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Kobelt

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[54] **APPARATUS FOR AUTOMATIC PITCH COMPENSATION IN MARINE VESSELS**

[76] Inventor: **Jack R. Kobelt,** 6110 Oak Street, Vancouver 13, British Columbia, Canada

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[52] **U.S. Cl.** **416/27**

[51] **Int. Cl.** **B63h 3/10**

[58] **Field of Search** 416/25, 27

[56] **References Cited**

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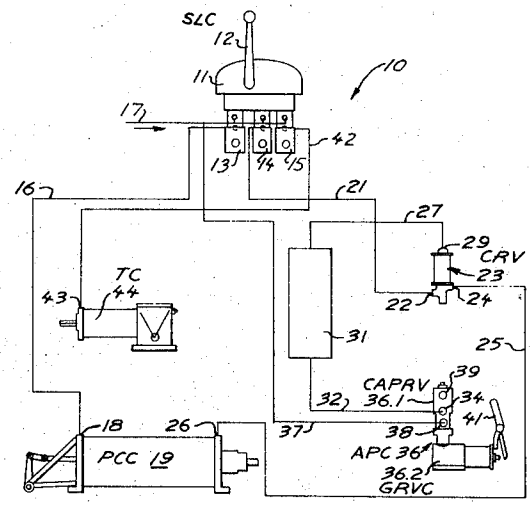
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Primary Examiner—Everette A. Powell, Jr.
Attorney—Brian J. Wood

[57] **ABSTRACT**

Pneumatic control circuit for marine vessel having controllable pitch propeller. Manual pneumatic signals of pitch and throttle are modified by automatic pitch compensator coupled to engine governor output, which senses changes in governor throttle demand, changes being a measure of change in engine load condition. Particular embodiments of automatic pitch compensators can be coupled to governor having output shaft of relatively high torque, or relatively low torque, or governor with hydraulic pressure ports.

11 Claims, 9 Drawing Figures



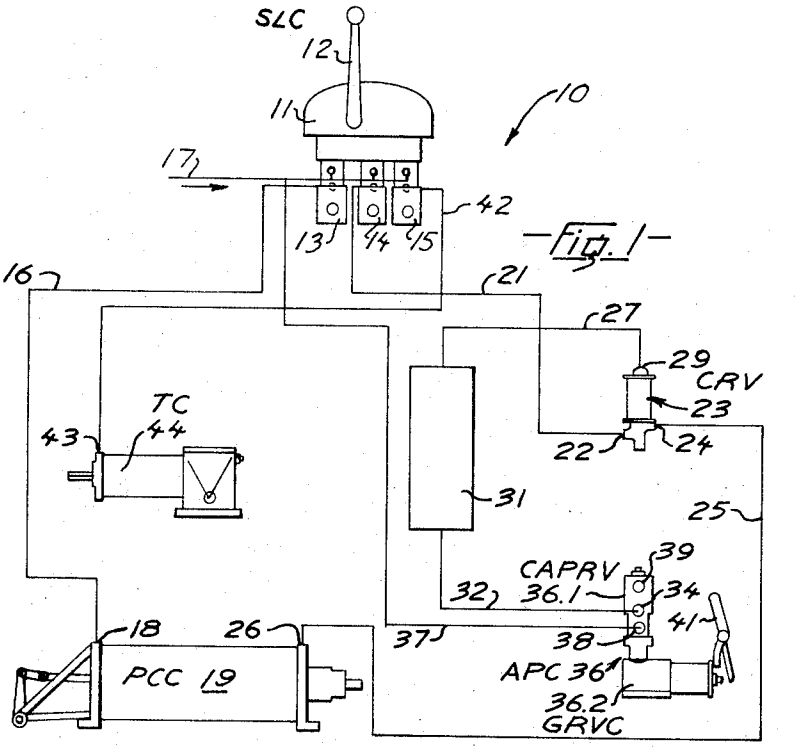


Fig. 1

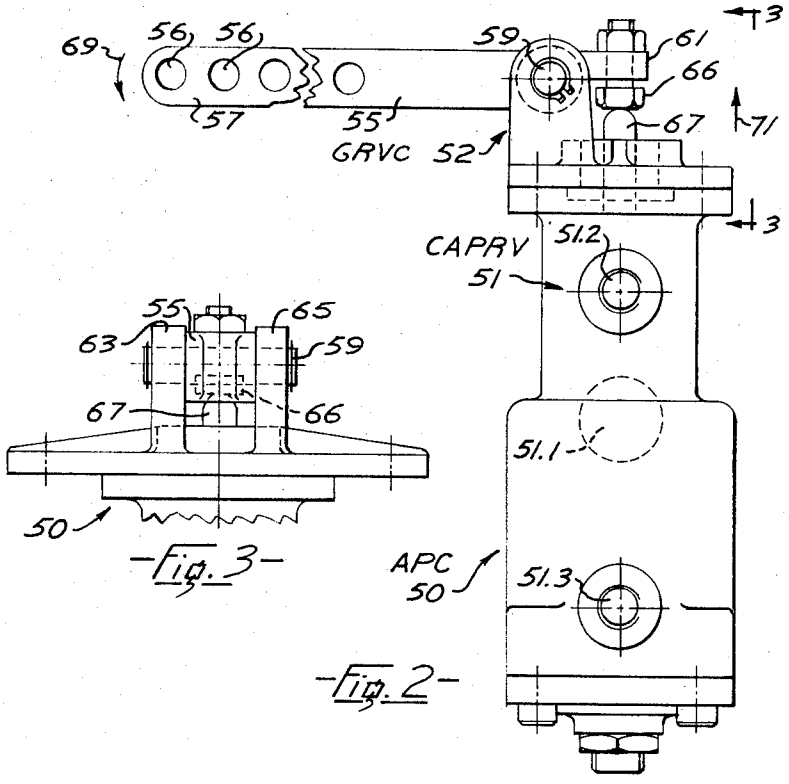
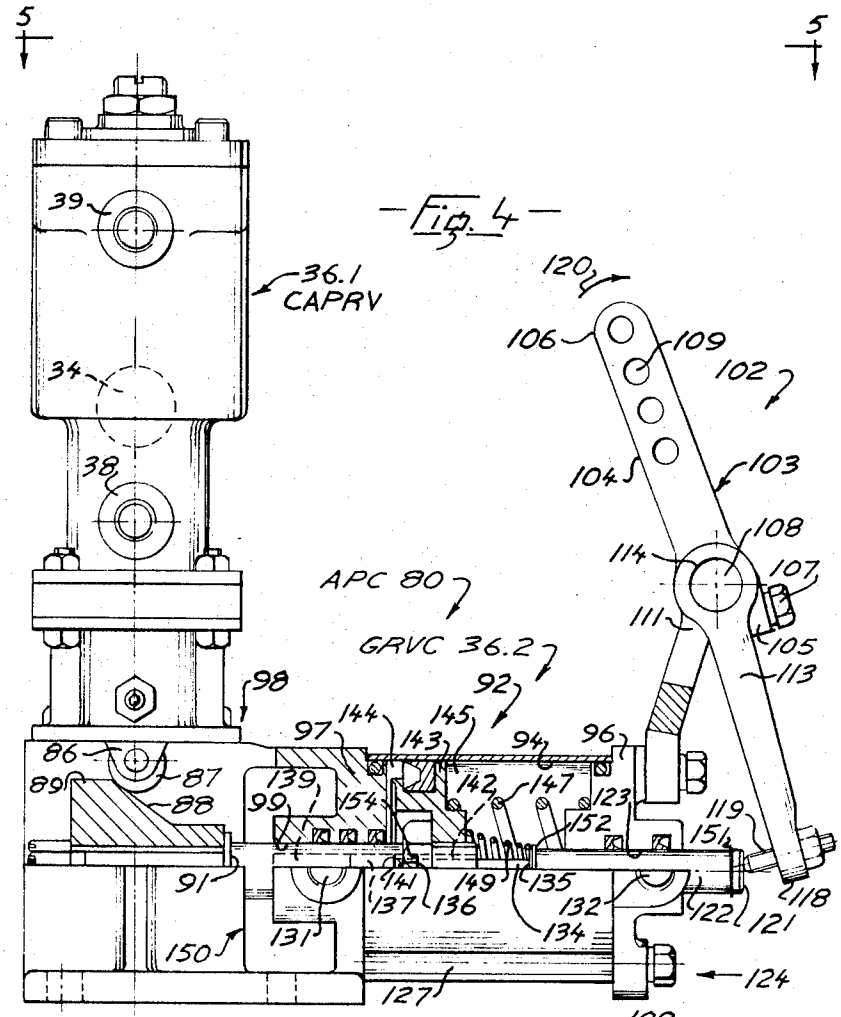


Fig. 2

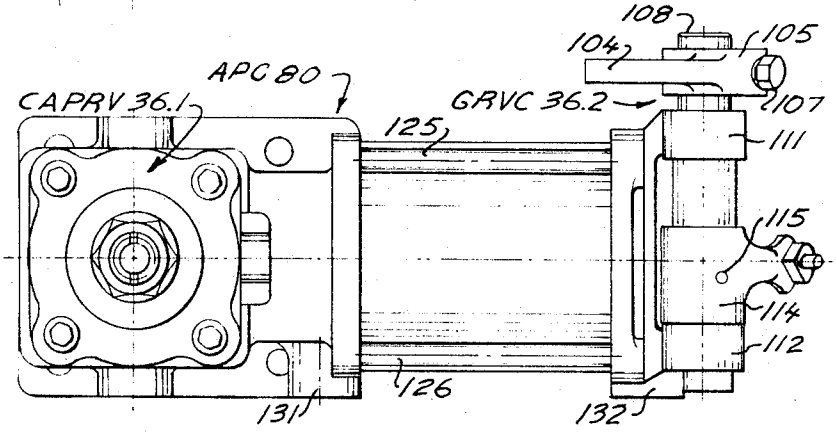
Fig. 3

Jack R. Kobelt,
Inventor

by Lyle G. Trorey,
Agent



—Fig. 4—



—Fig 5—

Jack R. Kobelt,
Inventor

by

Lyle G. Trorey,
Agent

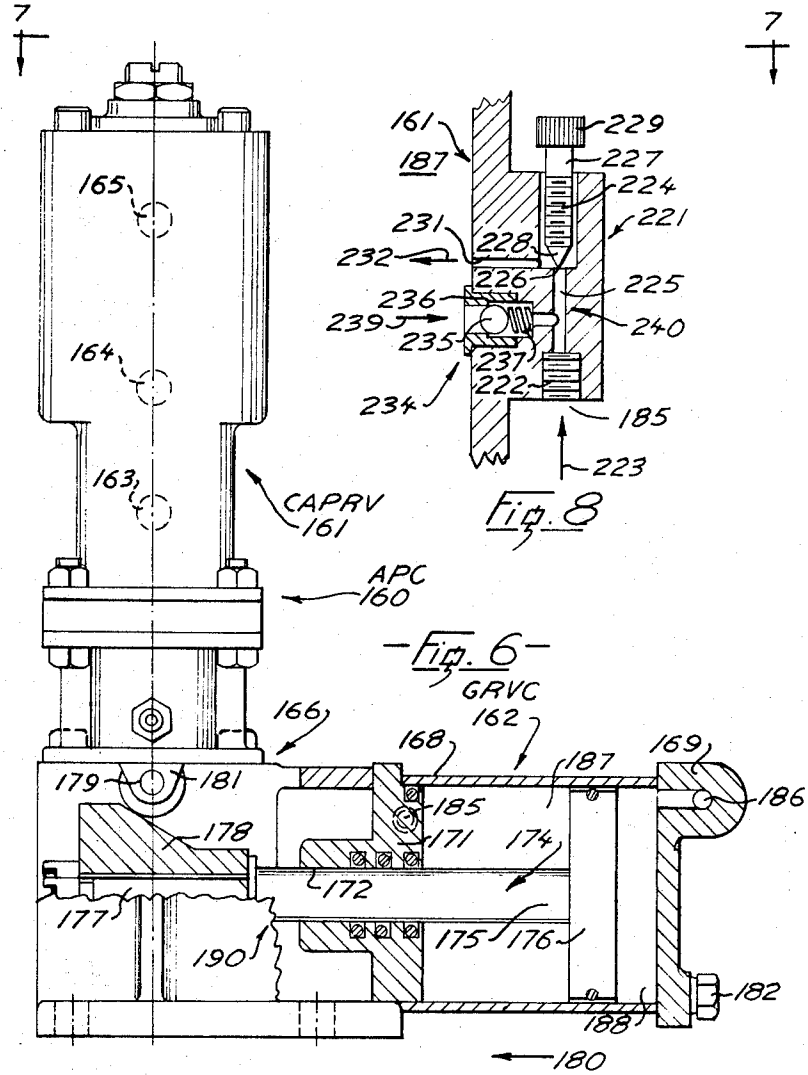


Fig. 8

Fig. 6

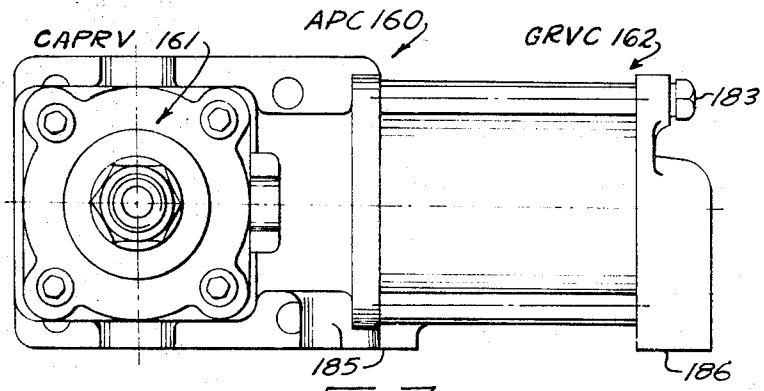


Fig. 7

Jack R. Kobelt,
Inventor

by

Lyle B. Trorey,
Agent

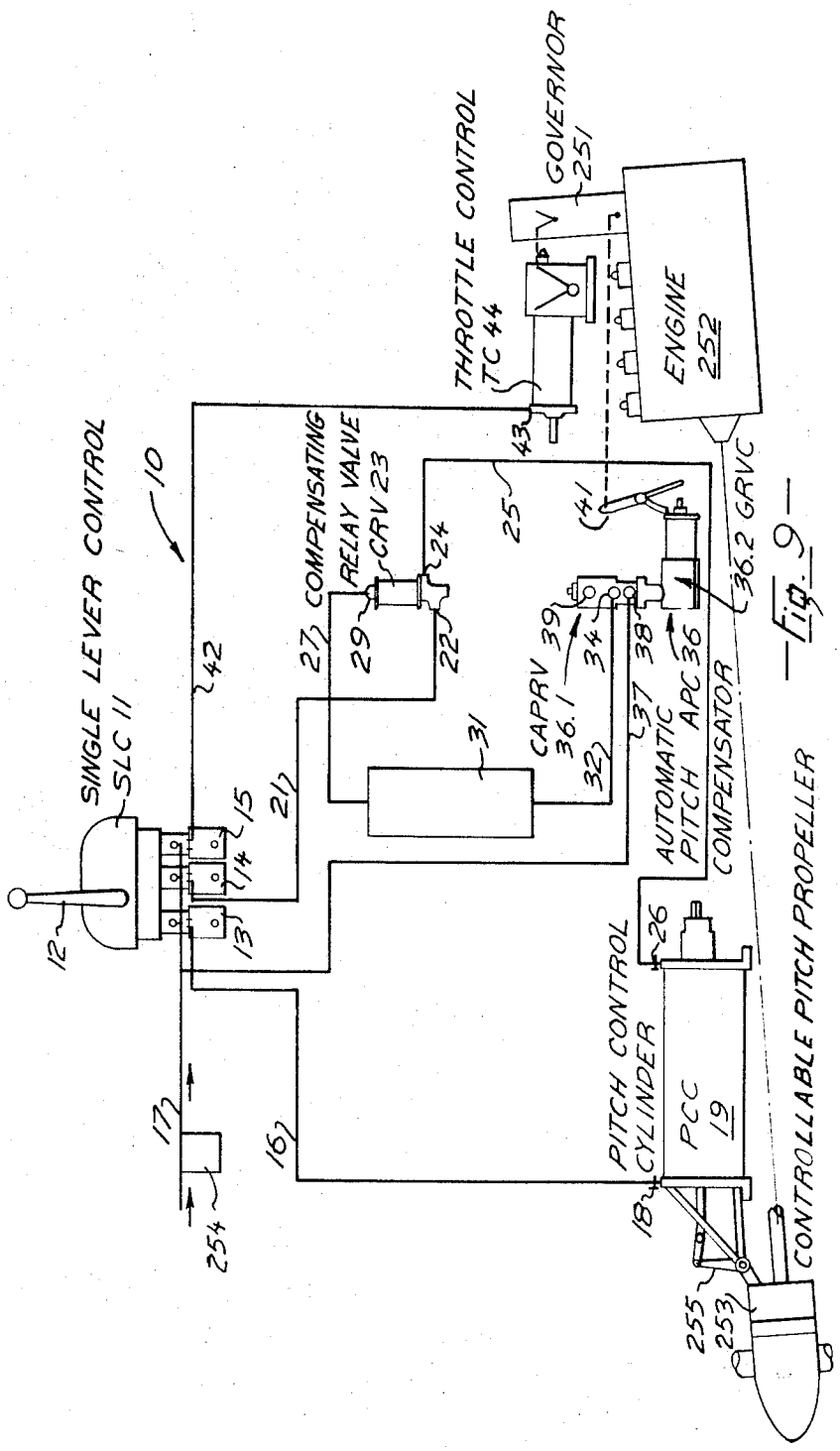


Fig. 9

Jack R. Kobelt,
Inventor

by

Lyle G. Trorey,
Agent

APPARATUS FOR AUTOMATIC PITCH COMPENSATION IN MARINE VESSELS

BACKGROUND OF THE INVENTION

1. FIELD OF THE INVENTION

The invention relates to a pneumatic circuit and components for use in power control installations in marine vessels, particularly tugs, in which a single lever, or equivalent two lever, control concurrently controls throttle, and pitch of a controllable pitch propeller.

2. PRIOR ART

Commonly, pneumatic control installations using a single lever control have a programmed sequence of increase of throttle setting. Conditions arise when, for a given pitch setting, the vessel does not attain sufficient speed to permit the engine to operate at the programmed rpm at the programmed pitch. Consequently the engine becomes overloaded in these conditions.

Load on the engine varies from a first condition when the tug is travelling at full speed without a barge, to a second condition when the tug is towing a maximum displacement barge. A third condition alternating between extremes of the first and second conditions occurs in heavy sea when with a long period of swell e.g. over thirty seconds between wave crests, wave motion causes relative movement between the tug and the barge slackening the tow line and producing fluctuations in engine load. The two extreme conditions require different pitch settings of the propeller, and compensation for the fluctuations in load without manual adjustment of the single lever control is desireable. Manual override systems are known, but require surveillence and, as such, are subject to human error with a risk of damage to the engine.

SUMMARY OF THE INVENTION

Hydraulic circuit and components according to the invention reduce difficulties above by modifying manually selected signals of pitch. The device compensates for differences in load of an engine due to variations in resistance to motion of the tug with or without a barge. The variations can originate from a change in barge resistance or from long period wave motion which produces relative movement of the barge and the tug. Response of the system is generally too slow for short period fluctuations to be compensated.

Compensation is initiated by an automatic pitch compensator according to the invention, the compensator being linked to a fuel rack of the engine or other rpm control device. A change in fuel rack position is detected by the compensator, such change resulting from a demand from the governor for a change in fuel delivery to the engine which, without such change in fuel would change speed from that initially set by a throttle control.

A forward pitch pneumatic line from the single lever control is in circuit with a compensating relay valve operated by pilot pressure from the automatic pitch compensator.

Three embodiments of the automatic pitch compensator are described. Each embodiment includes an air metering means incorporated in a compensating air pressure regulating valve, operatively connected to an engine rpm control device. Such connection is mechanical by way of a fuel rack of an engine, or by

rotation of a governor output shaft; or hydraulic by way of hydraulic pressure outputs from an engine governor. Air from a supply source is fed into the pitch compensator and is metered to pilot pressure, which pressure reflects an amount of pitch compensation required to maintain the engine at a pre-set rpm. The operative connection between the automatic pitch compensator and the engine rpm control device is selected to be compatible with characteristics of both.

The first embodiment of an automatic pitch compensator has a connection to the air metering means including a lever pivoted at a fulcrum, adjacent a pushrod end of the regulating valve, an opposite end of the lever being mechanically linked to the fuel rack of the engine or an output shaft of the governor.

The second embodiment of the automatic pitch compensator has a servo device in which engine oil flowing through the automatic pitch compensator is metered by relative positions of a plug and metering bore within a piston actuating rod, relative positions of both reflecting amount of pitch compensation required to maintain engine at selected rpm and being controlled by position of the fuel rack or governor output shaft. The piston rod has a cam, position of which cam determines degree of metering in the regulating valve. This type of compensator is selected for use with governors with inadequate torque at the output shaft to actuate the regulating valve directly.

A third type of automatic pitch compensator is selected for use with governors having hydraulic outputs and is connected to hydraulic pressure ports of the governor. Fluid volume flow and direction between the ports reflects an amount of pitch compensation and change in fuel rack position required to maintain selected engine rpm. The ports communicate with opposite ends of a fluid cylinder having a piston secured to one end of a piston actuating rod, an opposite end of the rod having a cam operating a cam follower of the regulating valve. Fluid flows to one side of the piston, and is scavenged from the other side of the piston, thus moving the piston through the cylinder, actuating the regulating valve.

Each of the three compensators respond to a signal from the governor producing a change in fuel delivery to maintain the predetermined rpm. The signal also changes pilot pressure output from the regulating valve of the compensator. Pilot pressure change modifies output of the compensating relay valve in such a manner that the signal to the pitch control cylinder for forward pitch as determined by the manual control is compensated to suit current engine loading conditions. Thus, the propeller operates at a compensated pitch, which pitch without the pitch compensator would cause undesirable engine operation. Fuel delivery to the engine is also changed by the compensation, thus by successive approximation, the pitch is matched to engine load.

A detailed description following, related to drawings, gives exemplification of apparatus and method according to the invention which, however, is capable of expression in method and means other than particularly described and illustrated.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic of a hydraulic circuit associated with an automatic pitch compensator, single lever control, and related components,

FIG. 2 is a fragmented side elevation of a mechanically actuated automatic pitch compensator,

FIG. 3 is a fragmented end elevation seen from 3—3 of FIG. 2,

FIG. 4 is a partially sectioned side elevation of a hydraulic servo actuated automatic pitch compensator,

FIG. 5 is a top plan view seen from 5—5 of FIG. 4,

FIG. 6 is a partially sectioned side elevation of a hydraulically actuated automatic pitch compensator,

FIG. 7 is a top plan view seen from 7—7 of FIG. 6,

FIG. 8 is a section of a biased flow control needle valve means used in the embodiment shown in FIGS. 6 and 7,

FIG. 9 shows the schematic circuit of FIG. 1 with diagrammatic representations of operative connection of components to a variable pitch propeller and to a governor of an engine.

Left hand and right hand designations apply to the automatic pitch compensators as shown in FIGS. 4 and 6.

DETAILED DISCLOSURE

FIGS. 1 and 9, circuit and components

A circuit **10**, shown schematically, is a means of compensating a manually selected signal for forward pitch of a controllable pitch propeller, **253** FIG. 9 only. Compensation of pitch is required because of changes of load imposed on the engine. The compensation alters fuel delivery to an engine **252**, FIG. 9 only, to maintain a predetermined rpm. A signal from a governor **251**, FIG. 9 only, to change the fuel delivery from that originally delivered by a throttle control is used as a measure of engine load.

A single lever control box **11**, hereinafter SLC **11**, operated by a hand lever **12**, has three cam operated pneumatic metering valves **13**, **14**, **15**, the valve **13** controlling reverse pitch of the propeller, the valve **14** forward pitch, and the valve **15** throttle control. With a single lever control, movement of the lever controls pitch and throttle concurrently. A two lever control (not shown) can be substituted for the single lever control — in which case the pitch and throttle signals are independent. Single lever and two lever controls as above are herein referred to generically as manual pitch and throttle controls. With either type of control box, movement of a lever or levers adjusts pitch of the propeller and the valve **15**, which valve is used to set input to the governor **251**, controls the rpm at which the engine is to operate. In usual engine/governor set-ups, the governor attempts to maintain the engine at the pre-set rpm by changes in the fuel supply. This can result in overload.

Filtered, compressed, and dried air from a source **254**, (FIG. 9 only), entering the valves through an inlet line **17** is directed and metered by means of the lever **12** and the valves to a pneumatic line associated with the control mechanism. To ensure reliable operation under severe conditions likely to be encountered in marine installations, dry filtered air is specified. Hereinafter and in the claims, air under pressure means dry filtered air as above.

The valve **13** has an outlet pneumatic line **16** leading to a reverse pitch port **18** of a pitch control cylinder **19**, hereinafter PCC **19**. The PCC **19** controls a known pitch changing mechanism of a hub of the controllable pitch propeller **253**, and is also known as a positioner cylinder. The PCC **19** is linked mechanically, as seen at **255** FIG. 9 only, to the pitch changing mechanism and effectively controls a hydraulic servo valve, which in turn controls hydraulic pressure to the pitch changing mechanism. The PCC and means of linking to the controllable pitch propeller are well known.

The valve **14** has an outlet pneumatic line **21** leading to an inlet port **22** of a compensating relay valve **23**, hereinafter CRV **23**. An outlet port **24** communicates with a forward pitch port **26** of the PCC **19**. Pressure in the pneumatic line **25** is modified from the pressure in the line **21** through a metering means within the CRV **23**, metering being controlled by pilot pressure within a pneumatic line **27** which enters the valve **23** at a pilot pressure input port **29**. The CRV **23** is of a known type such that, for a given input pressure at the port **22**, an increase in pilot pressure at the port **29** proportionally increase output pressure at the port **24**. Thus output pressure in the line **25** is regulated by an automatic pitch compensator as is later explained. An accumulator tank **31** is provided in the line **27**, the tank serves as a buffer means against sudden pressure changes which occur when adjusting response of the circuit in initial setting up, by providing a relatively large volume of air to decrease sensitivity of needle valve adjustments when setting response of the circuit.

A pneumatic line **32** connects the accumulator tank **31** with a pilot pressure outlet port **34** of an automatic pitch compensator **36** according to the invention, hereinafter APC **36**.

Three embodiments of the APC are described all serve essentially the same purpose, namely being means to effect automatic pneumatic modification of manually selected pitch signals to the PCC **19**, i.e. of the selected pitch output of the SLC.

A pneumatic line **37** transmits air from the line **17** to an inlet port **38** of the APC **36**. The APC **36** has an exhaust port **39** vented to atmosphere. An engine rpm control device, suitably a fuel rack (not shown) of an engine or a governor output shaft, **251.1**, FIG. **9**, is connected to a coupling means **41** of the APC, so that the APC senses a demand to the engine to change fuel delivery. The throttle control valve **15** is connected through a pneumatic line **42** to an inlet port **43** of a throttle control **44**, hereinafter TC **44**.

The circuit **10** does not provide automatic pitch compensation for reverse pitch. In the neutral pitch position the lever **12** does not actuate the valves **13** or **14** and pressure in the lines **21** and **16** is low and essentially constant. To attain forward pitch from neutral, the valve **14** is actuated and meters air into the forward pitch line **21**, pressure in the reverse pitch line **16** being unchanged. If forward pitch is reduced towards neutral, but not passing into reverse pitch, pressure in the line **21** is reduced and pressure in the line **16** still remains unchanged. The PCC **19** has a neutral position, i.e. neutral pitch, which is attained when there is no pitch signal to the PCC **19** when the SLC **11** is in a neutral pitch position.

If automatic pitch compensation for reverse pitch is required, an additional compensating relay valve (not

shown) is provided in the reverse pitch circuit and receives air at pilot pressure from the accumulator tank similarly to the circuit of the forward pitch line line.

AUTOMATIC PITCH COMPENSATORS

General description of the automatic pitch compensator

The APC **36** FIGS. **1** and **9**, is an assembly of a known compensating air pressure regulating valve **36.1**, hereinafter CAPRV, to a means generally designated **36.2**. The means **36.2** is connected by the coupling means **41** to the governor **251** as seen in FIG. **9** so as to be responsive to change in engine rpm. In some installations governor output is connected directly to a fuel rack, and, in such cases, the coupling means **41** can be connected to the fuel rack, since the fuel rack responds to rpm changes, at constant rack setting.

Hereinafter and in the claims all connections to detect governor response, whether coupled to output from the governor or to the fuel rack or equivalent, are referred to as being coupled to governor output.

The mechanism **36.2** and the coupling **41** are hereinafter referred to as a governor to regulating valve coupling, GRVC, being means transmitting governor response to the CAPRV so changing output pilot pressure at the port **34**. This controls fuel supply, as explained, tending to maintain predetermined rpm.

The three APC embodiments described have respectively mechanical servo and hydraulically actuated GRVC's. Details of each are described below with reference respectively to FIGS. **2** and **3**, **4** and **5**, and **6** and **7**, — operation of each being separately described in a detail description following.

Each APC embodiment includes the known CAPRV and a mechanical servo or hydraulic GRVC, selection of a particular GRVC being dependent on governor output as will be explained.

FIGS. **2** and **3**, mechanical APC

A mechanically actuated automatic pitch compensator **50**, hereinafter APC **50** includes a CAPRV **51** with outlet inlet and exhaust ports **51.1**, **51.2**, **51.3** and a GRVC **52**.

The APC **50** is connected to an engine rpm control device by means generally similar to the means **41** of FIG. **1**, and includes means, namely a hinged lever **55** mechanically linked to the fuel rack of an engine or the governor output shaft. A series of holes **56** is provided at an outer end **57** of the lever **55** to permit matching of characteristics of the hydraulic circuit with characteristics of the engine and governor.

The lever **55** has an outer end **57** and an inner end **61**, and is hinged for rotation on a pin **59**, the end **61** is closer to the pin than the end **57**. The pin **59** is carried in hinge mounts **63** and **65** best seen in FIG. **3**, and screw adjustment means **66** at the end **61** are in contact with an outer end of a pushrod **67** of the APC **50**, the pushrod being urged outwards against the end **61**.

A relatively large movement of the outer end **57** in a direction shown by an arrow **69** results in a comparatively small movement of the screw adjusting means **66**, and with it the pushrod **67**, in a direction shown by an arrow **71**. The pushrod **67** is a metering activation means, movement of the pushrod actuating metering means within the CAPRV **36.1**, the metering means producing a change in pilot pressure from the outlet port **51.1**. In an equilibrium position, at constant fuel flow to the engine, there is no movement of the lever **55**.

The mechanism described including the coupling means is thus means, responsive to governor output, i.e. to change in rpm, to operate the metering activation means **67** of the CAPRV **36.1**, and serves as the mechanical GRVC **52** as aforesaid.

Compensating air pressure regulating valve

The CAPRV **51** functions essentially as the CRV **23** in that a metering means within the valve meters inlet air pressure producing an outlet pressure lower than inlet pressure, excess air being exhausted through the exhaust port. This valve which, as stated, is common to all the APC's described, being well known, is not described in detail. Any type of CAPRV having characteristics above is suitable, namely having means to meter inlet pressure so as to produce a lower outlet pressure. For all the CAPRV's herein, extension of the end of the pushrod **67** decreases pilot pressure and the APC is coupled to the rpm control device, i.e. governor output, so that a demand for more fuel extends the end of the pushrod **67**.

The inlet and outlet ports **51.2** and **51.1** of the CAPRV have common flow control needle valves (not shown), the valves controlling the volume flow of air entering and leaving the APC **50**. Relative settings of the needle valves permit adjustment of response characteristics of the PCC **19**, particularly so as to reduce chance of the engine being overloaded following a sudden increase in load. Commonly the needle valve is a tapered needle axially adjustable of a metering bore, and serves as a means to adjust response of the system.

FIGS. **4** and **5**, servo operated APC

A servo operated automatic pitch compensator **80**, hereinafter APC **80**, has the CAPRV **36.1** substantially as before described and a GRVC designated generally **36.2**. The inlet and outlet ports **38** and **34** are provided with flow control needle valves as before.

The CAPRV **36.1** is operated by a pushrod **86** having a cam follower roller **87** at a lower end as shown, the pushrod **86** being a metering activation means of the CAPRV **36.1**. The roller **87** rolls on a cam face **88** of a cam **89**, the cam **89** being mounted on a left hand end, (as seen in FIG. **4**) of an actuating rod **91**. The rod **91** is aligned with a central axis of a cylinder **92**, the cylinder having a cylindrical inner wall **94**, and an outer end cap **96** at a right hand end. A right hand end wall **97** of a cam housing, generally **98**, provides a left hand closer or inner end wall of the cylinder and the actuating rod **91** is free to slide in a bore **99** passing through the end wall **97**.

A coupling means generally **102**, functions as the means **41** in FIG. **9**, being connected to an engine rpm control device. The means **102** has a cranked lever **103** having an upper arm **104**, an inner end **105** of the arm **104** being secured by pinch bolt means **107** to a rotatable pin **108**. An outer end **106** of the arm **104** is mechanically linked to a fuel rack or governor output shaft of the engine, a series of holes **109** being provided for matching as before. The pin **108** is journalled in bearings **111** and **112** and a lower arm **113** has an inner end **114** secured by screw means **115** (FIG. **5** only) to

the pin **108**. Screw adjusting means **119** are provided at an outer end **118** of the arm **113** and contacts a right hand end **121** of a control rod **122** which rod is free to slide axially in a bore **123** in the end cap **96**.

Undesignated seals are provided to stop material leakage between the rods **91** and **122** and the bores above, and leakage between joins of the cylinder **92** and the end cap and cam housing. Rotation of the upper arm **104** in a direction shown by an arrow **120** results in axial translation of the control rod **122** in the direction shown by an arrow **124**.

Two axial studs **125** and **126** (FIG. **5** only) are used to retain brackets for the bearing housings **111** and **112**, and two further similar studs, (one only being shown designated **127**) are used to secure the right hand end cap to the cylinder. An oil inlet port **131**, integral with the cam housing **98** is connected by hose (not shown) to an oil outlet port from the engine (not shown) and receives engine oil under pressure. An oil outlet port **132** is provided on the right hand end of cap **96** and is connected to an engine oil sump (not shown) by a scavenge hose (not shown).

The control rod **122** has a mid-portion **134** of reduced diameter defined in part by a right hand shoulder **135** and a left hand shoulder **136**, the shoulder **136** being shown in broken outline. A plug **137** is provided at the left hand end of the control rod **122**, the shoulder **136** being a right hand end of the plug. The plug **137** and a portion of the reduced portion **134** are fitted within a blind bore **139** of the actuating rod **91**, sufficient clearance being provided around the plug **137** to permit free sliding of the plug within the bore **139**. A port **141** passes through the rod **91** to the blind bore **139**.

A piston **143** is secured on a right hand end of the rod **91** on a side of the port **141** remote from the cam **89**. Thus the cylinder **92** is divided into two spaces, namely an or left cylinder space **144** defined in part by the piston and end wall **97** and communicating with the inlet port, and an outer space **145** defined in part by a right hand end of the blind bore **139** and the end cap **96**, and communicating with the outlet port **132**. An annular passage **142** (broken outline) is defined by the portion of reduced diameter and the bore **139** and extends between the port **141** and the outer space **145**. The bore **139**, hereinafter the control bore, and the port **141**, hereinafter the control port, communicate the space **144** with the space **145** through the annular passage **142**. In an equilibrium position at constant fuel flow to the engine, engine oil flow through the annular passage **142** is restricted at a left hand end by the plug **137** partially blocking the control port **141**.

A first compression spring **147** is provided in the right hand cylinder space **145** and is compressed between the piston **143** and a left hand face of the right hand end cap **96**. Thus the spring **147** forces the piston and end cap apart. A second compression spring **149** is compressed between the piston **143** and the shoulder **135**. This spring has a truncated-conical section and urges the control rod **122** outwards from the control bore **139**. Spring clips **151** and **152** serving as stops are provided at positions on the control rod **122** to restrict movement of the rod **122**.

A passage (not shown) connects the oil outlet port **132** with the right hand cylinder space **145** to provide means to scavenge oil from the space **145**. A radially disposed bleed passage **154** on the right hand side of the shoulder **136** communicates with an axial bleed passage (not shown) which extends leftwards communicating with the blind end of the bore **139**, the bleed passages above, hereinafter bleed means, are to prevent a hydraulic lock at the blind end of the bore **139**. Other means to prevent hydraulic locks are known. The hydraulic cylinder assembly including the activating and control rods **91**, **122**, the cam **89**, and the coupling means **102** is designated generally **36.2** and is seen to constitute servo means responsive to governor output, i.e. responsive to rpm change, to operate the CAPRV **36.1** as aforesaid, and thus serves as the servo GVRC **36.2**.

FIGS. **6** and **7**, hydraulic APC

A hydraulically-actuated automatic pitch compensator **160** hereinafter APC **160**, includes a CAPRV **161** and a GRVC **162**. The CAPRV has inlet port **163**, an outlet port **164**, and an exhaust port **165**, positions of which are shown in broken outline. For reasons which are later explained, these inlet and outlet ports are not provided with flow control needle valves thus contrasting with the CAPRV **51** (FIG. **2**) and the CAPRV **36.1** (FIG. **4**).

The CAPRV **161** is secured to a cam housing **166** of a cylinder **168**. The cylinder **168** has an end cap **169** at a right hand end and has a left hand end closed by a right hand end wall **171** of the cam housing **166**. The end wall **171** has a bore **172** in which an actuating rod **174** is a sliding fit. The rod **174** has an outer end **175** to which a piston **176** is secured, and an inner end **177** to which a cam **178** is secured. The piston **176** divides the cylinder **168** into two spaces, a left hand cylinder space **187** and a right hand cylinder space **188**. The cam **178** has a cam follower **179** on an end **181** of a push rod of the CAPRV **161**, the cam and operation of the CAPRV **161** being similar to that as described with reference to FIGS. **2** and **4**. Thus movement of the piston in a direction shown by an arrow **180** permits the cam follower to extend from the CAPRV **161** in response to a demand to decrease pitch by decreasing pilot pressure.

The end cap **169** is secured to the cylinder **168** by studs, two being designated **182** and **183**. Ports **185** and **186** communicate with pressure output ports from an engine governor (not shown) through hydraulic hoses (not shown) and, in an equilibrium position at constant fuel flow delivery, the output ports of the governor are closed and there is no fluid flow to or from the cylinder **168**, the piston **176** being hydraulically locked within the cylinder. Output connections from the governor are such that, from an equilibrium position, a demand from the governor to decrease fuel flow to the engine results in additional oil being fed into the port **185** into the space **187**, the piston moving to the right and scavenging oil from the space **188**, returning it to the governor through the port **186**. For an increase in fuel to the engine, flow directions are reversed, the piston then moving to the left in the direction of the arrow **180**.

Governor oil pressure and cross-sectional area of the piston **176** are such that pressure difference across the piston is sufficient to move the actuating shaft **174** against resistances inherent in pistons and cylinders.

The mechanism above described, including the cylinder, piston rod, piston, and cam and hydraulic

hoses (not shown) to the governor, is designated generally **162** in FIG. **6**. The mechanism is thus seen to be hydraulic means responsive to difference in pressure, i.e., to rpm change, to operate the CAPRV **161** as aforesaid, and thus serves as the hydraulic GRVC **162**.

FIG. **8** and reference to FIG. **6**

A biased flow needle valve **221** is fitted to the port **185** of the hydraulically actuated APC **160**, FIG. **6**, to provide a means of rapid scavenging of the space **187**, reducing chance of overloading the engine by increasing speed of response of compensation. The port **185** FIG. **8** has a threaded counter-bored passage **222** to accept a coupling at one end of the hydraulic hose from the governor, fluid from the bore entering the valve **221** in a direction shown by an arrow **223**. A metering bore **225** extends from the passage **222** and has a metering port **226** as shown communicating with a threaded passage **224**. A needle valve stem **227** threaded in the passage **224** can be turned by a knurled end **229**, and has a conical end **228** which cooperates with the port **226** so that axial movement of the end **228** controls volume flow of fluid through the port **226**. An infeed passage **231** extends from the metering port **226**, which when the port **226** is open, feeds fluid into the APC **60** in a direction shown by an arrow **232**.

A ball valve **234** is provided to permit a rapid drop of pressure in the CAPRV **161** (FIG. **6**) to reduce chance of overloading should a sudden increase in load on the engine occur. The valve **234** has a ball **235** urged against a valve seat **236** by a spring **237** augmented by fluid pressure within the bore **225**. Thus, if fluid pressure within the space **187** exerts a force on the ball **235** less than the force of the spring **237** plus fluid pressure within the bore **225**, the valve remains closed.

A rapid increase in load on the engine causes the governor to make a sudden demand to increase fuel. With reference to FIG. **6**, fluid is fed rapidly into the port **186** with a corresponding rapid increase in pressure in the space **187**. There is a sudden force on the ball **235** urging it to the right, which overcomes the force of the spring **237** and pressure in the bore **225**, moving the ball in a direction shown by the arrow **239**. Fluid from the space **187** is exhausted mainly through the valve **234**, a portion passing the needle valve **221**. Passage of fluid in a direction shown by the arrow **239** is several orders of magnitude greater than that through the needle valve **221**, providing relatively rapid response of the APC **160** to a sudden demand for more fuel. This results in correspondingly rapid pitch reduction.

Thus the needle valve and ball valve assembly, designated generally **240** in FIG. **8**, serve as a means of increasing speed of response of the APC in one direction, namely in the direction of pitch reduction, to produce a rapid change in pitch of the propeller to reduce load. Providing the valve **221** in the port **185** produces a more rapid response of the APC than would be obtained by fitting needle valves in the ports **163** and **164** of the CAPRV **161**. Thus the APC **160** more effectively safeguards the engine against overload In practice a second biased flow needle valve similar to the valve **221** can be fitted to the port **186** for simplicity of installation of the component.

Selection of APC

The APC **36** shown in FIGS. **1** and **9** is similar to the APC **80** described with reference to FIGS. **4** and **5**. Selection of a particular APC is dependent on governor output characteristics which themselves are dependent on type of governor used and size of the engine being governed. Thus, depending on the governor, the GRVC selected can be that as shown in the APC **50** in FIG. **2** or as the APC **160** as described in FIG. **6**.

Selection of Governor

A type of governor as used in diesel/electric locomotives and manufactured by Woodward or Regulator Europa has been found satisfactory for use with the circuit of FIG. **1**. A Woodward PGL governor has been found adequate when the governor output is coupled conventionally to the engine and also to the coupling means of the APC. The governor output is one of two types: an output shaft which rotates through **30** to **40** degrees from minimum to maximum speed settings and is controlled by the governor input, or can be two ports which provide a fluid pressure difference. The outputs above are connected to fuel delivery means of the engine, usually the fuel rack. With the first type of governor output, with no demand to change fuel delivery, the output shaft is stationary. With the second type of governor, with no demand to change fuel delivery, fluid flow to and from the governor ports is zero.

The APC **50** is used only with governors having a relatively high torque at the output shaft, as rotation of the lever **55** results in work to overcome resistances. Some governors have an output shaft torque insufficient to operate the APC **50** without reducing effectiveness of the governor; these require the APC **80**, wherein the lever **103** takes relatively little work from the governor due to servo-action of the control rod **122** and the actuating rod **91**, as below described. In such a mechanism a small force on the control rod **122**, which acts as a master controls a larger force on the actuating rod **91**, the slave. When using hydraulic pressure outputs of a governor, the APC **160** is described in FIG. **6** is used. Different sizes of piston can be used to accomodate relative pressure differences encountered with this type of governor.

Thus governor output determines selection of APC so that the GRVC thereof is compatible with governor output.

OPERATION

FIGS. **1** and **9**, pneumatic circuit generally

FIG. **9** shows the FIG. **1** schematic hydraulic circuit with relevant components diagrammatically shown operatively connected to an engine governor and to a variable pitch propeller of a vessel.

With prior art circuits without automatic pitch compensation, the SLC **11** controls pitch and throttle concurrently. With the vessel stationary and the engine idling, the propeller is at neutral pitch. A manually imparted signal to the SLC **11** by movement of the lever **12** adjusts relative metering of air through the control valves **13**, **14**, and **15**. From neutral pitch rotation of the lever increases pitch to approximately forty percent of full pitch, whilst the engine remains at idle. With further rotation of the lever **12**, engine rpm increases concurrently with pitch until approximately eighty percent of maximum is attained. Still further rotation of the lever **12** increases engine rpm to one hundred per-

cent, the pitch remaining at one hundred percent. With an automatic reverse pitch (not shown) the ratios above apply to the reverse pitch condition also, reverse pitch being accomplished by rotating the lever in an opposite direction. If a two lever control were used, as stated previously, pitch and throttle are controlled independently.

Using an automatic pitch compensated circuit according to the invention, if there is no demand from the governor for a change in fuel the coupling means **41** of the APC **36** is stationary and the APC **36** is in an equilibrium condition such that output pilot pressure from the APC **36** is constant.

With a vessel cruising at a speed and engine rpm both below maximum, an increase in load on the engine produces a signal from the governor to increase fuel delivery to the engine to maintain pre-set engine rpm. Without pitch compensation by the APC **36** an increase in load on the engine would produce too coarse a pitch condition causing an over-fueling of the engine, with the engine trying to accelerate in response to the over-fueling. The engine would thus labor undesirably for a period dependent on acceleration of the vessel.

With automatic pitch compensation, a typical sequence of events to correct a condition of too coarse pitch is as follows:

1. The propeller, running at too coarse a pitch, causes the engine rpm to decrease below the rpm pre-set by the governor.

2. The governor rapidly responds by moving the fuel rack to increase fuel delivery to the engine, to increase rpm to attain that pre-set by the governor. This action is rapid, decrease of a few revolutions per minute suffices to cause the sequence above.

3. Movement of the fuel rack to increase fuel actuates the APC **36**, which compensates for too coarse a pitch by decreasing the pitch.

4. A decreased pitch at the increased fuel delivery results in a small increase in rpm in increase less than the decrease in (1) above.

5. Increase in rpm is rapidly corrected by the governor, which reduces fuel delivery by a small amount, a smaller decrease than the increase in (2) above.

6. Decrease in fuel delivery causes a small increase in pitch, much smaller than the decrease in pitch above.

7. Thus by a series of successive approximations and minor overcorrections, the pitch of the propeller is adjusted to maintain essentially constant fuel delivery to the engine.

The above sequence of events occupies a few seconds, or less depending on the compensation required. Fluctuation of the rpm from the pre-set rpm is typically in a range of one or two revolutions per minute, depending on response of the governor. The return to the pre-set rpm can be likened to an asymptotic approach to an equilibrium position of a dampened vibrating system.

Maximum compensation of pitch occurs when the pilot pressure from the APC is at its lowest namely pressure at the port **29** of the CRV **23** being about one half of maximum value of pilot pressure, as read at gauge. This is termed 50 percent compensation. The maximum compensation above is rarely reached, a more usual compensation being about 35 percent, which generally occurs at about 80 percent engine rpm.

In the following description of operation, only the initial change in pitch is described, not the successive approximations above.

FIGS. **2** and **3**, alternative

A demand from the governor to change fuel delivery rotates the lever **55** about the pin **59** so that the end of the pushrod **67** moves in or out of the CAPRV **51** of the APC **50**.

If there is a sudden increase in load on the engine, the governor responds to increase fuel to maintain pre-set rpm, so the fuel rack moves to increase fuel delivery and also rotates the lever **55** in the direction shown by the arrow **69**. The outlet pressure (i.e. pilot pressure) from the APC **50** is thus decreased and, is fed to the CRV **23** via the accumulator tank **31**. A decrease in pilot pressure decreases output pressure from the CRV **23** and the PCC **19** experiences a drop in forward pitch pressure, this reduces forward pitch. The demand to reduce forward pitch is experienced by the PCC **19** without additional movement of the lever **12**. Thus reduction in pitch is fully automatic, the APC having sensed a demand for more fuel to maintain the rpm at that pre-set by the governor.

Load compensation for the APC **50** is about ninety percent, so that the engine tends to be somewhat overloaded with any sudden increase in load. Drop in power output from the engine due to, for instance, failure of perhaps two out of six cylinders of the engine causes the fuel rack to move to a new position to supply more fuel in an attempt to maintain the pre-set rpm. Such new position will result in over fueling the remaining operative cylinders, somewhat tending to overload the engine.

FIGS. **4** and **5**, alternative

At a constant fuel delivery, the APC **81** is in an equilibrium position and the lever **102** is stationary. Lubricating oil from the engine is fed into the port **131** essentially under constant pressure independent of engine rpm, and produces a force to the right on the piston **143**, which force is counteracted by fluid forces in the cylinder **145** combined with forces from the springs **147** and **149**.

Oil flows into the left hand cylinder **144** and leaves the cylinder through the control port **141**, flowing along the annular passage **142** into the space **145**. The right hand end of the plug **137** partially blocks the control port **141** and meters oil flow from the cylinder space **144**, thus reducing oil pressure in the space **145** from that in the space **144**. Relative positions of the plug **137** and the bore **141** are dependent mainly on viscosity and pressure of the oil, spring forces, and diameter of the piston. When there is constant fuel delivery to the engine, relative positions of the plug **137** and the control port **141** do not change being at an equilibrium position thus under this condition the piston does not move within the cylinder.

A sudden increase in load on the engine produces a demand from the governor to increase fuel, the fuel rack moving to increase the supply of fuel to the engine. The upper arm **104** of the coupling means **102**, being connected to the fuel rack, moves in the direction of the arrow **120**, the screw adjusting means **119** pushing the control rod **122** to the left into the cylinder. Thus the plug **137** is moved to the left, reducing the restriction of flow at the control port **141** and

increasing flow of oil from the space **144** through the annular passage **142** into the the right hand space **145**; oil pressure in the space **145** being higher than before movement of the plug because of less metering at the port **141**.

An increase in pressure in the space **145** forces the piston to the left, and, as the plug **137** remains stationary relative to the APC **80**, the port **141** again becomes restricted by the plug **137**. Increased restriction of the control port **141** reduces pressure in the space **145** and movement of the piston **143** ceases, the piston attaining a new equilibrium position somewhat to the left of the original position until there is further movement of the coupling means **102**.

Movement of the piston **143** to the left moves the cam **89**, to the left, causing the cam follower **87** to roll down the cam face **88**, with the pushrod end **86** moving outwards of the CAPRV **81**. Effect of the metering means within the CAPRV **81** is thus changed, extension of the end of the pushrod **86** reducing pilot pressure which in turn modifies the demand on the forward pitch port PCC **19** to a reduced forward pitch of the propeller.

A sudden decrease in load on the engine, results in a demand from the governor for less fuel to be delivered to the engine, resulting in movement of the coupling means **102** in the direction opposite to that of arrow **120**, thus moving the screw adjusting means **119** away from the end **121** of the rod **122**. The spring **149** urges the rod **122** outwards and thus the plug **137** increases metering at the port **141**, which metering decreases the pressure in the space **145** relative to the space **144**. Thus the piston **143** moves to the right, such movement of the piston decreasing metering at the port **141**. This increases pressure in the space **145** and stops the movement of the piston **143**, thus it attains a new equilibrium position to the right of the previous equilibrium position. This new equilibrium position is maintained until there is a further change in fuel delivery to the engine.

Movement of the piston **143** to the right results in a corresponding movement of the cam **89**, forcing the cam follower upwards and, with it, the end of the pushrod **86** into the CAPRV **81**, increasing pilot pressure output from the APC. This results in an increase in output pressure from the CRV **23** increasing the original forward pitch and reducing rpm of the engine.

FIGS. **6** and **7**, alternative

With reference to the operation of the APC **160**, in an overload condition, a demand from the governor to increase fuel supply to the engine causes fluid from the governor output to be fed into the port **186**. Fluid is scavenged from the port **185** as the piston **176** is forced to the left, and the pushrod is extended from the CAPRV **161**, reducing pilot pressure and therefore reducing forward pitch.

An opposite demand from the governor to decrease fuel to the engine results also in increased pitch as follows. Fluid from the governor output is fed into the port **185** and scavenged from the port **186**, thus resulting in movement of the piston **176** to the right, this causes the cam follower **179** to move the pushrod end into the CAPRV **161**, increasing pilot pressure. Thus there is a signal from CRV **23** to the PCC **19** to increase pitch.

Response of the APC **160** is dependent on many factors and is adjustable by means of the biased flow needle valves fitted at the ports **185** and **186**, operation of which are described with reference to FIG. **8**.

FIG. **8**

When setting up a circuit using the APC **160**, the governor output characteristics are matched to the engine characteristics and response of the CAPRV **161** by the needle valves incorporated into the outlet and inlet ports of the APC **160**.

Operation of the biased flow needle valve **221** at the port **185** of the APC **161** is as follows, operation of the second biased flow needle valve being within this description. Fluid from the governor output enters the inlet port **51** thorugh the metering bore **225** and volume of flow is controlled by position of the conical end **228** of the needle valve stem **227** relative to the metering port **226**. Metered fluid from the port **226** passes through the infeed passage **231** in the direction shown by the arrow **232** into the space **187** (FIG. **6**). The overload ball valve **234** is closed as fluid pressure difference between the fluid in the space **187** and the fluid in the bore **225** is below a predetermined value.

A sudden increase in the pressure difference above may exceed the predetermined value above which results in opening of the valve **234**, fluid from the space **187** passing the ball **235** into the metering bore **225**. Other factors being constant, fluid flow through the ball valve **234** is several orders of magnitude greater than the flow through infeed passage **231**, permitting relatively rapid decrease in fluid pressure in the space **187** increasing response of APC **160** by increasing speed of exhaust of the cylinder spaces.

FIG. **1** General summary of operation

Thus summarizing the operation of the three embodiments of the APC's described, a change in fuel delivery disturbs the equilibrium position of the APC which adjusts the metering means within the CAPRV of the APC. Thus pilot pressure is changed, which pressure affects the compensation of the output pressure from the CRV **23**. Degree of compensation of the output pressure from the CRV **23** controls final pitch selected by the PCC **19**.

I claim:

1. An automatic pitch compensator adapted for the use in a pneumatically operated control circuit in a marine vessel having an engine, rpm of the engine being selected by a speed setting of a governor, the governor having an output reflecting governor response coupled to an engine rpm control device to regulate engine rpm, the engine powering a controllable pitch propeller; the automatic pitch compensator including:

a. a compensating air pressure regulating valve (CAPRV) with an inlet port, an outlet port and an exhaust port, the valve having a metering means adapted to meter air under pressure received in the inlet port to a pilot pressure at the output port, the metering means being operably by a metering activation means,

b. a governor to regulating valve coupling (GRVC) connecting the metering activation means of the compensating air pressure regulating valve to the governor output so as to transmit governor response to the metering means,

so that change in engine rpm from governor set rpm is reflected in governor output, which change is transmitted to the metering activation means of the compensating air pressure regulating valve, so that metering of air pressure fed into the input port of the compensating air pressure regulating valve is changed so as to change pilot pressure in an amount proportional to change in engine rpm from governor set rpm.

2. An automatic pitch compensator as defined in claim 1 adapted for use with a governor having a mechanical output, in which the governor to regulating valve coupling includes:

c. a coupling means operatively connected to the governor output,

d. a servo means responsive to governor output to transmit governor response to the metering activation means, to operate the compensating air pressure regulating valve.

3. An automatic pitch compensator as defined in claim 2, in which the servo means includes:

e. a cylinder having inner and outer end walls, an oil inlet port adapted to receive oil under pressure, and an oil outlet port adapted to return oil to a sump,

f. a control rod slidable within the cylinder, the control rod having an outer end in engagement with the coupling means to receive governor response, an inner end having a plug, and a portion of reduced diameter,

g. an actuating rod aligned with, and adapted to slide axially relative to, the control rod, the actuating rod having an inner end cooperating with the metering activation means of the compensating air pressure regulating valve so that axial sliding of the actuating rod relative to the cylinder changes output from the compensating air pressure regulating valve, the actuating rod having an outer end having a control bore accepting the plug at the inner the control rod, the actuating rod having an inner end cooperating with the metering activation means of the compensating air pressure regulating valve so that axial sliding of the actuating rod relative to the cylinder changes output from the compensating air pressure regulating valve, the actuating rod having an end of the control rod, and a control port communicating with the control bore, an annular passage being defined by the portion of reduced diameter and the control bore and extending between the control port and the outer end of the actuating rod, the actuating rod further including a bleed means to scavenge oil from the control bore to prevent a hydraulic lock forming within the control bore,

h. a piston secured adjacent the outer end of the actuating rod on a side of the control port remote from the inner end of the actuating rod, the piston dividing the cylinder into inner and outer cylinder spaces, the inlet port communicating with the inner cylinder space and the outlet port communicating with the outer cylinder space,

i. a first compression spring extending between the piston and the outer end wall of the cylinder so as to urge the piston away from the cylinder,

j. a second compression spring extending between the piston and the control rod so as to urge the inner end of the control rod away from the control bore of the actuating rod,

relative disposition of the plug and the control bore being such that the plug partially blocks the control port, so that oil leaving the inner cylinder space passes through the control port, is metered by restriction at the control port, and passes at a reduced pressure along the annular passage into the outer cylinder space to leave the outer cylinder space through the outlet port, the oil in the outer cylinder space producing a force on the piston to combine with the spring forces in opposition to force from oil pressure in the inner cylinder space, so that in an equilibrium position force on the piston from oil in the inner cylinder equals force on the piston from oil in the outer cylinder plus force from the springs so that the piston is stationary relative to the cylinder; and movement of the coupling means resulting from a change in engine load moves the control rod relative to the actuating rod thus changing the metering of the oil flow and producing a force difference on the piston thus disturbing the equilibrium and moving the piston in a direction to regain the equilibrium position, movement of the piston moving the actuating rod axially so as to change position of the metering activation means thus altering pilot pressure at the output port in an amount proportional to change in engine load.

4. An automatic pitch compensator as defined in claim 3 in which the coupling means includes:

k. a lever having an upper arm adapted to be linked to the output of the governor, and a lower arm, the lever being hinged for rotation relative to the automatic pitch compensator,

l. screw adjusting means provided at the outer end of the lower arm, the means contacting an outer end of the control rod so as to transmit governor response to the control rod and to provide adjustment of the coupling means.

5. An automatic pitch compensator as defined in claim 3 in which:

m. the mid portion of reduced diameter is defined in part by a shoulder adjacent the inner end, the shoulder adapted to serve as a metering means when adjacent the control port.

6. An automatic pitch compensator as defined in claim 2 in which:

n. inlet and outlet ports of the compensating air pressure regulating valve includes flow control needle valves to permit adjustment of response of the automatic pitch compensator.

7. A pneumatically operated control circuit adapted for use in a marine vessel powered by an engine, rpm of the engine being selected by a speed setting of a governor, the governor having an output reflecting governor response coupled to an engine rpm control device to regulate engine rpm, the engine powering a controllable pitch propeller, the pneumatic circuit including:

a. a manual control (SLC) receiving air under pressure from a source, the control having valves communicating with a forward pitch line, a reverse pitch line and a throttle line, the valves being operable by an operator so as to selectively meter outputs to each line,

b. a pitch control cylinder (PCC) having a forward pitch port and a reverse pitch port, output from the pitch control cylinder being operatively con-

nected to a pitch changing mechanism of the controllable pitch propeller, the reverse pitch line communicating with the reverse pitch port,

c. a throttle control (TC) having an inlet port connected to the throttle line, and an output connected so as to control the governor speed setting,

d. a compensating relay valve (CRV) having an inlet port, an outlet port, and a pilot pressure input port, the compensating relay valve having a metering means provided between the inlet port and the outlet port, the metering means being activated by pilot pressure at the input port, the inlet port communicating with the forward pitch line and the outlet port communicating through a line with the forward pitch port of the pitch control cylinder,

e. an automatic pitch compensator (APC) having an inlet port accepting air under pressure from the supply, an outlet port discharging air at pilot pressure to the pilot pressure input port of the compensating relay valve, and an exhaust pipe to exhaust excess air to waste, the automatic pitch compensator having

i. a compensating air pressure regulating valve (CAPRV) having a metering means between the inlet and outlet ports, and a metering activation means actuating the metering means,

ii. a governor to regulating valve coupling (GRVC) connecting the metering activation means to the governor output so as to transmit governor response to the compensating air pressure regulating valve so as to change output pilot pressure at the outlet port, the governor response being dependent on load in the engine,

f. an accumulator tank in series with the outlet port of the automatic pitch compensator and the input port of the compensating relay valve,

so that a pitch signal of a manually imparted signal at the manual control to attain a perdetermined rpm and pitch is modified by a demand from the governor output dependent on load experienced by the engine, the modification altering the manually selected pitch so as to tend to maintain constant rpm.

8. A pneumatically operated control circuit as defined in claim **7**, in which the governor has a mechanical output and the governor to regulating valve coupling includes:

g. a coupling means cooperatively connected to the governor output,

h. a servo responsive to governor output to transmit governor response to the metering activation means to operate the compensating air pressure regulating valve.

9. A pneumatically operated circuit as defined in claim **8** in which the servo means includes:

i. a cylinder having inner and outer end walls, an oil inlet port adapted to receive oil under pressure, and an oil outlet port adapted to return oil to a sump,

j. a control rod slidable within the cylinder, the control rod having an outer end in engagement with the coupling means to receive governor response, an inner end having a plug, and a portion of reduced diameter,

k. an actuating rod aligned with, and adapted to slide axially relative to, the control rod, the actuating rod having an inner end cooperating with the metering activation means of the compensating air pressure regulating valve so that axial sliding of the actuating rod relative to the cylinder changes output from the compensating air pressure regulating valve, the actuating rod having an outer end having a control bore accepting the plug at the inner end of the control rod, and a control port communicating with the control bore, an annular passage being defined by the portion of reduced diameter and the control bore and extending between the control port and the outer end of the actuating rod, the actuating rod further including a bleed means to scavenge oil from the control bore to prevent a hydraulic lock forming within the control bore,

l. a piston secured adjacent the outer end of the actuating rod on a side of the control port remote from the inner end of the actuating rod, the piston dividing the cylinder into inner and outer cylinder spaces, the inlet port communicating with the inner cylinder space and the outlet port communicating with the outer cylinder space,

m. a first compression spring extending between the piston and the outer end wall of the cylinder so as to urge the piston away from the cylinder,

n. a second compression spring extending between the piston and the control rod so as to urge the inner end of the control rod away from the central bore of the actuating rod,

relative disposition of the plug and the control bore being such that the plug partially blocks the control port, so that oil leaving the inner cylinder space passes through the control port, is metered by restriction at the control port, and passes at a reduced pressure along the annular passage into the outer cylinder space to leave the outer cylinder space through the outlet port, the oil in the outer cylinder space producing a force on the piston to combine with the spring forces in opposition to force from oil pressure in the inner cylinder space, so that in an equilibrium position force on the piston from oil in the inner cylinder equals force on the piston from oil in the outer cylinder plus force from the springs so that the piston is stationary relative to the cylinder; and movement of the coupling means resulting from a change in engine load moves the control rod relative to the actuating rod thus changing the metering of the oil flow and producing a force difference on the piston thus disturbing the equilibrium and moving the piston in a direction to regain the equilibrium position, movement of the piston moving the actuating rod axially so as to change position of the metering activation means thus altering pilot pressure at the output port in an amount proportional to change in engine load.

10. An automatic pitch compensator as defined in claim **1** in which the governor has a mechanical output and in which the governor to regulating valve coupling includes:

o. a lever having an outer end adapted to be linked to the output of the governor and an inner end contacting and metering activation means of the compensating air pressure regulating valve, the lever being hinged for rotation relative to the automatic pitch compensator so that movement of the governor output is transmitted to the metering activation means.

11. An automatic pitch compensator as defined in claim 1 adapted for use with a governor having hydraulic output ports in which the governor to regulating valve coupling includes a hydraulic means responsive to difference in pressure in the governor outputs, the hydraulic means including:

p. a cylinder having closed ends, a port being provided adjacent each end with each port communicating with a respective port of the governor outputs,

q. an actuating rod having inner and outer ends, the rod being a sliding fit in a bore at one end of the cylinder, the inner end of the rod cooperating with the metering activation means of the compensating air pressure regulating valve,

r. a piston provided at the outer end of the actuating rod, the piston dividing the cylinder into cylinder spaced on either the side of the piston, a port communicating with each space, so that, in response to governor demand fluid flowing out of one governor output enters one port into one cylinder space and displaces the piston within the cylinder, fluid being scavenged from the other cylinder space through the other port, movement of the piston actuating the metering activation means of the compensating air pressure regulating valve so as to change pilot pressure in an amount proportional to governor response.

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