operable selectively to actuate the pumps during rotation of the assembly, a non-return valve between each pump and the motor pressure chamber supplied thereby, plungers supporting respective non-return valves, each plunger being movable by a predetermined back pressure from its connected motor chamber to cause the associated non-return valve to be opened, and cooperating fluid passage means so arranged that the non-return valve connected to one motor chamber can be forced open by exhaust fluid from the motor while the other motor chamber is being supplied by its pump.

4. In a rotary assembly, a reversible hydraulic motor, two alternately acting pumps supplying respective oppositely acting pressure chambers of the motor, means selectively to actuate the pumps during and as a result of rotation of the assembly, a non-return valve mechanism between each pump and the motor pressure chamber supplied thereby, each valve mechanism including a plug sealing an axial bore of a tubular piston capable of operation by a predetermined back pressure from its connected motor chamber, there being cooperating mechanically acting means to cause the associated non-return valve plug then to be moved to open position; and cooperating fluid passage means so arranged that the non-return valve mechanism connected to one motor pressure chamber can operate as an exhaust valve for that chamber while the other motor chamber is being supplied by its pump.

5. In a rotary assembly, means rotatable with the assembly and forming a hydraulic fluid supply chamber, a pump including a pressure chamber and a reciprocable piston therein operable to open the pump pressure chamber to the supply chamber on each return stroke of the piston, means to reciprocate the piston during and as a result of rotation of the assembly, a motor having a fluid connection with the pump pressure chamber, said connection including a non-return valve, and a vacuum breaker valve in the piston between the pump pressure chamber and the supply chamber and operating oppositely of the non-return valve on the return stroke of the piston.

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