United States Patent

Pedersen

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[54] CONTROLLABLE PITCH PROPELLER

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[21] Appl. No.: **11,686**

[52] **U.S. Cl.** **416/157**

[51] **Int. Cl.** **B63h 3/08**

[58] **Field of Search** 416/157, 156; 91/420, 378, 91/422

[56] **References Cited**

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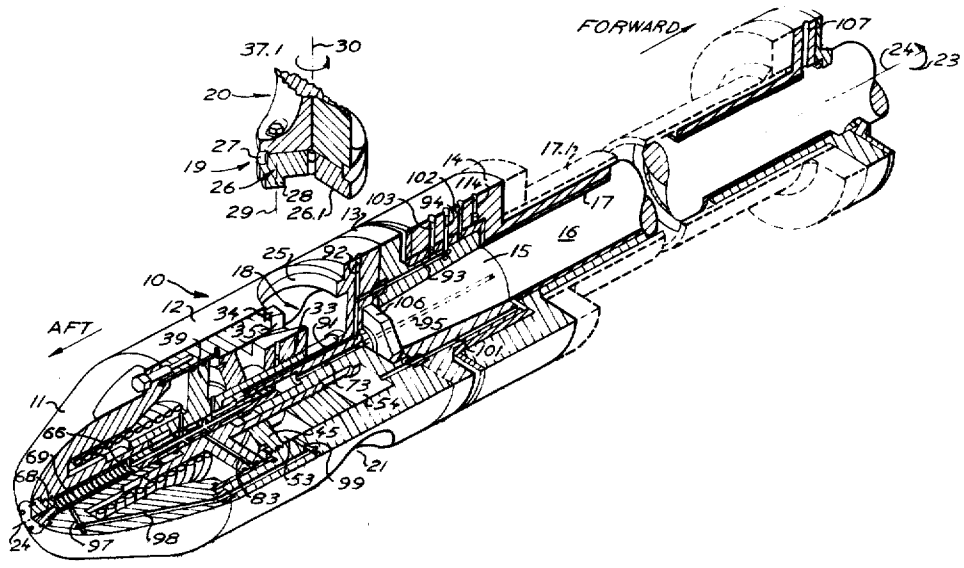
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[57] **ABSTRACT**

Hydraulically activated controllable pitch propeller mechanism adapted to be secured to a tailshaft of a vessel without material modification of the shaft. Can be used to convert existing installations to controllable pitch, and in new installations. Propeller hub has attachment flange and rotary seal, internal hollow shaft has a piston and a crosshead, axial motion of the shaft effecting alteration of pitch of propeller blades. Hydraulic fluid introduced into the hollow shaft through the seal moves a pilot spool within the shaft movement being responsive to difference in pressure upon opposite ends of the spool. Pilot spool acts as a valve, motion admitting hydraulic fluid to move the piston so altering the pitch. Full hydraulic pressure within the hub only during pitch alteration. The rotary seal eliminates hollow tailshaft used on many prior art constructions.

5 Claims, 14 Drawing Figures



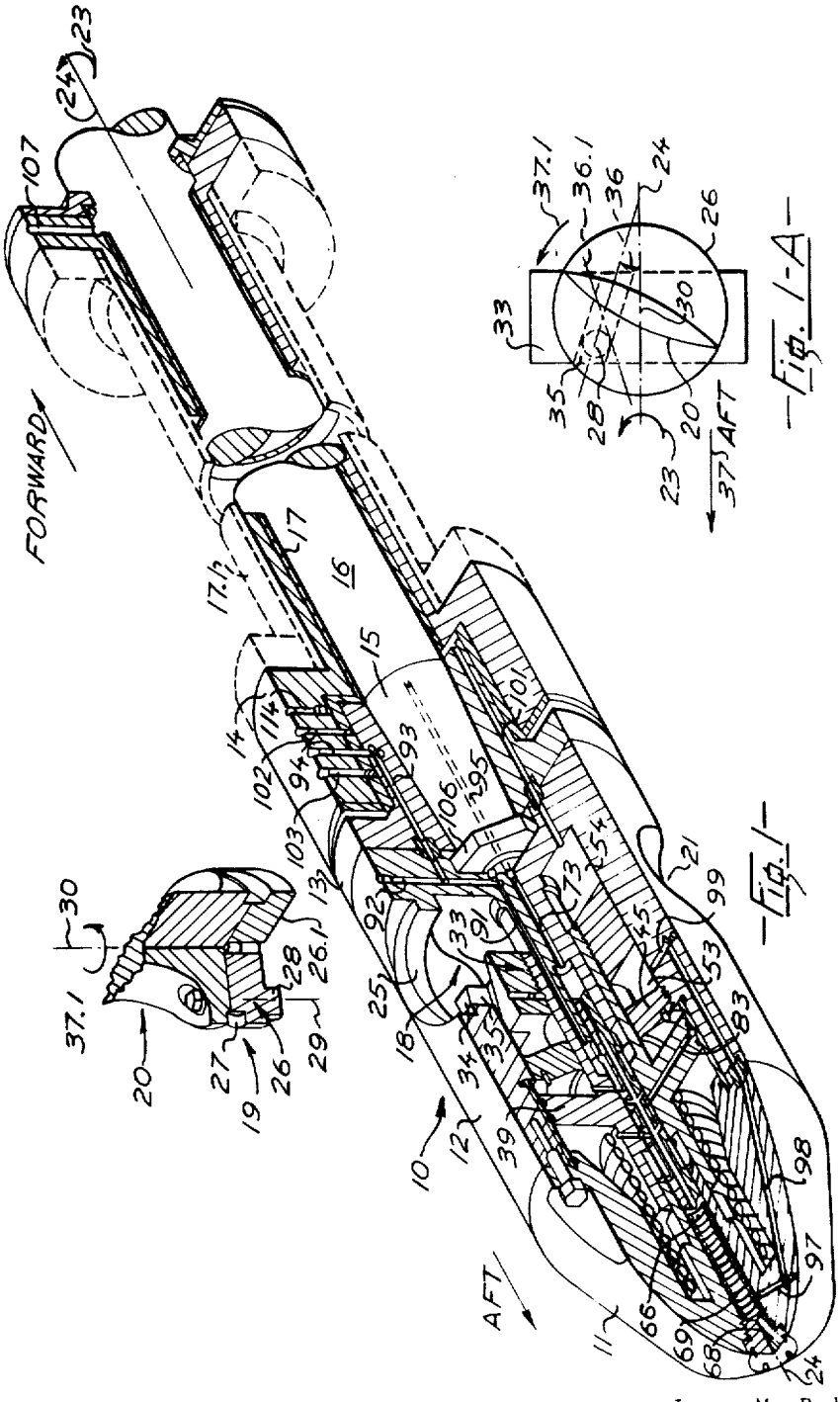


Fig. 1

Fig. 1-A

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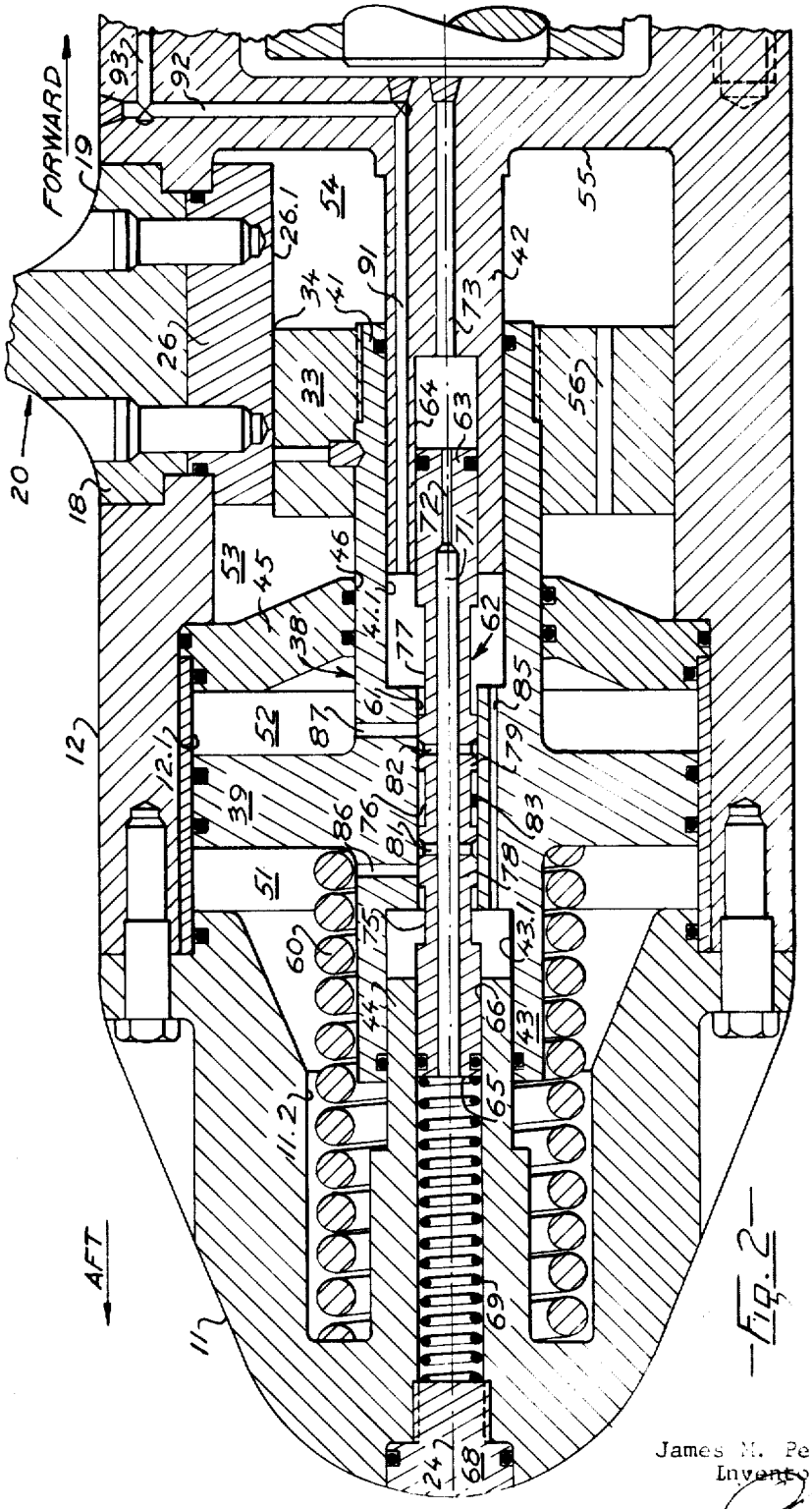


FIG. 2

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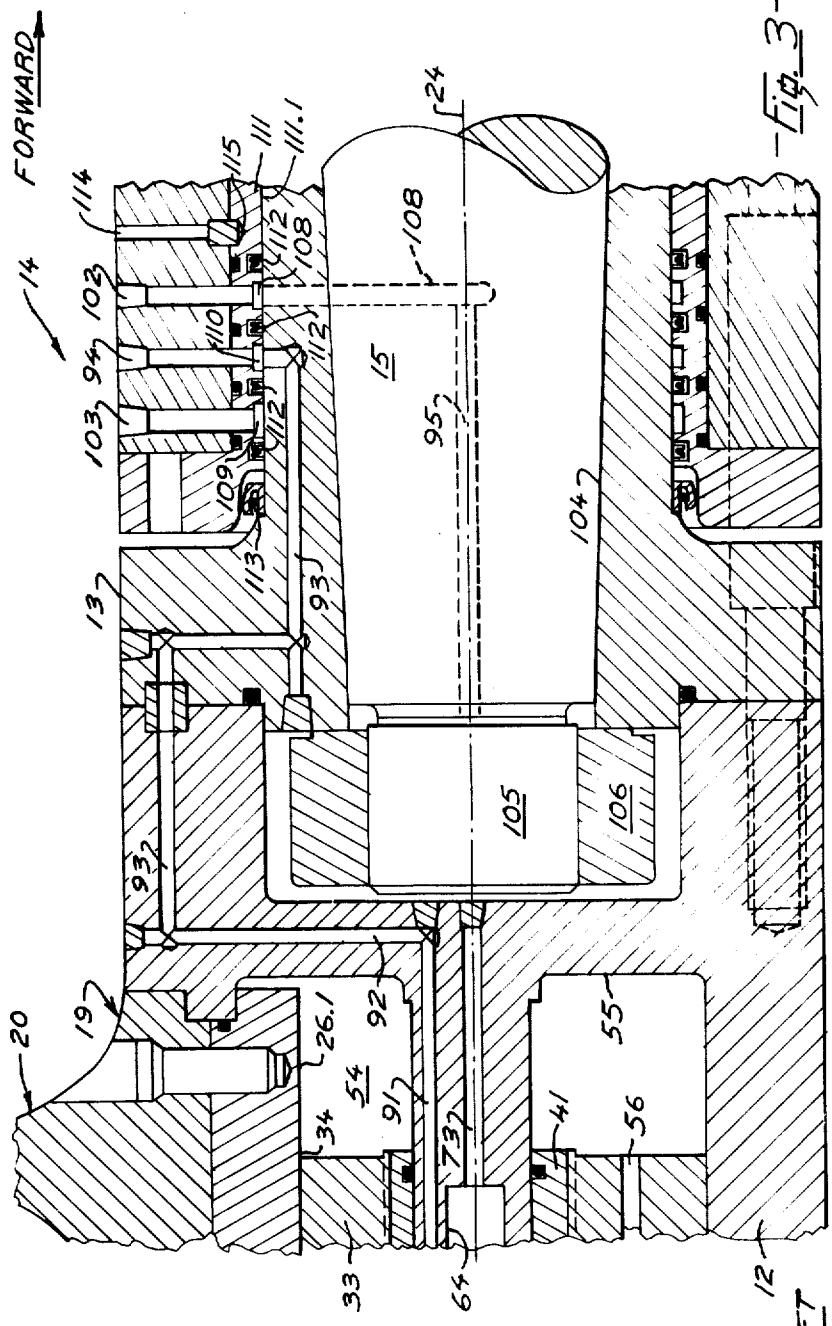


Fig. 3

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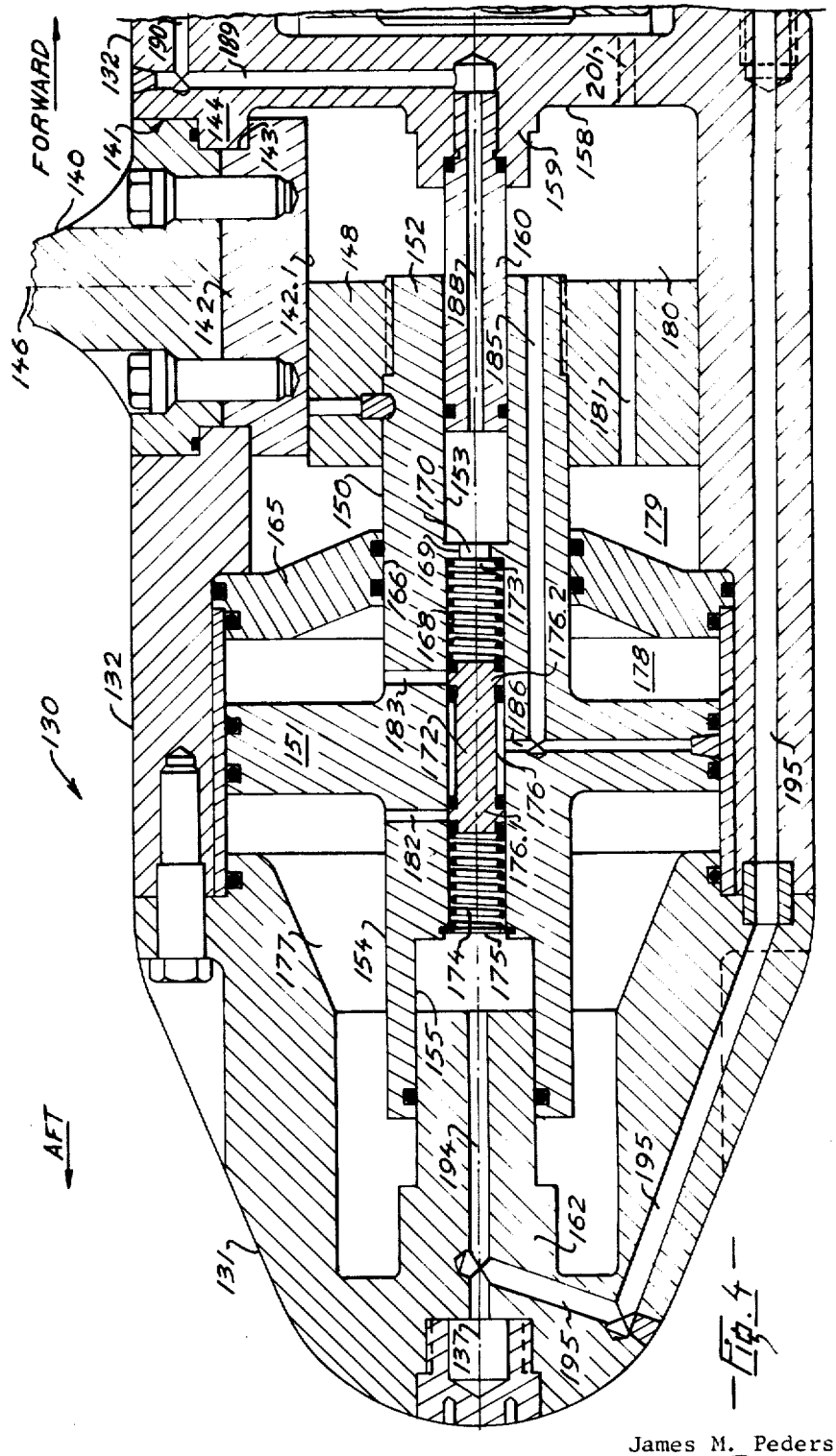


Fig. 4

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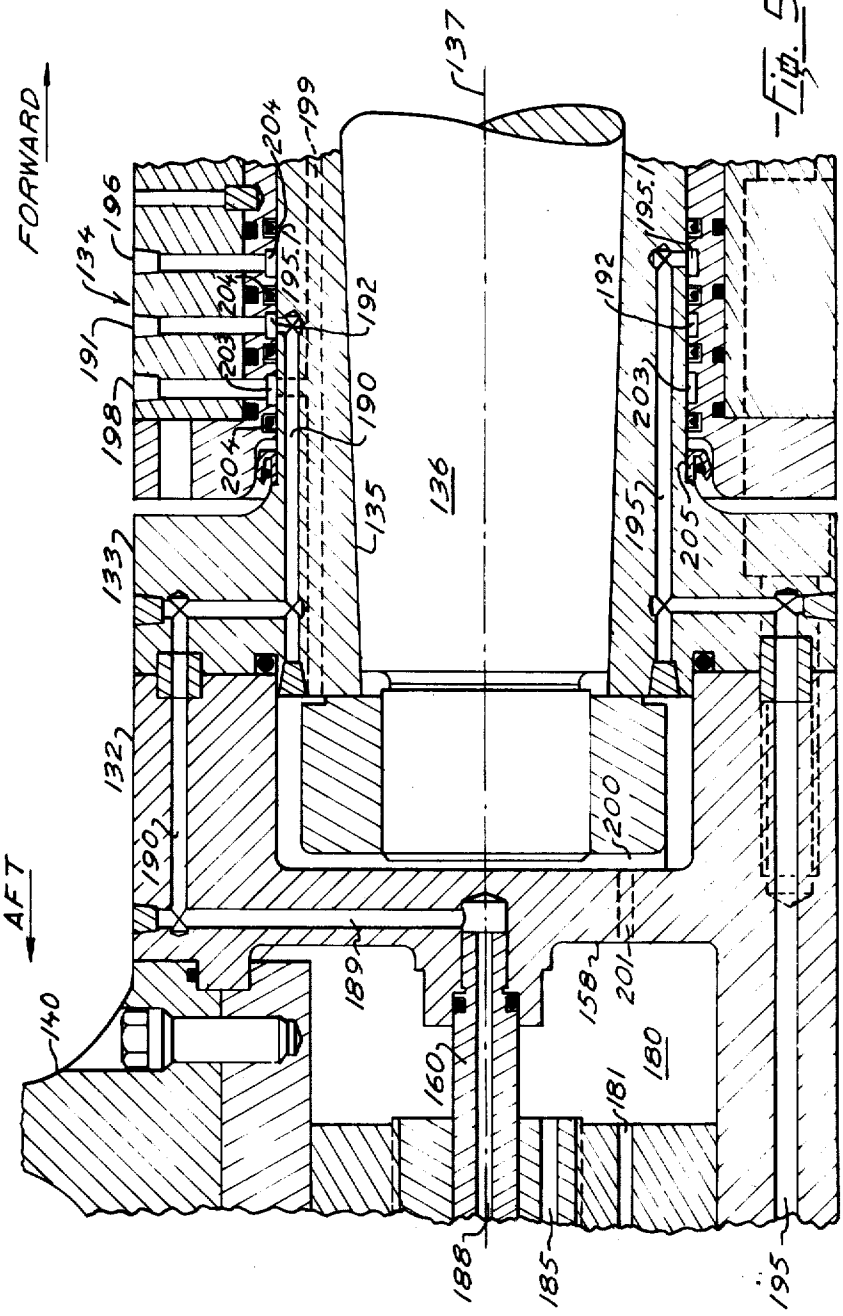


Fig. 5

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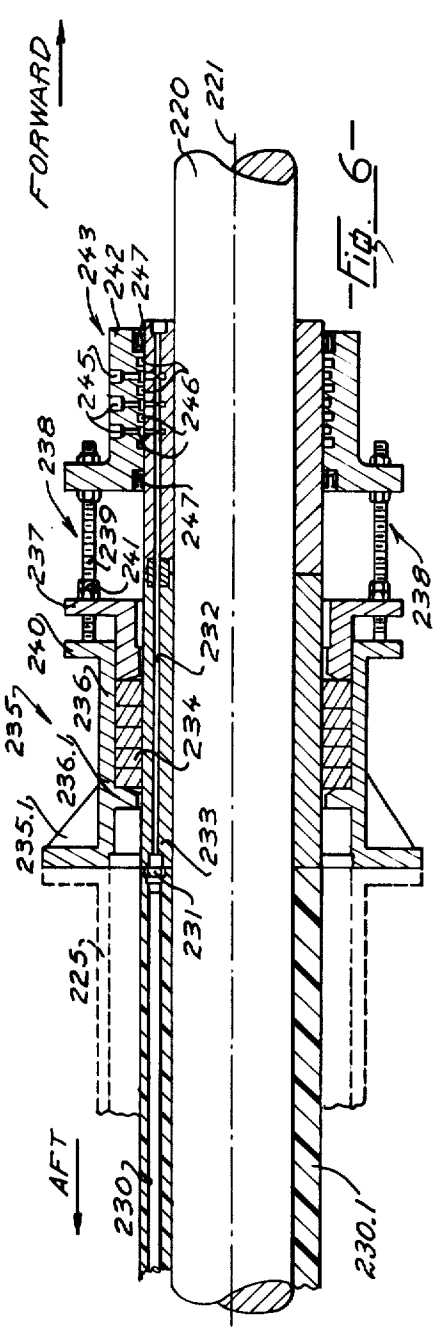


Fig. 6

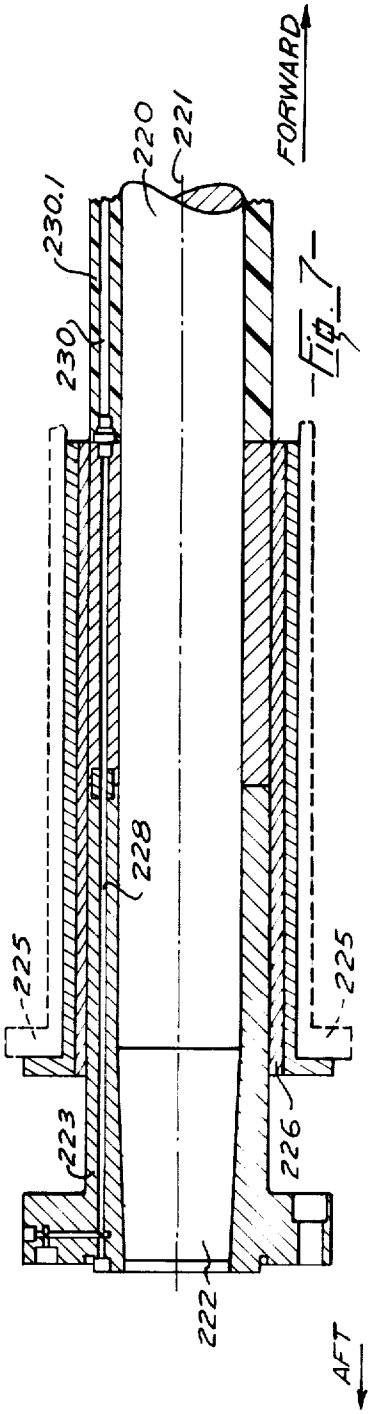
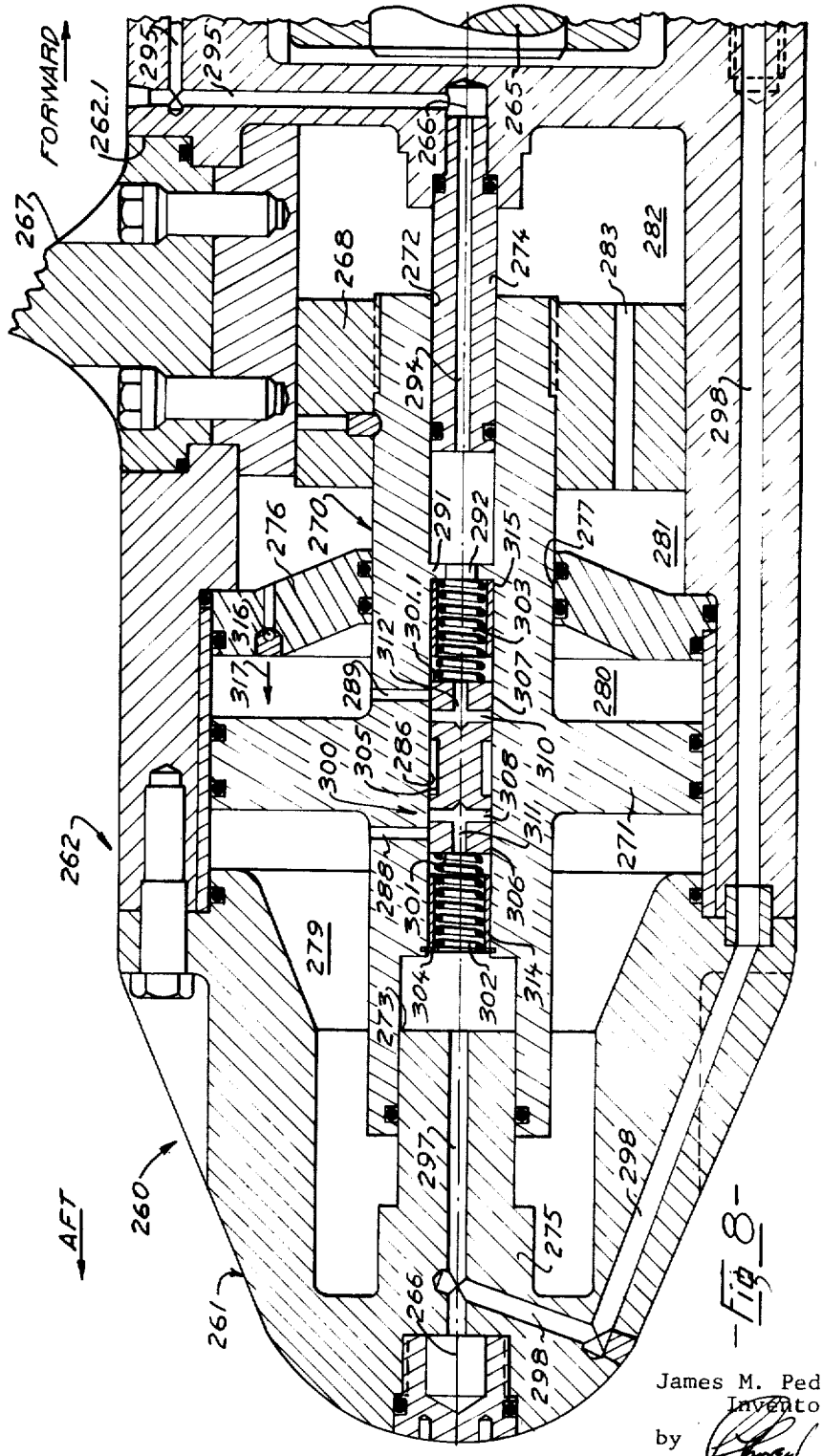


Fig. 7

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-Fig. 8-

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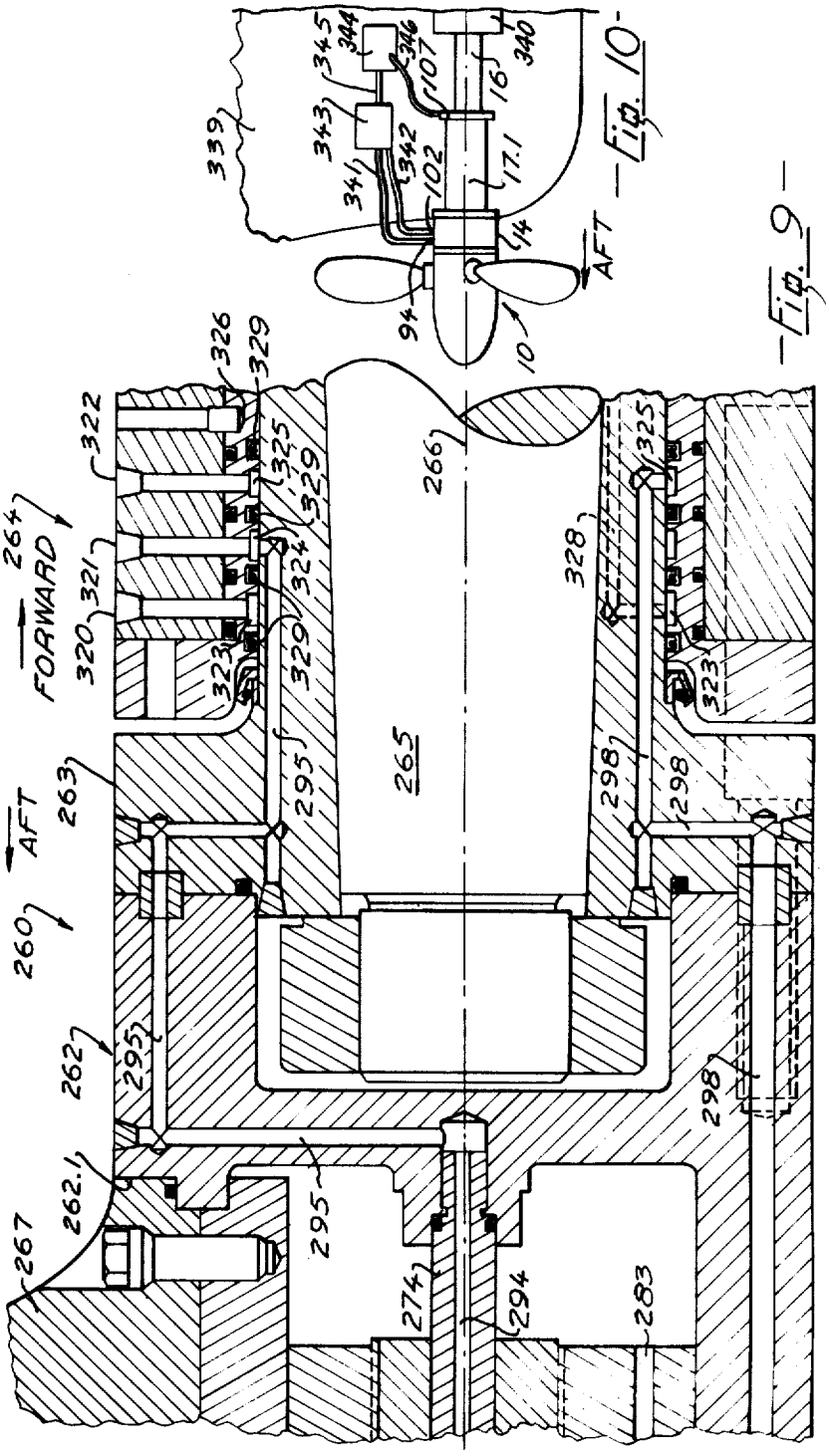


Fig. 9

Fig. 10

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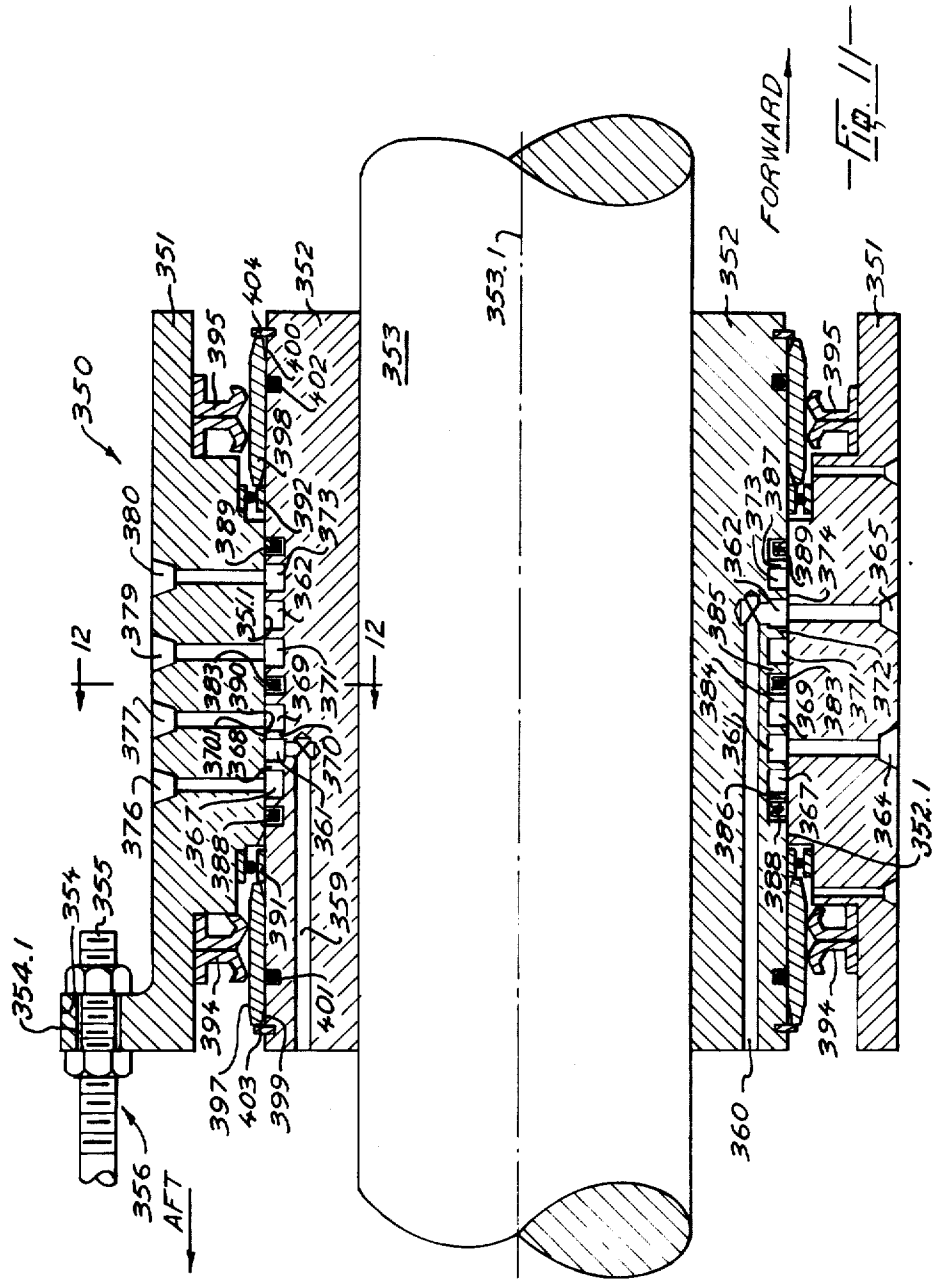


FIG. 11

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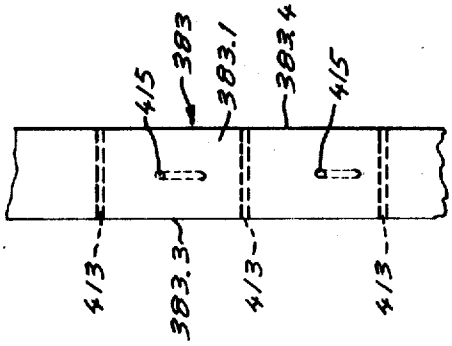


Fig. 13

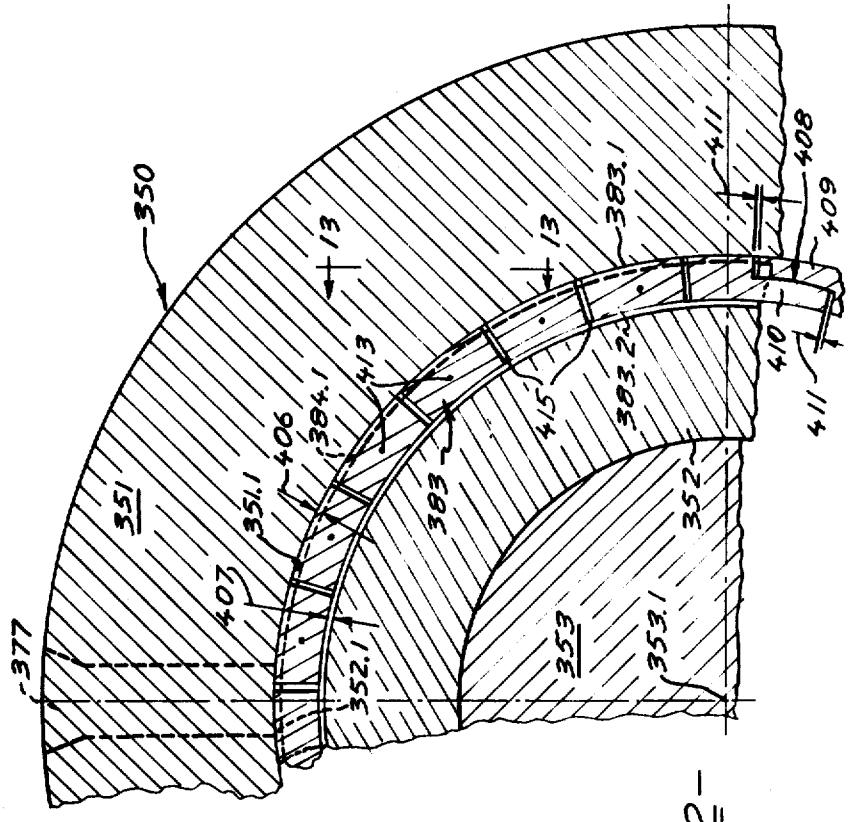


Fig. 12

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CONTROLLABLE PITCH PROPELLER

BACKGROUND OF THE INVENTION

1. Field of the Invention

The invention relates to a controllable pitch propeller mechanism adapted to be secured to a tailshaft of a vessel without material modification of the tailshaft.

By controllable pitch propeller herein is meant, a propeller, blades of which are rotatable about a longitudinal blade axis so as to alter pitch. The invention relates to a mechanism to effect such rotation changing the pitch, the blades themselves are according to prior art blades of this kind.

2. Prior Art

Advantages of controllable pitch propellers in certain vessels and services are well known, and are not further discussed herein.

Commonly propellers of this kind require a hollow tailshaft for supply of hydraulic fluid at a high pressure for activating pitch change mechanism. Moreover, this high pressure — say 1,000 psi — is required to be maintained whether or not pitch is being changed. To install a propeller of this type in an existing vessel thus requires, either installation of a hollow drive shaft, or boring out an existing shaft — both of which procedures are expensive and, furthermore, only highly specialized plants have boring mills capable of accomplishing work of this nature.

The present invention provides controllable pitch propeller mechanism including a hub capable of assembly to existing tailshafts without material alteration of the latter, thus the invention eliminates the costly hollow shaft both in new installations and in installations to existing vessels since a hollow drive shaft is not used. Further, the invention requires the high pressure aforesaid only while effecting change of pitch.

Typical installations contemplated are for tow and trawler vessels of 150 to 1,500 HP, wherein material cost saving is effected relative to cost of existing prior art controllable pitch propellers known to the present inventor.

Additionally, the mechanism is simple rugged and relatively easy to fabricate.

SUMMARY OF THE INVENTION

The pitch mechanism itself is built within a hub having a flange, and a portion of the flange is fitted in a rotary seal assembly secured to the vessel, the flange having a tapered internal bore to fit a corresponding taper on the tailshaft of the vessel.

The hub has an internal hollow shaft to which is secured a piston, and a crosshead having a groove engaging a boss of a root of the propeller blade. The groove is at an angle so that the motion of translation of the crosshead, the boss being engaged in the groove, rotates the blade root changing pitch of the propeller.

In all embodiments exemplified, a means slidable within the hollow shaft, namely a spool, is held stationary by balanced forces applied to its opposite ends. In a balanced position, hydraulic fluid fills cylinder spaces on each side of the piston, so effectively locking it and accordingly the crosshead, so that the propeller is held at constant pitch. Means are provided to vary the force applied to one end of the spool causing it to move axially, which axial motion opens a passage permitting hydraulic fluid under pressure to be applied to one side of the piston so that the piston, the hollow shaft, and the crosshead, are urged to move axially so altering pitch of the propeller.

The shaft follows the spool so that the spool regains a balanced position cutting off fluid flow to and from the cylinder spaces and effectively locking the blade at a fixed pitch.

In one embodiment the spool has a metering bore, change of rate of flow of fluid through the metering bore upsets the balance causing the spool to move axially. In this embodiment there is separate supply of hydraulic fluid under actuation pressure for admission to a cylinder space to move the piston as has been explained. In this embodiment rate of flow through the metering bore is constant for a constant pitch, change of pitch being effected as explained by a change of flow rate through the metering bore.

In a further embodiment, equal forces are applied to opposite ends of the spool by opposed compression spring means which, in a balanced position, exert equal forces on opposite ends of the spool so that it remains stationary with respect to the bore. Axial motion aft is effected by hydraulic fluid under pressure acting upon one end of the spool, the motion being resisted by one of the compression springs and continuing for a particular distance until balance is again effected. Forward motion is correspondingly effected by directing the fluid to act upon the opposite end of the spool.

In the further embodiment fluid under pressure admitted to the hollow shaft is selectively directed to one of the opposite ends of the spool, so providing an unbalanced force moving the spool axially as explained. This motion as before admits fluid under pressure to a cylinder space moving the piston and altering pitch. However, in this embodiment the fluid causing the unbalance also moves the piston.

A third embodiment differs from the further embodiment in arrangement of passages for the hydraulic fluid.

One embodiment of a rotary seal assembly cooperates with the tailshaft in a position aft of an aft bearing. Ports in the rotary seal assembly cooperate with grooves in the tailshaft, the grooves communicating with passages in the propeller hub. Seals provided between the annular grooves reduce leakage of fluid from the grooves, thus permitting fluid to enter and leave the hub as the shaft rotates. Fluid leaking from the grooves is passed to the aft bearing, providing bearing lubrication, after which the fluid is scavenged to a fluid supply tank.

An alternative embodiment of a rotary seal assembly is mounted forward of the aft bearing, which bearing in the alternative embodiment is water lubricated. A common stuffing box is mounted between the aft bearing and the rotary seal resulting in a substantially dry tailshaft. The alternative rotary seal is a floating assembly, adjusting means cooperating with radial slots permitting movement of the seal relative to the tailshaft, thus accommodating small eccentricities of the tailshaft and controlling fluid leakage.

A detail description following related to drawings gives exemplification of embodiments of the invention which, however, is capable of expression in structure other than that particularly described and illustrated.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is a fragmented, partly sectioned and partly exploded perspective view of a propeller hub according to the invention,

FIG. 1-A is a simplified diagram of a pitch changing mechanism viewed inwards along an axis of rotation of a propeller blade,

FIG. 2 is a fragmented simplified section on an aft end of the propeller hub in a plane containing axes of rotation of the hub and a propeller blade,

FIG. 3 is a simplified fragmented section continued forward of the section as shown in FIG. 2,

FIG. 4 is a simplified fragmented section on an aft end of an alternative embodiment of a propeller hub, the section being in a plane containing the axes of rotation of the propeller blade and the hub,

FIG. 5 is a simplified fragmented section continued forward of the section as seen in FIG. 4,

FIG. 6 is a fragmented section on a centerline of a forward portion of a tailshaft showing a floating rotary seal assembly,

FIG. 7 is a fragmented section on a centerline of a tailshaft continued aft of the portion as shown in FIG. 6,

FIG. 8 is a simplified fragmented section on an aft end of a further alternative hub embodiment the section being an a plane containing axes of rotation of a propeller blade and the hub,

FIG. 9 is a simplified fragmented section continued forward of the section as seen in FIG. 8,

FIG. 10 is a fragmented simplified diagram of a portion of a stern of a vessel showing hydraulic components associated with the propeller,

FIG. 11 is a section of an alternative floating rotary seal assembly to be used in combination with a stuffing box, the section being in a plane containing the axis of rotation of the tailshaft,

FIG. 12 is a fragmented section of a portion of the tailshaft and rotary seal assembly as seen from 12—12 of FIG. 11,

FIG. 13 is a fragmented detail view of a portion of a piston ring type seal, as seen from 13—13 of FIG. 12, portions of the tailshaft and seal assembly being omitted.

DETAILED DESCRIPTION

FIGS. 1, 1-A

In FIG. 1, a controllable pitch propeller mechanism according to the invention includes a generally hollow hub **10** having an aft portion **11** secured by bolt means to a hollowed mid-portion **12** having an inner wall **12.1**. A propeller flange **13** is secured to a forward end of the hub and a forward portion of the flange fits within a rotary seal assembly **14** secured to a vessel, shown in FIG. **10** only. The flange **13** has a tapered bore complementary to a taper **15** on an aft end of a tailshaft **16**, the taper being adjacent an aft bearing **17** of a stern tube **17.1** of the vessel, the stern tube being shown in broken outline and being secured to the assembly **14**.

An opening **18** in the mid-portion **12** is adapted to receive a root **19** of a propeller blade fragment **20**. A similar opening **21** is shown for a second propeller blade (not shown) and, for a three-bladed propeller, a third propeller blade (not shown) is provided, the blades being spaced 120°. The propeller hub has an axis of rotation **24** coaxial with the bearing **17**. The propeller as described is adapted for left hand rotation (i.e., counterclockwise rotation when viewed from the stern as shown by an arrow **23**) the blade being shown in a forward pitch position.

The opening **18** has an annular shoulder **25**, the root **19** of the propeller blade being secured to a blade connection plate **26**, the plate **26** and the root **19** defining an annular groove **27** which accepts the annular shoulder **25**. A boss **28** extends from the blade connection plate **26**, the boss having a longitudinal axis **29** parallel to a longitudinal axis **30** of the propeller blade and perpendicular to the plate **26**. Bearing means provided by the annular shoulder and the groove **27** permit rotation of the blade about the axis **30** and provide restraint from force on the blade as the hub rotates about the axis **24**. The axis **30** intersects the axis **24**, as do axes of the second and third propeller blades not shown.

A crosshead **33** is adapted for longitudinal translation along the axis **24** and has a flat **34** parallel to and continuous with inner face **26.1** of the plate **26**. The flat **34** is perpendicular to the axis **30** and has a groove **35** at an angle to the axis **24** the angle, designated **36** in FIG. 1-A only, hereinafter groove angle. The boss **28** is in sliding engagement with the groove **35** forming means, responsive to crosshead movement, to change blade pitch. Since the boss **28** is constrained within the groove, longitudinal translation of the crosshead along the axis **24** causes the boss **28** to slide along the groove **35**, thus moving the boss laterally, rotating the blade about the axis **30** and changing the pitch of the propeller blade. Bearing means (not shown) can be provided in the groove, the means being a shoe journalled on the boss fitting in the groove, or needle bearing means reducing wear and friction between the boss **28** and the groove.

With reference to FIG. 1-A, aft is shown in direction by an arrow **37** and corresponding rotation of the propeller blade **20** about the axis **30** is shown in direction by an arrow **37.1**. The boss **28** is shown in the groove **35**, the groove being inclined, as aforesaid, to the axis **24** at the angle **36**. Rotation of the propeller blade about the axis **30** is dependent on the groove angle and longitudinal translation of the crosshead. For a given propeller hub, the groove angle is dependent on pitch variation requirements.

To simplify manufacturing and stocking of the propeller hubs, crossheads differing only in the groove angle are used, thus enabling one hub to serve several pitch variation requirements. The groove angle for use on a propeller adapted for right-hand rotation is a mirror image of the groove angle used for left-hand rotation. Thus, a centerline of a groove for use on a propeller adapted for right-hand rotation is shown in broken outline at **36.1**.

All of the propeller blades are similar and each blade has a corresponding groove in the crosshead, the groove angles being equal. Thus longitudinal translation of the crosshead **33** produces equal rotation of each propeller blade, which results in equal change of pitch of each blade. Longitudinal translation of the crosshead is effected by hydraulic means later described.

FIG. 2 (with references to FIG. 1)

The crosshead 33 is secured on a shaft 38 which has a piston 39 at an approximate midpoint on the shaft, the piston 39, the shaft 38, and crosshead 33, moving as one unit. The shaft 38 has a forward end portion 41 having a forward bore 41.1 that fits over a spigot 42 of the portion 12, and has an aft end portion 43 having a bore 43.1 that fits over a spigot 44 of the aft portion 11. The end 41 passes through a central bore 46 of a fixed dividing wall 45 which is secured to the inner wall 12.1.

The propeller hub is thus divided internally into several spaces, the wall 45 dividing the hub into an aft cylinder and a forward cylinder. The piston 39 is in the aft cylinder forming an aft cylinder space 51 aft of the piston and a forward cylinder space 52 forward of the piston. The crosshead 33 forms an aft crosshead space 53 aft of the crosshead and a forward crosshead space 54 forward of the crosshead, a passage 56 being provided through the crosshead permitting fluid to flow through the crosshead, so as to prevent the crosshead behaving as a piston. The space 51 is defined in part by an aft face of the piston 39, a portion of the inner wall 12.1 and a cylindrical passage 11.2. The space 52 is defined in part by a forward face of the piston 39 and an aft face of the annular wall 45. The space 53 is defined in part by a forward face of the annular wall 45 and an aft face of the crosshead 33. The space 54 is defined in part by a forward face of the crosshead 33 and a forward wall 55 of the portion 12.

The space 51 has a come home compression spring 60 which tends to urge the piston 39 to a forward position. The spring 60 encircles the spigot 44 and the portion 43 and becomes operative to attain maximum forward pitch should hydraulic pressure fail.

When the propeller hub rotates at slow speed, this spring has strength sufficient to return the pitch of the propeller blades to full ahead position without hydraulic means, as is later explained.

The shaft 38 has a central bore 61 containing a pilot spool 62, slidable in the bore, having a forward end portion 63 which is a sliding fit in a bore 64 of the spigot 42, and an aft end portion 65 which is a sliding fit in a bore 66 of the spigot 44.

The bore 66 is sealed at an aft end by a screw 68 and has a spring 69, hereinafter the spool spring compressed between the screw 68 and the spool. This spring tends to urge the spool to a forward position within the bore 64. The spool 62 has a central bore 71 which narrows to a metering bore 72 at the forward end of the spool. The bores 71 and 72 thus communicate with the bores 66 and 64 at opposite ends of the spool. A passage 73 enters the bore 64 supplying fluid at control pressure from a source later described.

The spool 62 has an outer cylindrical surface that is relieved at 75, 76, and 77, producing two lands 78 and 79 as shown. The lands are a sliding fit in the bore 61, the land 78 having a radial bore 81 and the land 79 having a radial bore 82. The two radial bores connect the bore 71 with the bore 61 and can be registered with other passages to be described. A pin 83 which is best seen in FIG. 1 protrudes inwards into the bore 61 between the lands 78 and 79; only longitudinal location of the pin 83 is shown simplified in FIG. 2. Motion of the spool relative to the bore 61 is thus limited by spacing of the lands 78 and 79, the lands interfering with the pin 83. The pin is particularly adapted for rapid pitch changes or if there is loss of hydraulic pressure as is later explained being a means to locate the spool relative to passages in the shaft 38.

A longitudinal passage 85 connects the bore 43.1 with the bore 41.1. A radial passage 86 connects the bore 61 with the space 51 and a radial passage 87 connects the bore 61 with the space 52, the radial passages being longitudinally spaced so that one radial passage (86 or 87) is in register with one radial bore, i.e., 81 or 82, in the spool when a land exposes a remaining radial passage.

A longitudinal passage 91 carries fluid at actuation pressure from the propeller flange 13 (FIG. 1) into the bore 41.1, the fluid passing through the passage 85 into the bore 43.1. Thus fluid at actuating pressure appears on each end of the central bore 61.

Undesignated seals are used throughout the propeller hub, reducing undesirable fluid flow particularly past the pilot spool end portions within the bores 64 and 66, past the piston 39 and the inner wall 12.1, and between the bore 46 of the wall 45 and the shaft 38. These seals are O-ring seals that permit sliding movement.

Rigid seals are used between the portions 11, 12, and the flange 13. These seals are not required to withstand much movement between parts to be sealed.

FIG. 1, passages (with references to FIGS. 2 and 3)

The passage 91 through a radial passage 92 communicates with a passage 93 to the rotary seal assembly 14 and a pressure port 94 adjacent the propeller flange 13. (Routing of the passage 93 is shown modified in FIG. 3).

A scavenge passage 97 leads from an aft end of the bore 66 returning fluid to the space 53 via passages 98 and 99. Fluid can pass through the passage 56 (FIG. 2.) from the space 53 into the space 54, from where a passage 101 leads to an annular groove (not shown) associated with a port 103 in the rotary seal assembly 14. The port 103 receives fluid for tailshaft lubrication purposes and fluid at lubrication pressure from the port 103 mixes in the annular groove to be described with fluid scavenged from the hub via the passage 101, the mixture passing forward lubricating the tailshaft 16 and returning to a fluid tank via a passage 107 at a forward end of the aft bearing 17.

A pressure port 102 in the rotary seal assembly 14 supplies fluid at control pressure to the passage 73 via a groove (FIG. 3 only) and passages shown in broken outline and designated 95. There is controlled leakage of fluid from the port 94 (later explained), and if there is sufficient leakage to provide shaft lubrication, the port 103 becomes redundant and can be ommited.

FIG. 3 (with references to FIGS. 2 and 10)

The propeller flange 13 has a tapered bore 104 which complements the taper 15 on the aft end of the tailshaft 16. The shaft has a threaded aft end 105 which accepts a propeller hub retaining nut 106 to secure the propeller hub assembly on the shaft 16. Suitable keys and key ways (not shown) are fitted to provide drive between the shaft and the propeller hub.

The ports 94 and 102 in the rotary seal assembly 14 communicate, via annular grooves as aforesaid, with passages in the propeller flange, the passage 93 being shown clearly only in FIG. 3. Each of the ports 102, 103, and 94, communicates with corresponding annu-

lar grooves **108**, **109**, **110** respectively in a bore **111.1** of a rotary seal inner sleeve **111**, and fluid is transmitted into the propeller flange then along the shaft as the flange rotates. A passage **95** (shown in broken outline) connects the groove **108** (shown partly in broken outline) with the passage **73**, permitting fluid to pass from the port **102** to the passage **73**. (Passage routing is not shown completely).

Four seals **112** are provided in the sleeve **111** and one seal between each annular groove to control leakage between fluids at different pressures in adjacent grooves. Total fluid leakage during pitch change in the order of 1 to 1 ½ gallons per minute from the port **94** is provided for. With frequent pitch changes this leakage may be sufficient to lubricate the aft bearing, making the port **103** supplying lubrication fluid redundant.

These four seals can be piston ring type seals, each being a spring steel ring having a gap, permitting insertion of the seal and providing a spring effect similar to piston rings. The seal can be made to grip either inwards onto the flange **13** or outwards onto the bore **111.1**. In either case above, fine grooves are provided on oppositely aligned radial faces of each seal to permit fluid to penetrate on either side of the seal to lubricate surfaces, reducing friction between the seals and sleeve **111** increasing life of the seals.

A low pressure seal **113** is provided between the propeller flange and the rotary seal reducing ingress of water into the rotary seal and reducing loss of oil outwards from the seal. A passage **114** in the rotary seal permits insertion of a screw plug **115**, the plug preventing rotation of the sleeve **111** relative to the rotary seal **14**.

Two hydraulic fluid lines from a control unit (FIG. **10**) are connected to the pressure ports **94** and **102**, valves for each line being controlled by the control unit. The pressure port **102** via the control unit is supplied with hydraulic fluid from a variable volume pump. This port supplies fluid at control pressure, volume flow being selected by an operator for a desired pitch. The passage **73** (FIG. **2**) receives fluid at a controlled volume flow and controlled pressure, the flow determining position of the pilot spool which, as will be described, determines piston position. The pressure port **94** is supplied with hydraulic fluid at actuation pressure from a pressure compensated variable volume pump and is used to change the propeller pitch. Compensation results in oil being supplied to the port **94** at a constant pressure and at a rate demanded, for example, 0 to 3 gallons per minute. The control unit is adapted to supply fluid continuously at control pressure at a volume flow rate necessary to maintain a particular desired pitch. When changing pitch by changing the volume flow rate into the port **102**, interconnection between the valves permits flow of fluid at actuating pressure into the port **94** for a period of pitch change only.

OPERATION

FIG. **2** (with references to FIGS. **1** and **3**)

Fluid at control pressure enters the bore **64** from the passage **73**, leaving the bore **64** through the metering bore **72** of the pilot spool **62**, the bore **72** restricting volume flow rate of fluid into the central bore **71**. Restriction of flow in the bore **72** produces a pressure drop across the bore **72**, which pressure drop results in a force on the forward end of the spool tending to move the spool aft. Movement of the spool aft is counteracted by a force on the aft end of the spool from compression of the spring **69**. An equilibrium position of the spool is attained in which position the force aft on the forward end of the control spool, due to fluid, is balanced by the force due to the compression spring **69**. In the equilibrium position the land **79** closes the passage **87** and the land **78** closes the passage **86**. Hydraulic fluid entering the bore **71** flows into the bore **66** and is returned to the space **53** via the passages **97** and **98**, (FIG. **1** only), the passage **101** returning scavenged fluid from the space **54** and stern tube. In FIG. **3**, in the equilibrium position, (i.e., constant pitch), while the port **102** receives fluid at a constant volume flow at control pressure the port **94** does not receive fluid at actuation pressure.

To change pitch of the propeller blades, e.g., to make the pitch finer, volume flow rate at control pressure is increased at the port **102**. An increase in control pressure volume flow rate increases force on the forward end of the spool **62** resulting in movement of the pilot spool aft relative to the shaft **38** until the pin **83** interfers with the land **79**. The lands **78** and **79** are thus moved aft and expose the passages **86** and **87** to the bores **61** and **41.1** respectively.

Fluid at actuation pressure concurrently enters the rotary seal through the pressure port **94** and flows along the passages **93**, **92**, and **91**, into the bore **41.1**. The passage **85**, connecting the bore **41.1** with the bore **43.1**, permits fluid at actuation pressure to enter the bore **43.1**, thus providing fluid at actuation pressure on each end of the central bore **61**. Fluid within the bore **41.1** enters the passage **87** and flows into the space **52**, causing an increase in pressure on the forward face of the piston **39** producing a resultant force moving the piston aft to a new position. The resultant force is greater than maximum forward force from the spring **60**.

Concurrently with opening of the passage **87**, the bore **81** comes into register with the passage **86**, permitting exhausting of fluid from the space **51** as the piston **39** is moved aft. Fluid from the space **51** flows through the passage **86**, through the bore **81** and into the bore **71**, thus mixing with the fluid from the bore **66**.

When the volume flow at control pressure ceases to increase, the spool stops moving aft and the shaft **38** assumes the equilibrium position relative to the spool, so that the passages **86** and **87** are closed by the lands **78** and **79** respectively. The spool effectively locks the piston **39** hydraulically in position, and pitch creep is negligible. Supply of high pressure fluid at the port **94** is stopped, being required only for pitch change operations, interconnection as aforesaid between valves in the control unit providing this requirement. The piston **39** maintains the new position until there is a change in volume flow into the port **102**.

It is seen that the piston **39** follows longitudinal movement of the control spool. The spool is thus a master and the piston **39** is a slave.

With reference to FIG. **1-A**, movement aft of the shaft **38** shown by the arrow **37** results in movement of

the crosshead 33 aft, which movement produces rotation of the propeller blade about the axis 30 in a direction shown by the arrow 37.1, the boss 38 being swung to starboard being restrained by the groove angle. Thus the pitch is made finer.

Movement of the crosshead to a maximum aft position results in reverse pitch, the pitch having passed through a neutral position.

A decrease in fluid flow at control pressure results in a decrease in force at the forward end of the spool producing a resultant forward force from the spring 69 urging the spool forward until the land 78 interfers with the pin 83. The land 78 thus moves forward exposing the passage 86 to fluid at actuation pressure within the bore 43.1, which fluid enters the space 51 through the passage 86. Pressure on the aft face of the piston 39 is increased, thus producing a resultant force being augmented by relatively small force from the spring 60. The bore 82 is brought into register with the passage 87 permitting fluid exhausted from the space 52 to pass into the bore 71 thus mixing with fluid from the bore 64. The piston 39 moves forward until fluid flow at control pressure becomes constant and the equilibrium position is attained in which the lands 78 and 79 close the passages 86 and 87 as before explained.

Movement of the piston 39 forward moves the crosshead 33 forward, this movement swings the boss 28 to port because the groove is at an angle, 36 FIG. 1-A, adapted to rotate the blade in a direction opposite to that of the arrow 37.1. The pitch is thus made coarser.

It is seen that, at the equilibrium position of the spool, there is a balance of opposite forces from the spring 69 and fluid pressure forces on the spool due to the pressure drop across the bore 72. Hydraulic pressure failure results in the spool 62 being urged forward by an unbalanced force from the spring 69. The land 78 interfers with the pin 83 preventing further movement forward of the spool relative to the shaft 38, thus bringing the bore 82 into register with the passage 87, permitting fluid to be exhausted from the space 52 into the bore 71. The land 78 exposes the passage 86, admitting fluid into the space 51 from the bore 43.1 as the piston 39 is urged forward by the spring 60. As the piston 39 is urged forward by the spring 60 the spool 62 is urged forward also since the spring 69 urges the land 78 against the pin 83. The spool 62 thus follows the piston 39 and the propeller thus attains maximum forward pitch. Fluid pressure in passages in the hub is relieved by fluid leaking through seals in the rotary seal, inhibiting formation of hydraulic locks.

If the spring 60 were not fitted, the propeller blades would remain in the pitch at which the propeller was immediately prior to hydraulic pressure failure, or would attain a pitch, possibly undesirable, imposed by forces on the propeller blades. For a propeller of 4 feet in diameter, actuation pressure is about 1,000 psi, control pressure is in a range of 25–200 psi and volume flow rate is about 4 gallons per minute maximum.

In recapitulation of the foregoing, pilot spool 62 FIG. 2 is a means within the bore 61 adapted for slidable forward and aft movement within the bore and relative to the hub. The said means, viz the spool 62, moves responsive to a difference between a force tending to move it forward, and a force tending to move it aft. In FIG. 2 the force tending to move the spool aft is supplied hydraulically as has been explained the force tending to move it forward is that exerted by the spool spring 69, a compression spring. When these forces are unbalanced the spool moves relative to the bore, and relative to the hub, there being no movement relative to the bore when these forces are balanced, i.e., equal.

After the means, the spool 62, has moved as aforesaid through a particular distance, then means again to equalize the forces are provided as has been explained, means responsive to the movement causing the shaft 38, and the piston 39 and the crosshead 33 both of which are secured to the shaft, to move axially relative to the hub through the particular distance — the shaft follows the spool — so that the spool attains a new balanced position stationary relative to the hub, with the spool then having the same position relative to the shaft bore as initially.

FIGS. 4 and 5 (With reference to other figures)

In the FIG. 1 embodiment a continuous flow of fluid is fed to one end of the spool 62 producing a force on spool which is balanced by the spring 69. A hub according to alternative embodiment is supplied with fluid during pitch change only, thus reducing possible fluid loss as will be explained.

The spool 62 of the FIG. 1 embodiment is eliminated and an alternative spool is substituted. The spring 69 and the continuous flow of fluid is replaced by two spool springs — one on either end of the alternative spool.

An alternative embodiment of a controllable pitch propeller hub mechanism is designated 130 in FIG. 4. The hub has an aft portion 131, a midportion 132, and a propeller flange 133 (FIG. 5), secured by bolt means. A rotary seal assembly 134 admits fluid to the propeller hub via the propeller flange by means similar to those described with reference to FIG. 3, thus further detail description is deemed unnecessary. The propeller flange is secured on a taper 135 of a tailshaft 136 having an axis of rotation 137. Aft bearings in a stern tube of the vessel (not shown) are similar to those described with reference to FIG. 1.

A propeller blade root 140 is shown within an opening 141 in the midportion 132. Blade bearing means are similar to those previously described, namely a blade connection plate 142 secured by bolt means to the blade root 140, forming an annular groove 143 which accepts an annular shoulder 144 within the midportion 132. The propeller blade can rotate about an axis 146, rotation being controlled by a boss (not shown) slidably fitted in a groove (not shown) of a crosshead 148. The boss extends from an inner face 142.1 of the blade connection plate and is similar to that as shown in FIG. 1. As previously described, movement of the crosshead 148 aft results in a finer pitch and movement of the crosshead forward results in a coarser pitch. Other details relating directly to the pitch changing mechanism are similar to those previously described.

The crosshead 148 is secured to a shaft 150 having a piston 151 at an approximate mid-point of the shaft. The shaft 150 has a forward end portion 152 having a bore 153, and an aft end portion 154 having a bore 155.

The midportion 132 has a face 158 having a boss 159 protruding aft. A shaft 160 is secured to the boss 159, the shaft being a sliding fit within the bore 153. The aft

portion **131** of the hub has a forward facing spigot **162**, the spigot being a sliding fit within the bore **155**. The shaft **150** is thus slidably supported at both ends permitting longitudinal movement of the piston disc within the hub. A fixed dividing wall **165** has a bore **166** to accept the shaft **150**, the shaft being a sliding fit in the bore.

The shaft **150** has a central bore **168**, an annular shoulder **169** separating the bore **168** from the bore **153**, a bore **170** connecting the bore **168** with the bore **153**. A pilot spool **172** within the bore **168** has a forward spring **173** adjacent a forward end of the spool between the shoulder **169** and the spool, an aft spring **174** being adjacent an aft end of the spool **172**. A stop **175** is provided at an aft end of the bore **168**. The spool **172** is thus sandwiched between two springs and is slidable within the bore **168**. The spool **172** has a recessed central portion **176** forming lands **176.1** and **176.2** at opposite ends of the spool.

The propeller hub is divided into four separate spaces. An aft cylinder space **177** is defined in part by an aft face of the piston **151** and a forward face of the portion **131**. A forward cylinder space **178** is defined in part by a forward face of the piston **151**, and an aft face of the dividing wall **165**. An aft crosshead space **179** is defined in part by an aft face of the crosshead **148**, and a forward face of the dividing wall **165**. A forward crosshead space **180** defined in part by the aft face **158**, and a forward face of the crosshead **148**. A passage **181** in the crosshead **148** permits fluid to pass between the spaces **179** and **180**.

Passages **182** and **183** connect the space **177** and the space **178** respectively with the central bore **168**. Longitudinal separation of the passages **182** and **183** is such that when the spool **172** is in a centered or equilibrium position within the bore **168**, the lands **176.1** and **176.2** close the passages **182** and **183** respectively. The space **180** is connected to the bore **168** by a longitudinal passage **185** and a radial passage **186**.

The shaft **160** has a passage **188** communicating with a radial passage **189** which, in turn, communicates with further longitudinal passage **190**. As seen in FIG. **5** the passages **190**, via an annular groove **192** in the rotary seal communicate with a pressure port **191** in the rotary seal assembly **134**.

The spigot **162**, FIG. **4**, has a passage **194** which connects with passages severally **195**. As seen in FIG. **5**, the passage **195** via several passages, communicates with a pressure port **196** via an annular groove **195.1** in the rotary seal assembly **134**. The pressure ports **191** and **196** are connected by hydraulic lines to a control unit (not shown), the control unit selectively directing fluid at actuation pressure from a hydraulic pump to the port **196** or the port **191** depending on whether the pitch is to be made coarser or to be made finer. The rotary seal has a further pressure port **198** which is used for aft bearing lubrication if necessary. The port **198**, via an annular groove **203** in the rotary seal, communicates with a longitudinal passage **199** (shown in broken outline in FIG. **5**). The passage **199** communicates with the aft bearing (not shown) and a space **200** at an aft end of the taper **135** of the tailshaft. A passage **201**, shown in broken outline, connects the spaces **200** and **180** permitting fluid scavanged from the hub to be used for aft bearing lubrication via the passage **199**. If the port **198** is not used for lubrication it can be sealed. Fluid scavenged from the hub via the passage **195** leaks past seals in the rotary seal into the groove **203**, similar to that described with reference to FIG. **3**, and lubricates the aft bearings; after which the fluid leaves the bearing in a manner similar to that described with reference to FIG. **1**.

The control unit can be provided with valve means to permit fluid scavenged from the hub to return to the tank via the control unit rather than the aft bearing, in which case one pressure port at any one time, either **196** or **191**, would be a scavenge port passing fluid to the control unit which, through valves, returns fluid to the tank.

A low pressure seal **205** is provided, between the rotary seal **134** and the propeller flange **133**, to reduce ingress of water into the rotary seal and leakage from the seal to the water.

A come home compression spring (not shown) can be provided in the space **177**. This spring urges the piston **151** forward and is for use in an emergency should hydraulic pressure supply to the hub fail. It operates similarly to the come home spring as described with reference to FIGS. **1** and **2**, effecting maximum forward pitch for the propeller. A hand pump is provided for emergency use to overcome any hydraulic lock produced by the spool closing the spaces **177** and **178**. Fluid is pumped by the hand pump into the bore **155** to move the spool forward, admitting fluid into the space **177** and exhausting fluid from the space **178** as the piston **151** is urged forward by the come home spring. If no spring is fitted, the pitch is substantially unchanged after hydraulic pressure failure.

Wear resisting O-ring seals are provided on peripheries of the piston **151**, the bores **155** and **153**, and the spool **172**, as shown. These seals reduce fluid leakage between sliding components, and other undesignated seals are used to reduce fluid leakage.

As seen in FIG. **5**, the rotary seal **134** has three pressure ports as shown, and four wear-resisting seals each designated **204** are provided, one between each port, to control fluid leakage from the ports **191** and **196**. The seals are piston ring type, as described with reference to FIG. **3**.

OPERATION OF THE FIG. 4 ALTERNATIVE FIG. 4

In a constant pitch condition, the spool **172** is held in a balanced, or centered, position by actions of the springs **173** and **174**. In this position the passages **182** and **183** are closed by the lands **176.1** and **176.2** on the spool **172**, and there is no fluid flow to or from the hub. To change the pitch, fluid at actuation pressure is supplied at one end of the spool on demand from the control unit, which spool end depends upon whether pitch is to be made finer or coarser.

To make the pitch of the propeller blades finer, fluid at actuation pressure is supplied to the port **191**, (FIG. **5**) from which the fluid leaves the assembly **134** via the groove **192**, and the passage **190**, enters the passage **188** in the shaft **160** and enters the bore **153**. The fluid flows through the bore **170** and exerts a force on the forward end of the spool **172**, thus urging the spool aft and exposing the passage **183** to fluid at actuation pressure. Fluid at actuation pressure flows into the space

178 through the passage 183 and exerts a force urging the piston 151 aft, following the spool 172, and exhausting fluid from the space 177 through the passage 182.

The equilibrium position of the spool is attained when flow to the bore 153 ceases, re-centering the spool, thus closing the passages 182 and 183. Movement of the piston 151 aft moves the crosshead 148 aft, causing the propeller blade to rotate about the axis 146 to a position of finer pitch. Fluid exhausted from the space 177 enters the bore 168, passes around the recessed central portion 176, and leaves through the passages 186 and 185, to enter the space 180 where it is scavenged from the hub through the passage 201 to the space 200. The passage 199 connected to the space 200 carries fluid to the aft bearing for lubrication. Fluid within the bore 155 is scavenged through the passages 194 and 195 to the groove 195.1 from which it leaks past the seals to the groove 203. From the groove 203, scavenged fluid leaves the flange 133 via the passage 199 which carries the fluid to the aft bearing for lubrication, being similar to the passage 101 FIG. 1.

To make the pitch coarser, fluid at actuation pressure is supplied to the pressure port 196 from which via the groove 195.1 it enters the passages 195 and 194 and the bore 155. The fluid exerts a force on the aft end of the spool 172 urging the spool forward and exposing the passage 182 to the fluid at actuation pressure. The fluid enters the space 177 through the passage 182 and exerts a force on the aft face of the piston 151, urging the piston forward and with it the crosshead 148, resulting in a coarser pitch of the propeller. Fluid in the space 178 is exhausted through the passage 183 into the bore 168, then, via the passages 186 and 185, it enters the space 180, leaving it as before stated. The piston 151 moves forward until fluid flow at actuation pressure ceases, so that the lands 176.1 and 176.2 close the passages 182 and 183 when the spool is re-centered.

Fluid from the bore 153 is exhausted through the passages 188 and 189 into the passage 190 (FIG. 5), from which it is scavenged from the hub by leaking past the seals into the groove 203 then through the passage 199 to the aft bearing.

The spool 172 when in an equilibrium position forms a hydraulic lock for the piston 151, inhibiting movement of the crosshead 148 and pitch creep of the propeller blades.

The high pressure hydraulic pump runs continuously supplying fluid at about 1,000 psi to the control unit (not shown) which has a three-position valve that, when pitch change is required, directs fluid to either the port 191 or to the port 196. A low pressure hydraulic pump (not shown) can be provided for lubrication of the aft bearings through the port 198. The control unit valve has three-positions namely, shut-off, opening to the port 191, and opening to the port 196. The valve is spring-loaded to return it to the shut-off position when pitch change is not required, thus maintaining the spool 172 centered. If pitch change is not required for long periods during tail rotation, fluid at actuation pressure does not reach the pressure ports and aft bearing lubrication through the port 198 is provided.

The pilot spool 172 is means within the bore adapted for forward and aft movement therewithin, as is the spool 62 FIG. 2. The spool 300 is moved by hydraulic force the motion being resisted by a force applied by either the spring 174 or the spring 173 depending upon direction of motion. Re-centering in balance is accomplished by the springs when the hydraulic force ceases to act.

FIGS. 6 and 7 (with reference to FIG. 3)

The rotary seal described in the two previous embodiments may have disadvantages due to wide variations of clearances in running fits between the rotary seal inner sleeve 111 (FIG. 3 only) and the tailshaft, due to practical manufacturing limitations in a fixed rotary seal. Control of fluid leakage between the seals that surround actuation pressure ports is required to effect control of volume flow and to reduce contamination of fluid at actuation pressure with fluid in adjacent pressure ports. There is also a chance of failure of the low pressure seal causing water contamination of the rotary seal or loss of fluid. Fluid at actuation pressure is felt by the rotary seal during pitch change only and controlled leakage is used for lubrication purposes as previously stated. To effect greater control of fluid leakage, a novel floating rotary seal assembly is used in combination with a water-lubricated bearing assembly, as shown in FIGS. 6 and 7. Floatation effect of the rotary seal substantially accommodates eccentricities and unevenly worn portions of the tailshaft. Relatively close running fits can be maintained on the seals between adjacent pressure ports, substantially increasing control of fluid leakage. One example of a floating seal will be described. This type of seal can be used on either embodiment of propeller hub as previously described.

FIG. 7

A tailshaft 220 has an axis of rotation 221 and at an aft end has a taper 222 to fit a taper within a propeller flange 223. A propeller hub (not shown) according to the invention is fitted to the propeller flange 223 aft of FIG. 7. A stern tube 225 is shown in broken outline indicating securing means to a stern of a vessel. Tailshaft aft bearings are shown as 226 open to ambient water, being known water-lubricated bearings.

A passage 228 within the flange 223 communicates with a corresponding passage in the hub (not shown) and two further passages (not shown) in the flange communicate with passages in the propeller hub. The passage 228 in the propeller flange is connected to a hydraulic line 230 which is parallel to and adjacent the shaft 220. The two further passages are similarly connected to two further lines (not shown) and are spaced circumferentially around and parallel to the shaft 220. All three lines are secured to the shaft suitably by a glass fiber and resin composite material to form an enveloping protective sleeve 230.1 enclosing the lines. Only one fluid route from the hub will be described, namely hydraulic lines and passages associated with the passage 228. Other fluid routes are parallel to this and are similar.

FIG. 6

A forward end of the line 231 communicates with a drilled passage 232 within a rotary seal inner sleeve 233 secured to the tailshaft. An annular bracket 235.1 of a known stuffing box 235 surrounds the sleeve 233 and is secured to the stern tube 225. A shoulder 236.1 protrudes inwards from a sleeve portion 236 of the box 235 and provides a stop for packing material 234. A

collar 237 is a sliding fit within the sleeve portion 236, and has at least three similar screw adjusting means 238 (two only shown) securing it to a flange 240 of the box 235. One means 238 only will be described, having a stud 239 secured to the flange 240, the stud projecting axially forward from the flange through a clearance hole in the collar 237 and a nut 241 drawing the collar 237 towards the flange 240. The material 234 provided between the collar 237 and the shoulder 236.1 is compressed for sealing purposes as known in the art. Rotation of the nuts 241 of all the means 238 compresses the packing material, increasing sealing between the shaft and the material compensating to some extent for wear of the material 234. The above describes a conventional stuffing box.

The stud 239 projects through a radially disposed slot (not shown) in a rotary seal flange 242 of a rotary seal assembly 243, nut and washer means 242.1 being provided to restrain the flange 242 longitudinally on the stud 239 and prevent rotation. The rotary seal assembly 243 encloses the shaft 220 and is supported in three places by the studs 239 from the collar. The slots permit small radial movements of the assembly 243 relative to the stuffing box thus permitting the assembly 243 to follow minor eccentricities of the shaft 220 as will be explained.

Three pressure ports severally 245 are provided in the rotary seal 243, four seals 246 being provided, one seal between each port. The seals control fluid leakage between the ports similar to the seals previously described with reference to FIGS. 3 and 5. Two low pressure seals 247 are provided, one seal at each outer end of the rotary seal assembly 243. The stuffing box 235 maintains a substantially dry tailshaft forward of the stuffing box.

Rotation of the shaft 220 rotates the hydraulic lines within the glass fiber composite sleeve 230.1 and rotates the rotary seal inner sleeve 233. Eccentricities of the shaft or inner sleeve 233 produce small radial variations of a periodic nature that are experienced by a fixed point relative to the vessel for example on the stern tube. Such small radial variations on rotation of the shaft can be accommodated by radial movement of the assembly 243, which moves in sympathy with the shaft by sliding of the studs 239 within the radial slots.

A floating seal means as described above can utilize seals 246 having a smaller clearance relative to the main shaft, thus effecting improved control of fluid leakage between the seals. Seapage of water is materially reduced by the packing material 234, this results in a substantially dry shaft adjacent the rotary seal, with the tailshaft bearing being conventionally lubricated by the water.

FIGS. 8 and 9 with references to FIGS. 4 and 5

A further alternative embodiment shown in FIGS. 8 and 9 differs from the embodiment described in FIGS. 4 and 5 in passages associated with a modified pilot spool, and in minor modifications to passage routing within the propeller hub.

The further alternative hub embodiment 260 has an aft portion 261 bolted to a mid-portion 262, a propeller flange 263 being secured to the mid-portion and enclosed at a forward end by a rotary seal assembly 264. The propeller is mounted on a tailshaft 265 having an axis of rotation 266. A propeller root 267, shown fragmented, is mounted for rotation within an opening 262.1 in the mid-portion 262. A crosshead 268 effects pitch change as previously described with reference to the crosshead 148 FIG. 4.

As seen in FIG. 8, the crosshead is secured to a shaft 270 having a piston 271 mounted centrally as shown. The shaft has a forward bore 272 and an aft bore 273. A spigot 274 projecting aft is a sliding fit within the forward bore and a spigot 275 projecting forward is a sliding fit in the aft bore. A fixed dividing wall 276 has a bore 277, the shaft 270 is a sliding fit therein, thus being free to slide axially within the hub.

The hub is divided internally as previously described namely, an aft cylinder space 279, a forward cylinder space 280, an aft crosshead space 281, and a forward crosshead space 282, a passage 283 through the crosshead connecting the space 281 with the space 282. The shaft 270 has a central bore 286 extending forward from the aft bore 273, and passages 288 and 289 connect the spaces 279 and 280 with the central bore. A shoulder 291 separates the central bore from the forward bore 272, the shoulder having an inner bore 292. A passage 294 within the spigot 274 communicates with the bore 272 and communicates with the rotary seal assembly 264 via passages severally 295. A passage 297 within the spigot 275 communicates with the bore 273 and, via passages severally 298, communicates with the rotary seal assembly 264 FIG. 9.

Structure above is generally similar to that shown in FIGS. 4 and 5. The passages 186 and 185 in FIG. 4 have been eliminated in FIG. 8 and the passage 199 in FIG. 5 has been eliminated in FIG. 9 Further differences particularly in passages associated with a pilot spool 300 are described below.

FIG. 8

The pilot spool 300 has an aft face 301 and a forward face 301.1, is a sliding fit within the bore 286, and has a compression spring 302 bearing against the aft face 301, and a compression spring 303 bearing against the forward face 301.1. The spring 302 has an aft end bearing against a stop 304 and the spring 303 has a forward end bearing against the shoulder 291, which shoulder serves as a stop for the spring 303. Thus both springs are compressed and the spool is held in an equilibrium position between them. The spool has a central recessed portion 305 which defines lands 306 and 307 at outer ends of the spool. In a constant pitch condition the spool is held in the equilibrium position in which the lands 306 and 307 close the passages 288 and 289. The spool has diametral passages 308 and 310 at the lands, the passages communicating with longitudinal passages 311 and 312 in the spool as shown. Longitudinal separation of the passages 308 and 310 is such that, when the spool is displaced forward, the passage 288 as exposed to the bore 286 and the passage 310 is in register with the passage 289. Similarly when the spool is displaced aft, the passage 289 is exposed to the bore 286 and the passage 308 is in register with the passage 288. Means to prevent movement of the spool beyond positions in register are provided, namely sleeves 314 and 315 surrounding the springs 302 and 303, the sleeves being of such length that movement beyond the positions of register is prevented. The sleeves 314 and 315 can be omitted if the springs have such a length that, when fully compressed, translation

of the spool beyond registration is prevented, such springs being used in the FIG. 4 embodiment.

A one-way valve **316** is provided in the wall **276** to reduce accumulation of fluid in the spaces **281** and **282**, the valve passing fluid in a direction shown by an arrow **317**.

FIG. 9

The rotary seal assembly **264** has three fluid pressure ports **320**, **321** and **322** communicating with associated annular grooves **323**, **324**, and **325**, within an inner sleeve **326** of the rotary seal assembly **264**. The passage **295** communicates with the annular groove **324** and the passage **298** communicates with the annular groove **325**. A passage **328** shown in broken outline connects the groove **323** to an aft bearing, not shown. Four seals **329** are provided between the grooves **323**, **324**, **325** of the rotary seal assembly **264** being similar to seals in the assembly **134** FIG. **5**. The seals between the pressure ports control leakage of fluid. If fluid leakage is sufficient for aft bearing lubrication the port **320** can be sealed. The ports **321** and **322** are connected to a control unit not shown, and supply fluid at actuation pressure for pitch change purposes only. Should hydraulic pressure supplies to the hub fail a hand operated hydraulic pump is provided to supply hydraulic pressure for use in an emergency. In such an emergency, speed of rotation of the shaft is reduced or stopped, the pump being used to pressurize fluid to displace the spool to eliminate hydraulic locks formed in the cylinder spaces, as previously described.

OPERATION

FIG. **8**, with references to FIG. **4** and **9**

As described with reference to FIG. **4**, translation of the crosshead aft results in a finer pitch of the propeller blade and translation of the crosshead forward results in a coarser pitch. Fluid is supplied to pressure ports **321** and **322** (FIG. **9**) as previously described, and fluid is scavenged from the hub by controlled leakage past the seals **329** for aft bearing lubrication as previously described.

To make the propeller pitch finer, fluid at actuation pressure is fed into the port **321** which supplies fluid through the passages **295** and **294** to the bore **272**. The fluid flows through the bore **292** and produces a force on the spool **300**, moving the spool aft and compressing the spring **302** until further aft translation is prevented by the sleeve **314**. The passage **289** is exposed to fluid at actuation pressure within the bore **286**, the fluid thus entering the space **280** through the passage **289**. Fluid at actuation pressure within the space **280** forces the piston **271** aft, thus moving the crosshead **268** aft making the pitch finer. As the piston moves aft, fluid is exhausted from the space **279** through the passage **288** which is in register with the passage **308** within the spool. The fluid from the space **279** passes through the passages **308** and **311** into the bore **273**, leaving through the passages **297** and **298**. From the passage **298** fluid scavenged from the hub enters the annular groove **325** and leaks past seals **329** into the groove **323**, then through the passage **328** to lubricate the aft bearing (not shown). When the required pitch is reached, delivery of fluid at actuation pressure is stopped, permitting the spool to attain the equilibrium position.

To make the pitch coarser, fluid at actuation pressure is fed into the port **322** which, via the annular groove **325**, passes fluid to the passages **298** and **297** into the bore **273**. Fluid enters the bore **286** and produces a force urging the spool forward, compressing the spring **303** until further forward movement of the spool is prevented by the sleeve **315**. The passage **288** is thus exposed to fluid at actuation pressure within the bore **273**, the fluid entering the space **279** and producing a force forward on the piston **271**. The piston moves forward moving the crosshead forward, thus making the pitch coarser. Fluid within the space **280** is exhausted through the passage **289** into the passage **310** within the spool. Fluid flows through the bore **292** into the bore **272**, which it leaves through the passages **294** and **295** to enter the rotary seal. The passage **295** leads the fluid into the annular groove **324** from where the fluid leaks past one seal **329** into the groove **323** which transmits scavenged fluid through the passage **328** to lubricate the aft bearing. The force on the piston **271** slides the shaft **270** forward and with it the crosshead **268**, thus making the pitch coarser. As before stated, when the required pitch is attained delivery of fluid at actuation pressure ceases, and the spool returns to the equilibrium position. The spool **300** is means within the bore adapted for movement generally as described with reference to the spool **172**.

A come home spring can be provided in the space **279**, if desired, to return the pitch to full ahead position should hydraulic pressure fail, this being used in emergency as previously described.

ANCILIARY EQUIPMENT ASSOCIATED WITH THE PROPELLER HUB

FIG. **10**

The propeller hub **10** of the embodiment of FIG. **1** is secured to the tailshaft **16** and journalled in the stern tube **17.1** at a stern **339** of a vessel, and is driven by an engine gearbox and clutch assembly **340**. Hydraulic lines **341** and **342** are connected to the ports **94** and **102** respectively in the rotary seal **14**, the lines transmitting fluid from a control unit and pump assembly **343**. A fluid supply tank **344** is connected to the assembly **343** by a hose **345**, and a scavenge hose **346** returns fluid from the passage **107** to the tank **344** after the fluid has lubricated the stern tube. An extra hose carrying fluid for aft bearing lubrication to the optional port **103** (not shown) is omitted in this figure.

Environment above accommodates the propeller hub embodiments described with reference to FIGS. **4** and **5** and FIGS. **8** and **9**. If a floating rotary seal (FIGS. **6** and **7** or FIGS. **11**, **12** and **13**) is used, the hydraulic lines connect with a rotary seal forward of the stern tube (not shown in FIG. **10**).

Pitch Readout

For the embodiment illustrated in FIGS. **1–3**, pitch of the propeller is known from the position of the spool **62** relative to the hub, which position can be calculated from known characteristics of the spring **69**. Determination of fluid pressure in the bore **64** (measured by a pressure gauge, not shown) indicates forces on the spool **62**, determining the position of the spool relative to the hub, hence the pitch.

Referring particularly to the embodiments illustrated in FIGS. **4**, **5**, **8**, and **9**, exhaust temperature is a sensi-

tive indicator of engine load, and pitch is a parameter affecting load. Consequently, optimum load can be approached by varying the pitch. The embodiments above are well adapted for use with known automatic means to effect the foregoing.

ALTERNATIVE ROTARY SEAL ASSEMBLY (DRY TAILSHAFT)

FIG. 11

An alternative rotary seal assembly **350** has a fixed outer sleeve **351** having an inner surface **351.1** and an inner sleeve **352** secured to a tailshaft **353**, the tailshaft having an axis of rotation **353.1**. The sleeves **351** and **352** can each be made in two halves and are shown thus in FIG. 11. The outer sleeve **351** has a rotary seal flange **354** which has a radial slot **354.1** to carry a stud **355** of a screw adjusting means **356**. The screw adjusting means is secured to a fixed part of the vessel aft of the rotary seal. The rotary seal assembly has at least three screw adjusting means similar to those as described with reference to FIGS. **6** and **7**, the slots providing accommodation to eccentricities of rotation of the tailshaft by permitting movement of the studs relative to the radial slots.

The inner sleeve **352** has an outer surface **352.1** and two passages **359** and **360** to accept fluid at actuation pressure for delivery to the propeller flange (not shown). The passages **359** and **360** communicate with inlet annular grooves **361** and **362** respectively of the rotating inner sleeve **352**. Pressure ports **364** and **365** communicate with the annular grooves **361** and **362** respectively and during pitch change admit fluid at actuation pressure to one of the passages **359** or **360**. An outlet annular groove is longitudinally spaced on either side of each inlet annular groove and is separated from the inlet annular groove by an annular element, hereinafter dam. The inlet annular groove **361** is separated from outlet annular grooves **367** and **369** by dams **368** and **370** as shown. The inlet annular groove **362** is separated from outlet annular grooves **371** and **373** by dams **372** and **374**. Outlet ports **376** and **377** communicate with outlet annular grooves **367** and **369** and outlet ports **379** and **380** communicate with outlet annular grooves **371** and **373**.

A seal **383**, sandwiched between two annular baffles **384** and defining in part a further undesignated groove, is provided between an inner pair of outlet grooves **369** and **371**. The seal **383** is a piston ring type seal similar to those used in piston engines, and is further described with reference to FIGS. **12** and **13**. Similar annular baffles **386** and **387** defining in part further undesignated grooves, are adjacent the annular grooves **367** and **373** and provide partial location for further piston ring seals **388** and **389**. The piston rings are sprung radially outwards, thus gripping the inner surface **351.1** of the rotary seal outer sleeve **351**.

The dams serve as fluid leakage control means between inlet grooves and outlet grooves, an outer cylindrical surface **370.1** of the dam **370** being separated from the inner cylindrical surface **351.1** of the sleeve **351** by a clearance gap **390** having a radial dimension limited by maximum variation of clearance between the inner sleeve **352** and the outer sleeve **351** when one rotates relative to the other. This variation is typically within a range of one- to three-thousandths of an inch. The clearance gaps are best seen in FIG. **12**. High volume rate of fluid flow into the outlet flow at reduced pressure to the outlet port, control of pressure and volume flow rate being effected by the gap **390**, and a valve that controls fluid flow from the outlet ports as is later explained. The baffles serve as means to locate the seals axially and also, similarly to the dams aforesaid, to reduce fluid pressure at the seal. The baffles have a similarly controlled clearance, later described.

Ball bearings **391** and **392** journal the outer sleeve on the inner sleeve, and twin seals **394** and **395** reduce leakage from the rotary seal assembly and reduce possible contamination of fluid in the seal from external sources. Maximum run-out or eccentricity of the bearing is a main factor in determining a minimum limit of the clearance gap. Inner faces of the twin seals are in contact with bearing sleeves **397** and **398**, the sleeves having chamfered outer edges **399** and **400**, the chamfers easing assembly of the sleeves over the piston ring seals. O-ring seals **401** and **402** reduce leakage of fluid between the bearing sleeves and the rotary seal inner sleeve **352**. The bearing sleeves locate the ball bearings and are held in place by spring clips **403** and **404**.

FIGS. 12 and 13

The piston ring seal **383** is shown between the inner sleeve **352** and the outer sleeve **351**. The seal **383** has outer and inner cylindrical surfaces **383.1** and **383.2**, and parallel radial faces **383.3** and **383.4**, the seal being an annulus with a gap and having a rectangular cross-section. An outer cylindrical surface **384.1** of the baffle **384** is shown in broken outline and is separated from the inner cylindrical surface **351.1** of the outer sleeve **351** by a radial clearance gap **406**, which, similarly to the clearance gap of the dams, is typically of the order of a few thousands of an inch. This gap is dependent of fluid viscosity actuation pressure and other operational requirements, and has a minimum size limited by maximum run-out of the ball bearings.

The seal outer surface **383.1** is sprung outwards against the inner surface **351.1**, effectively in metal to metal contact to prevent fluid flow, but is lubricated and is relatively free to slide circumferentially if required, i.e., to rotate about the axis **353.1**. Such springing is well known in the art of piston rings as used in internal combustion engines.

The inner cylindrical surface **383.2** is spaced from the outer cylindrical surface **352.1** of the inner sleeve **352** providing a radial gap **407** between the two surfaces. The gap **407** is for fitting of the piston ring type seal, as is well known. The seal **383** has a conventional circumferential lap join as shown at **408**, in which two circumferential fingers **409** and **410** overlap, circumferential clearance provided by two circumferential gaps **411** of two- to three-thousandths of an inch each. The gaps **411** are close tolerance gaps to reduce excessive fluid leakage past the overlap.

A plurality of passages **413** is provided through the piston ring seals **383**. The passages have a diameter of approximately twenty-thousandths of an inch, and are disposed parallel to the axis **351.1**. A passage of this diameter can pass fluid from the face **383.3** to the face **383.4** and vice versa. A plurality of radially disposed passages **415** also about twenty-thousandths of an inch in diameter is provided through the piston ring seal

connecting the outer surface **383.1** with the inner surface **383.2**. The passages **413** and the passages **415** are circumferentially spaced alternatively around the piston ring seals, spacing between the passages being equal.

These passages provide lubrication between faces in contact with the piston ring seals. Such lubrication reduces wear of the piston ring seal increasing operating life of the seal.

OPERATION OF THE ALTERNATIVE ROTARY SEAL

Fluid at actuation pressure is fed either into the port **364** or **365**, depending on whether the pitch is to be made finer or coarser. Typically a flow of fluid into the hub at 3 gallons per minute for about 8 seconds effects pitch change from fine to coarse.

If the pitch is to be made coarser, fluid at actuation pressure is fed into the input port **364** at a volume rate of flow in excess of that required to operate the hub, e.g., a volume rate of flow of 9 gallons per minute. The groove **361** receives the fluid from the port **364** and the flow rate of 3 gallons per minute of fluid leaves the groove through the passage **359** to enter the propeller hub. Excess fluid at a rate of 6 gallons per minute leaks past the dams **368** and **370** into the annular grooves **367** and **369**, leaving the rotary seal assembly through the outlet ports **376** and **377** in scavenge hoses not shown. The scavenge hoses are siamesed together to a main scavenge hose, flow through the hoses being controlled by a scavenge flow valve, not shown. This excess fluid flow is at reduced pressure, as explained below. Fluid delivery pressure into the annular groove **361** is of the order of a thousand psi, but after fluid is metered by flowing past the dams pressure in the annular grooves **367** and **369** is considerably less. Due to reduced pressure in the grooves **367** and **369**, a relatively small amount of fluid leaks past the baffles **386** and **384**, the baffles producing a further metering of the fluid which produces relatively low fluid forces on the seals **388** and **383** resulting in reduced wear of the seals. Wear resulting from tendency of the seals **388** and **383** to move relative to faces with which the seals are in contact, is reduced by the lubrication provided by the passages **413** and **415** and relatively low forces between the surfaces because of relatively low ambient fluid pressures. Axial forces on the seal being thus reduced, service life of the seals is increased.

If the fluid pump fails, the tailshaft can be stopped, the scavenge flow valve closed, and pitch changed by fluid flow from the emergency hand pump, flow from which would be fed into the propeller hub to change to a desired pitch with substantially no fluid leakage past the dams due to the closed scavenge flow valve. When the desired pitch is attained, the hand pump is stopped, the tailshaft is restarted, and the vessel proceeds.

What I claim is:

1. A controllable pitch propeller mechanism having a generally hollow hub having an axis, blades mounted in the hub, the hub being adapted to be secured to a tailshaft of a vessel the hub rotating with the tailshaft; in combination with the foregoing,

a. a shaft within and co-axial of the hub, and adapted for axial movement relative to the hub; a piston and a crosshead mounted on the shaft moving therewith; the shaft having a bore,

b. means, within the bore, slidably adapted for forward and aft movement within the bore and relative to the hub, the movement being responsive to difference between a force applied to the said means tending to move it forward, and a force applied to the said means tending to move it aft, the means aforesaid remaining in an initial balanced position stationary with respect to the hub and to the bore so long as the forces remain equal, the means including:

b.1. a pilot spool within the shaft bore, a compression spring bearing against the forward end and a compression spring bearing against the aft end applying equal forces to each end of the spool and centering the spool in a balanced position within the bore,

b.2. the spool having a recessed central portion forming lands at opposite ends of the spool,

b.3. a passage extending from the bore to an aft cylinder space, a passage extending from the bore to a forward cylinder space, the passages being closed by the lands when the spool is in the balanced position,

b.4. means selectively to supply and direct hydraulic fluid under pressure to either end of the spool so that there is a difference in force applied to each end of the spool effecting axial movement as aforesaid, which movement opens both of the passages,

b.5. means admitting the hydraulic fluid under pressure to the recessed central portion of the spool so that, when both of the passages are opened as aforesaid by movement of the spool, the piston is urged to move, and with it the shaft and crosshead, altering the pitch of the blade,

b.6. and means to scavenge fluid displaced by movement of the piston, the means being constructed and arranged so that, with closing of a valve of the means selectively supplying and directing the hydraulic fluid as aforesaid, there is no longer difference in force supplied to each end of the spool, and the spool resumes the balanced position at which both the passages opened as aforesaid are closed, effectively locking the blade in a changed pitch position,

c. means responsive to the movement as aforesaid concurrently to cause the piston, and with it the shaft and the crosshead, to move axially relative to the hub through the particular distance, so that in the new balanced position, the pilot spool has the same position relative to the shaft bore as in the initial position,

d. means responsive to the movement of the crosshead through the particular distance to change pitch of a blade, the means including:

d.1. a propeller blade having a longitudinal axis, a root, a blade connecting plate at the root, the connecting plate having a boss,

d.2. the hub having a mid portion with an opening, and means mounting the blade within the opening for rotation about the longitudinal axis and providing restraint from force on the blade as the hub rotates,

d.3. the crosshead having a flat parallel to an inner face of the plate and perpendicular to the blade longitudinal axis, the flat having a groove at an

angle to the shaft axis, adapted for the crosshead movement to rotate the blade about its longitudinal axis so altering the pitch.

2. A controllable pitch propeller mechanism having a generally hollow hub having an axis, blades mounted in the hub, the hub being adapted to be secured to a tailshaft of a vessel the hub rotating with the tailshaft; in combination with the foregoing,

a. a shaft within and co-axial of the hub, and adapted for axial movement relative to the hub; a piston and a crosshead mounted on the shaft moving therewith; the shaft having a bore,

b. means, within the bore, slidably adapted for forward and aft movement within the bore and relative to the hub, the movement being responsive to difference between a force applied to the said means tending to move it forward, and a force applied to the said means tending to move it aft, the means aforesaid remaining in an initial balanced position stationary with respect to the hub and to the bore so long as the forces remain equal,

c. means at each end of the means (b) to apply equalized forward and aft resilient forces when the means (b) is in the balanced position,

d. means to apply a hydraulic force to either end of the means (b) when change of pitch is required, so as to produce an unbalanced force tending to move the means (b) relative to the bore,

e. means responsive to the movement as aforesaid concurrently to cause the piston, and with it the shaft and the crosshead, to move axially relative to the hub through the particular distance, so that in the new balanced position, the means (b) has the same position relative to the shaft bore as in the initial position,

f. means responsive to the movement of the crosshead through the particular distance to change pitch of a blade, the means including a propeller flange secured to the tailshaft forward of the hub and a fixed rotary seal assembly circumferentially enclosing and co-operating with the propeller flange, clearance being provided between the seal and the flange, with means to introduce hydraulic fluid under pressure from the seal assembly to the flange for operation of the mechanism the seal assembly having;

a. a bore having ports adapted to be fed with hydraulic fluid under pressure,

b. annular grooves within a sidewall of the bore of the seal, the grooves communicating with the ports,

c. passages in the flange, the passages communicating with the grooves and being adapted to transmit the fluid through the flange to the hub so as to apply the force hydraulically to the means (b) as aforesaid.

3. A combination as defined in claim 1 including, a rotary seal assembly circumferentially enclosing and co-operating with the tailshaft, the rotary seal assembly being adapted to introduce hydraulic fluid under pressure from the seal assembly to the flange for operation of the mechanism, the seal having:

a. an inner sleeve secured to the tailshaft,

b. a rotary seal flange having an outer sleeve with an inner surface and at least three radial slots,

c. adjusting means co-operating with the radial slots and providing accommodation to eccentricities of rotation of the tailshaft,

d. pressure ports in the fixed outer sleeve adapted to receive hydraulic fluid under pressure,

e. annular grooves in the inner surface of the outer sleeve communicating with the ports,

f. passages in the inner sleeve connecting the annular grooves with the hub, constructed and arranged to transmit fluid into the bore of the hub to apply the said force hydraulically as aforesaid.

4. A combination as defined in claim 1 in which the propeller has a propeller flange secured to the tailshaft forward of the hub, and a rotary seal assembly circumferentially enclosing and co-operating with the tailshaft, including means to introduce hydraulic fluid under pressure from the seal assembly to the hub for operation of the mechanism, the means having:

a. an inner sleeve secured to the tailshaft, the inner sleeve having an outer surface,

b. a fixed outer sleeve having an inner surface spaced from the outer surface of the inner sleeve by a clearance, the outer sleeve having a flange with at least three radial slots,

c. adjusting means co-operating with the radial slots and providing accommodation to eccentricities of rotation of the tailshaft,

d. the outer sleeve having inlet ports and outlet ports connectable to hydraulic pressure supply lines and scavenge lines respectively,

e. the inner sleeve having annular grooves communicating with the inlet ports of the outer sleeve, and outlet grooves communicating with the outlet ports of the outer sleeve, the inlet grooves being spaced from the outlet grooves,

f. dams provided between the inlet and outlet grooves, the dams having outer cylindrical surfaces separated from the inner cylindrical surfaces by a clearance gap, constructed and arranged so that a portion of the fluid entering the inlet groove from the inlet port enters the hub, a remaining portion of the fluid leaking at a controlled rate through the clearance gaps between the dams and the inner cylindrical surface into the outlet grooves, leaving the rotary seal assembly through the outlet port.

5. A combination as defined in claim 4 including:

a. seals provided in further grooves in the inner sleeve between an inner pair of outlet grooves and at either end of the rotary seal to reduce loss of fluid from the seal, the further grooves being separated from the outlet grooves,

b. annular baffles sandwiching the seals, the baffles having outer surfaces separated from the inner cylindrical surface by a clearance gap as aforesaid, the baffles separating the further grooves from the annular grooves, constructed and arranged so that a portion of the fluid entering the inlet groove from the inlet port enters the hub, a remaining portion of the fluid leaking at a controlled rate through the clearance gaps between the dams and the inner cylindrical surface into the outlet grooves, leaving the rotary seal assembly through the outlet port, and fluid flow from the outlet grooves across the baffles is at a reduced pressure

producing low forces on the seals reducing wear of the seals.

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