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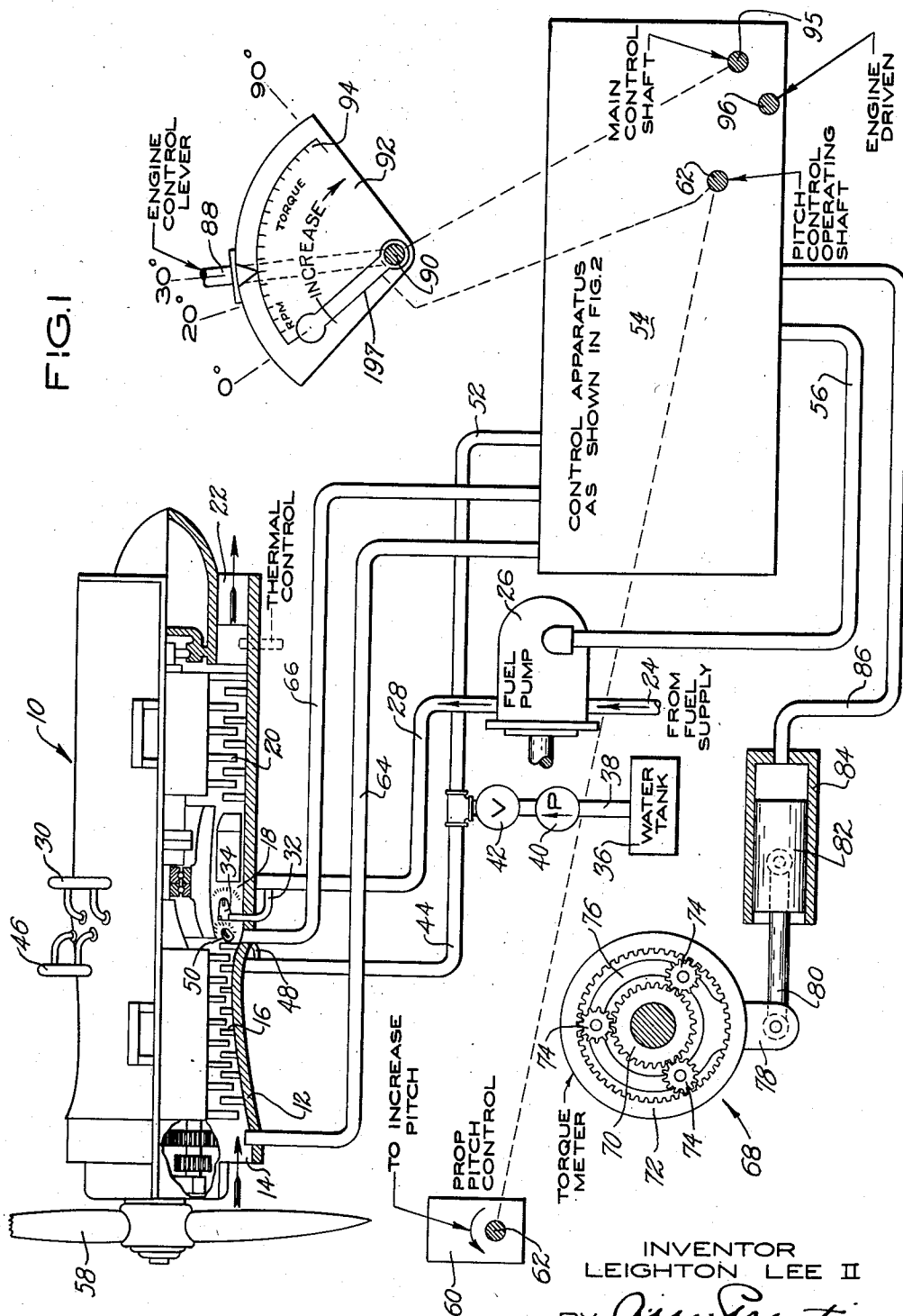
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2,657,530

CONTROL APPARATUS FOR TURBOJET ENGINES

Filed Nov. 21, 1947

4 Sheets-Sheet 1



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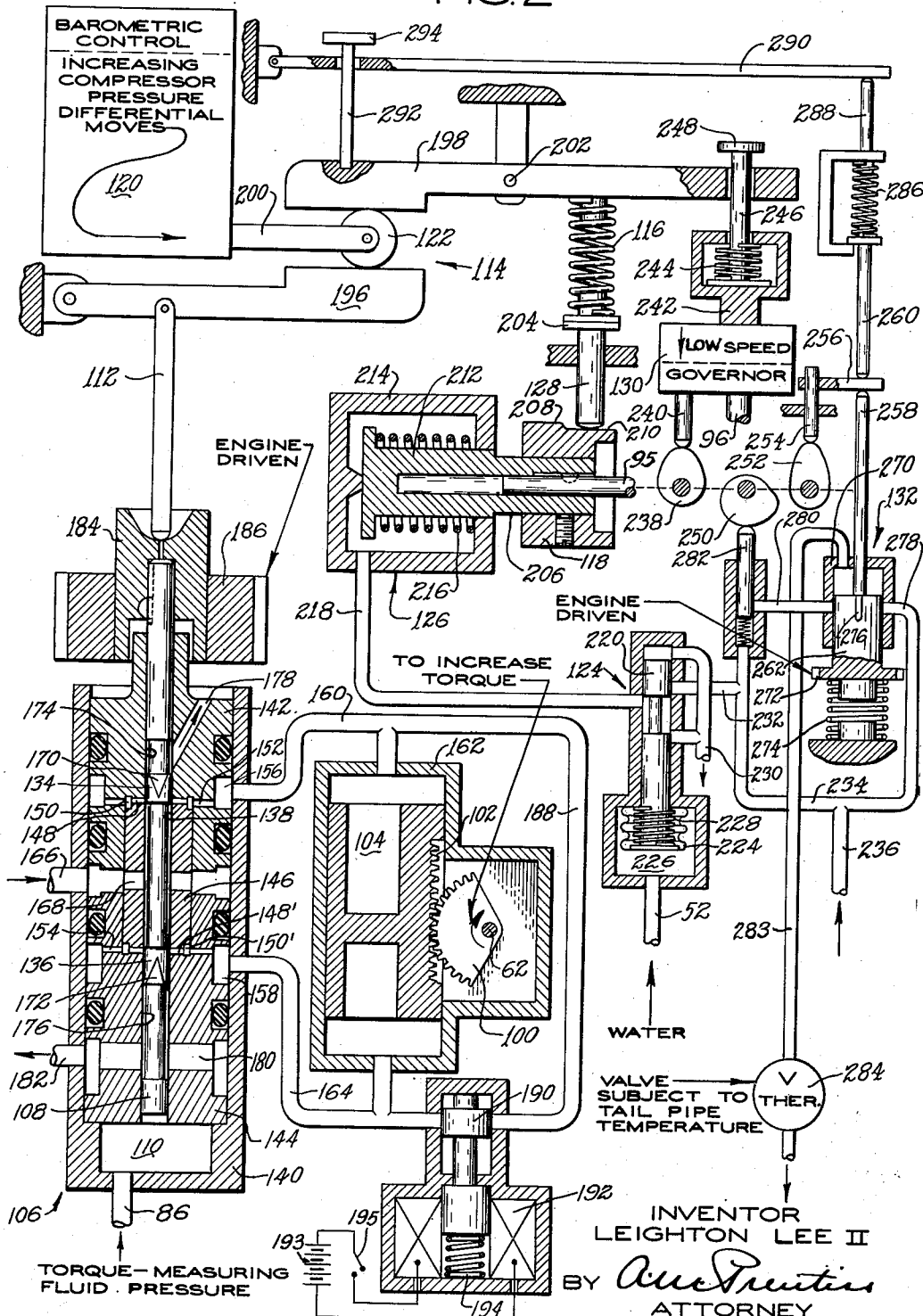
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CONTROL APPARATUS FOR TURBOJET ENGINES

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FIG. 2



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CONTROL APPARATUS FOR TURBOJET ENGINES

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FIG. 6

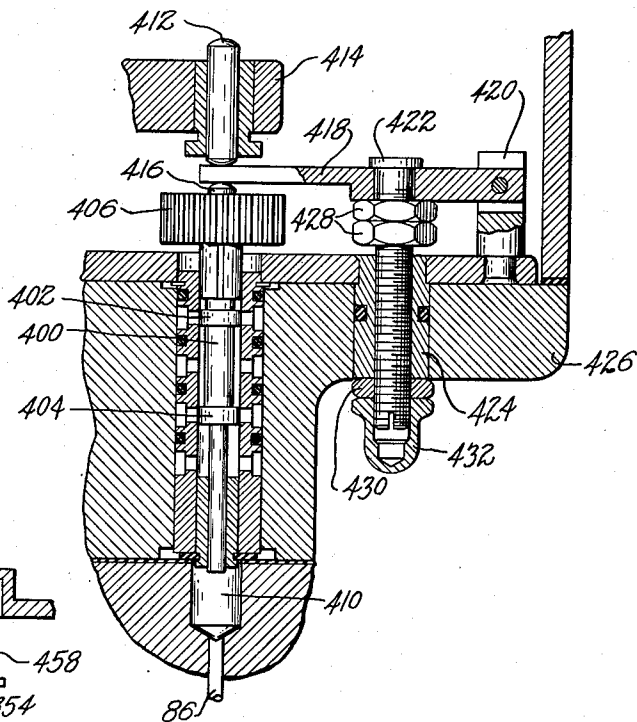
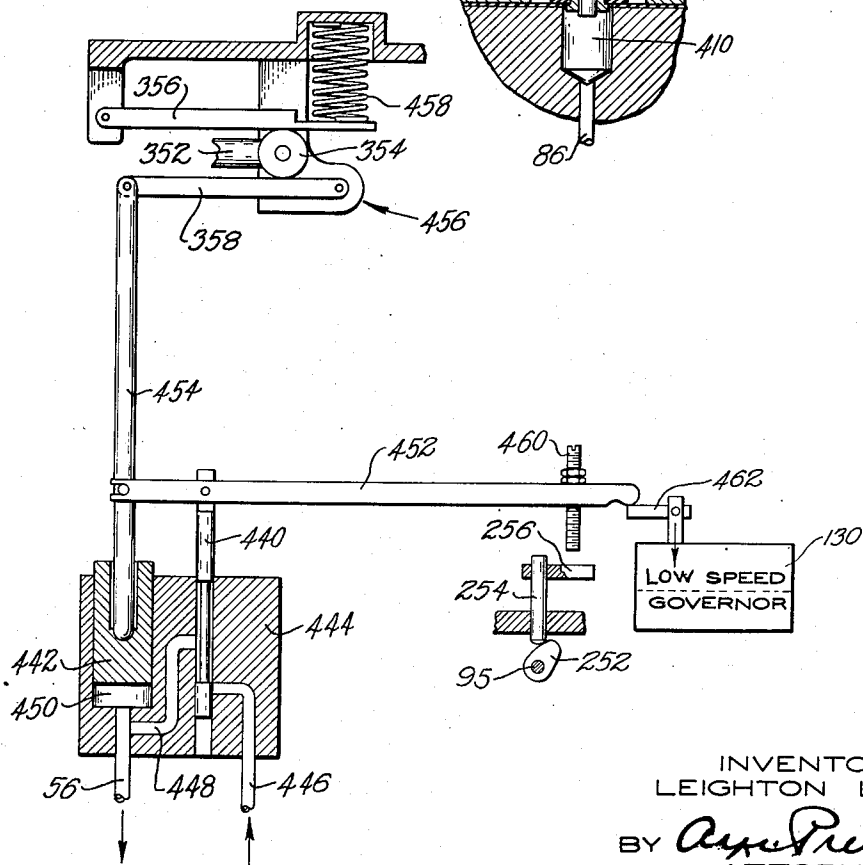


FIG. 7



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2,657,530

CONTROL APPARATUS FOR TURBOJET
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Application November 21, 1947, Serial No. 787,416

11 Claims. (Cl. 60—39.28)

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The present invention relates to hydraulic control apparatus. It is illustrated as applied to an internal combustion engine of the turbine type, certain features of the invention are of utility in connection with control apparatus generally, while others are useful only in hydraulic control apparatus, and still others are useful only in connection with control systems for internal combustion engines.

The particular embodiments of my invention described herein are intended for the control of the fuel flow, speed, and output torque of an internal combustion turbine adapted to drive an aircraft by means of a discharge jet and a variable pitch propeller.

The present invention is intended as an improvement of the apparatus shown and claimed in my copending application Serial No. 762,038, filed July 19, 1947. In that prior application, I have shown a control system in which the fuel flow and torque load of an internal combustion engine are controlled in response to the rate of air flow through the engine, the engine speed, the engine temperature, and the position of a manually movable control device.

A general object of the present invention is to improve and simplify the propeller pitch or torque control illustrated in my said prior application.

Other and more specific objects are: (1), to provide for manual control of the propeller pitch in emergencies; (2), to provide compensation for the rate of change of the engine torque to prevent hunting of the control system; and (3), to provide compensation for the operation of water injection or other combustion modifying apparatus.

Another general object is to provide, in a control system of the type described, improved response to the rate of flow of air through the engine and to the air density.

Still another general object is to provide improved hydraulic control apparatus. More specific objects in connection with this general object are to provide improved emergency manual control mechanism, improved mechanism responsive to the rate of change of a controlling condition, and improved valve mechanism for controlling a hydraulic servomotor.

Other objects and advantages of the present invention will become apparent from a consideration of the appended specification, claims and drawings, in which

Figure 1 is a schematic drawing of a complete control system embodying my invention,

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Figure 2 is a somewhat diagrammatic illustration of a propeller torque control system which forms a part of the system of Figure 1,

Figure 3 illustrates a pressure responsive control mechanism which forms a part of the system of Figure 1, and shows one form which the barometric control of Figure 2 may take.

Figures 4 and 5 illustrate modified forms of barometric controls which may be substituted for the barometric control of Figure 3,

Figure 6 illustrates a modified form of valve mechanism which may be substituted for one of the valve mechanisms of Figure 2, and

Figure 7 illustrates the fuel supply control system which forms a part of the complete system of Figure 1.

Referring to the drawings, the internal combustion turbine engine is generally indicated at 10, and includes a casing 12, an air inlet 14, a multiple stage compressor 16, a series of combustion chambers 18, a multiple stage turbine 20, and discharge pipe or tail pipe 22.

Fuel is supplied to the engine from a tank (not shown) through a conduit 24, a pump 26, and a conduit 28 leading to a fuel manifold 30, from which a series of distributing conduits 32 lead to a number of fuel nozzles 34 located in the combustion chamber 18.

Provision is also made for supplying water, a water-alcohol mixture, or other suitable combustion modifying liquid to the combustion chambers 18. The water supplying mechanism includes a tank 36, from which the water flows through a conduit 38, a pump 40, a valve 42, and a conduit 44 to a water manifold 46, from which a number of distributor conduits 48 lead to individual water nozzles 50 in the combustion chambers 18. The valve 42 may be provided, for the purposes of the present invention, with any suitable manual or automatic control. A branch conduit 52 conveys the pressure of the water being delivered to the water nozzles to the control apparatus shown diagrammatically at 54 in Figure 1 and described in greater detail in Figure 2.

The fuel pump 26 is of the variable delivery type, and its delivery is controlled by the pressure conveyed to it from control apparatus 54 through a conduit 56.

The engine 10 drives, through a suitable reduction gearing, a propeller 58, whose pitch is controlled by a suitable mechanism, indicated at 60, operated by shaft 62 of control apparatus 54.

Conduits 64 and 66 convey the pressures exist-

ing at the inlet and outlet, respectively, of the compressor to the control apparatus 54.

A torque meter generally indicated at 68 supplies to the control apparatus 54 a pressure which measures the torque load on the engine. This torque meter comprises a planetary gear mechanism which forms parts of the reduction gear between the engine and the propeller. This planetary gear includes a sun gear 70, a ring gear 72, and planet gears 74 carried by a spider 76. The sun gear 70 and the ring gear 72 are connected to the engine and to the propeller. The spider 76 has an arm 78 connected through a rod 80 to a piston 82, which moves in a cylinder 84. The reaction on the spider 76 due to the torque load of the propeller is transmitted by the piston 82 to suitable fluid in the cylinder 84 and is communicated through the conduit 86 to the control apparatus 54.

The pilot of the aircraft is provided with an engine control lever 88 fixed on a shaft 90 and movable with respect to a control quadrant 92. The quadrant 92 is preferably provided with a scale 94, illustrated as being marked in degrees. The shaft 90 is connected through a suitable linkage to a shaft 95 in the control apparatus 54. The control apparatus 54 is also provided with an engine driven shaft 96, which supplies power to it for control purposes.

The torque meter 68 is shown diagrammatically only. A more complete description of another type of torque meter which may alternatively be used is found in my copending application Serial No. 654,979, filed March 16, 1946, entitled Indicating Apparatus, now Patent No. 2,450,835.

Figure 2

There is shown in Figure 2 the torque regulating mechanism in the control apparatus 54. This mechanism regulates the torque by controlling the pitch of the variable pitch propeller driven by the engine.

There is shown in Figure 2 a shaft 62, corresponding to the shaft 62 of Figure 1, whose angular position determines the pitch of the variable pitch propeller. Attached to the shaft 62 is a segmental gear 100, which engages a rack 102 in the side of a piston 104. The piston 104 is subject to fluid pressures applied to its opposite ends, which determine its position. These pressures are controlled by a valve mechanism generally indicated at 106. The valve mechanism 106 includes a piston valve 108 subject on its bottom surface to the torque measuring fluid pressure applied through conduit 86 to a chamber 110. This pressure acts upwardly on the valve 108. The pressure is opposed by a downwardly acting force applied through a rod 112 and a lever mechanism 114 by a spring 116. The force of spring 116 is determined by the angular position of a cam 118. Cam 118 is keyed to the manually operated main control shaft 95.

The torque control is operated primarily in response to the balance or unbalance between the torque measuring fluid pressure in chamber 110 and the force of spring 116. The torque control may also be modified by a barometric control schematically indicated at 120, which varies the position of a bearing 122 in the lever mechanism 114 in accordance with either the compressor inlet absolute pressure (when in the form shown in Figure 3), or the differential between the compressor inlet and discharge pressures (Figure 4), or the sum of the compressor inlet and discharge absolute pressures (Figure

5); all of which vary the rate of flow of combustion air through the compressor. Provision is also made for modifying the torque control when water or other combustion modifying liquid is being supplied to the combustion chambers. The water pressure responsive mechanism includes a control valve 124 which operates a fluid motor 126 to shift the cam 118 so that it presents a different surface to its follower 128 when water is being supplied to the engine.

Limit controls are also provided which vary the torque control in response to engine speed and engine temperature. The speed limit control includes a governor 130 and the temperature limit control includes a control valve mechanism generally indicated at 132.

The valve mechanism 106 which controls the piston 104 includes the valve 108, which has a pair of lands 134 and 136 separated by a recess 138. The valve mechanism is enclosed within a casing 140. The valve 108 moves within a pair of generally cylindrical seat members 142 and 144. The seat members 142 and 144 are provided with central bores which extend axially through them from end to end to receive the valve 108. Each of these seat members 142 and 144 also has a larger bore concentric with its central bore and extending inwardly from one of its ends. The seat members 142 and 144 are arranged within the casing 140 so that these larger bores face each other. A sleeve 146 is received within the two larger bores.

The sleeve 146 is made slightly longer than the combined depth of the bores in the seat members 142 and 144, so that when the sleeve and the two seat members are assembled, the facing end of the two seat members are held spaced apart by the sleeve. Sleeve 146 has radial grooves 148 cut in its end faces and annular recesses 150 at the periphery of each of its end faces. The seat members 142 and 144 are provided with ports 152 and 154 which connect the annular recesses 150 with external grooves 156 and 158, respectively, in the seat members 142 and 144. Groove 156 is connected to a conduit 160 which leads to the upper end of the cylinder 162 in which the piston 104 moves, and groove 158 is connected through a conduit 164 to the lower end of cylinder 162.

An inlet for hydraulic fluid under pressure is provided in the valve mechanism 106 through a conduit 166 which opens into the space between the seat members 142 and 144. The fluid then passes through one or more radial inlet ports 168 in the sleeve 146 to the recess 138 between the lands 134 and 136 on valve 108. The lands 134 and 136 are provided with flat surfaces 170 and 172, which are of triangular or curved contour so as to provide ports of tapering depth. These triangular or curved flats have their apexes pointed towards the recess 138, while their widest portions open into recesses 174 and 176 on the valve 108. The recess 174 is connected, at all operating positions of the valve 108, through a passage 178 to a drained chamber (not shown). Similarly, the recess 176 is connected at all operating positions of the valve 108 through a drain port 180 and a conduit 182 to a suitably drained chamber.

When the valve 108 is in the position shown in the drawing, all the ports leading to the opposite ends of cylinder 162 are closed by the valve. If the valve moves upwardly from the position shown, then the hydraulic pressure fluid may flow through inlet port 168, recess 138, radial passage 148, passage 150, drain port 152,

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groove 156, and conduit 160 to the upper end of cylinder 162. At the same time, the lower end of cylinder 162 is connected through conduit 164, groove 158, ports 154, and passages 150' and 148' to the valve 108 and thence through the port formed by flat 172 to the drain recess 176 and out through conduits 180 and 182 to the drain.

The upper end of valve 108 is keyed to a hub 184 which carries a gear 186 driven by the engine through suitable driving mechanism, not shown. The valve 108 is thereby continuously rotated when the engine is running. Hence, when the valve 108 is moved upwardly from the position shown, the upper end of cylinder 162 is continuously connected to the source of fluid pressure, but the lower end of cylinder 162 is connected to the drain only intermittently, as the tapered port formed by flat 172 passes the ports 148 in the sleeve 146. As the distance of valve 108 from its neutral position increases, the width of the port 172 which passes the port 148 on rotation of the valve, increases. Thus the proportion of the total time during which the pressure is being drained from the lower end of chamber 162 also increases. If the valve 108 is moved upwardly sufficiently far so that land 136 clears the port 148 completely, then the lower end of chamber 162 is continuously connected to the drain, and the piston 104 moves downward rapidly. Because of the intermittent action of the valve, the motion of the piston 104 takes place at a rate which depends upon the distance of valve 108 from the intermediate neutral position shown in the drawing.

The force due to the pressure acting upwardly on the bottom of valve 108 is a measure of the actual torque load on the engine. The force acting downwardly on the top of valve 108 is a measure of the desired torque load on the engine. When the actual torque is not equal to the desired torque, the valve 108 moves to cause operation of piston 104 to restore the actual torque to its desired value. The use of the flats 170 and 172 and the rotation of the valve ensures that the rate of movement of the torque controlling piston 104 is varied in accordance with the amount of departure of the actual torque load from its desired value.

Provision is made for a by-pass connection between the ends of cylinder 162, to permit manual operation of the torque control under emergency conditions. This by-pass includes a conduit 188 connecting the conduits 160 and 164. This conduit is controlled by a valve 190 operated by a solenoid 192. The electrical solenoid 192 may be energized by an electrically connected battery 193, upon closing a switch 195. A spring 194 biases the valve 190 to a position wherein the by-pass is closed. Upon manually closing switch 195, solenoid 192 is energized and overcomes spring 194, whereupon the valve 190 is opened, thereby permitting manual movement of the pitch control by lever 197, mounted to rotate freely on shaft 90 and connected to shaft 62.

The rod 112 is provided with a rounded lower end, and rides in a similarly rounded crater in the top of hub 184, so as to provide for angular misalignment between the rod 112 and hub 184.

The lever mechanism 114 includes a lever 196 and a lever 198 having opposed flat surfaces which engage opposite portions of a bearing 122. The bearing 122 is carried at the end of a rod 200, which is moved laterally by a barometric control shown diagrammatically at 120 in Figure 2 and in greater detail in Figures 3, 4 and 5. The

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rod 112 is pivoted to the lever 196. The lever 198 is pivoted near its center at 202, and the spring 116 acts upwardly on its right end. The lower end of spring 116 engages a retainer 204 which is positioned by cam follower 128.

Cam 118 is fixed on a hub 206, which is in turn keyed to the main control shaft 94 for rotation therewith. The connection of hub 206 to shaft 94 permits longitudinal movement of hub 206 along the shaft. The cam 118 has two cam surfaces 208 and 210, of different contours. These two cam surfaces lie side by side on the cam 118, so that the particular cam surface engaged by follower 128 is determined by the lateral position of the cam. The left end of hub 206, as it appears in the drawing, is extended to form a piston 212 within a cylinder 214. The end of piston 212 is provided with a flange, and a compression spring 216 retained between that flange and the cylinder 214 biases the piston to the left within the cylinder. Hydraulic fluid under pressure may be supplied to the cylinder 214 through conduit 218, and is controlled by control valve mechanism 124.

The valve mechanism 124 includes a piston valve 220 operated by a bellows 224 located in a chamber 226. The pressure of the water or other combustion regulating fluid being supplied to the combustion chambers of the engine is communicated to chamber 226 through conduit 52. A spring 228 biases valve 220 to the position shown in the drawing, in which conduit 218 is in direct communication with a drain conduit 230. When water under pressure is supplied to chamber 226, bellows 224 and spring 228 are compressed, thereby moving valve 220 upwardly so as to connect conduit 218 to a conduit 232 which is supplied with hydraulic fluid under pressure through conduits 234 and 236. When fluid under pressure is admitted through conduit 218 into cylinder 214, the piston 212 is moved to the right, thereby moving the cam surface 208 under the follower 128, so that the force of spring 116 is determined by the contour of cam surface 208, rather than by the contour of cam surface 210. When water is not being supplied to the engine, the spring 228 moves valve 220 to the position shown in the drawing, and the conduit 218 is connected to drain conduit 230. Spring 216 then moves the cam 118 to the position shown, where follower 128 rides on the cam surface 210. The engine is capable of developing greater power and hence carrying a greater load when the combustion modifying fluid is being supplied to the engine. Hence the contour of cam surface 208 is such that the force of spring 116 is increased when follower 128 is riding on that cam surface.

Provision is made for reducing the torque load on the engine whenever the engine speed falls below the value for which the manual control lever 88 of Figure 1 is set. The manual control lever 88 rotates the shaft 95 which carries a cam 238 operating a follower 240 which controls the speed setting of the governor schematically indicated at 130. The governor is rotated by the engine driven shaft 95. When the engine speed falls below the value determined by the position of follower 240, the governor 130 operates to lower a rod 242, which is connected through a strain relief spring 244 to another rod 246. The rod 246 extends upwardly through an opening in the lever 198 and is provided at its upper end with a flange 248. When the rod 246 is moved downward because the engine speed falls below the desired value, the flange 248 engages the upper surface of lever 198, rotating it clockwise,

and thereby reducing the spring force acting downwardly on valve 108. This operates the torque control system to partially unload the engine, and thereby allows the speed to recover its desired value more quickly.

The control shaft 95 also carries a temperature reset cam 250 and a fuel cam 252. The fuel cam 252 positions a follower 254 which determines the rate of flow of fuel to the engine by the mechanism described in connection with Figure 7. The follower 254 also carries an arm 256 which extends between a pair of push rods 258 and 260.

The push rods 258 and 260 are positioned by the fluid motor device 132 in the temperature override control mechanism. This control mechanism operates to reduce the torque load on the engine and to reduce the fuel flow whenever the tail pipe temperature exceeds a predetermined value.

The fluid motor device 132 includes a piston 262 moving in a cylinder 270. The lower end of piston 262 is provided with a gear 272, driven by the engine, so that the piston is continuously rotated. A spring 274 biases the piston 262 upwardly against the pressure in cylinder 270. A lateral surface of piston 262 is provided with one or more slots 276 extending downwardly from the upper end of the piston. As the piston rotates, these slots 276 provide intermittent communication between fluid supply conduits 278 and 280 and the cylinder 270. The conduit 280 is normally closed by a valve 282 controlled by the temperature reset cam 250. An outlet conduit 283 is provided for the chamber 270, which outlet conduit is controlled by a thermostatic valve, diagrammatically indicated at 284, subject to the temperature in the tail pipe of engine. The construction of the valve is such that it is normally closed when the tail pipe temperature is in the allowable range of values. The valve 284 opens whenever the tail pipe temperature exceeds the desired value, thereby reducing the pressure above the piston 262 in the cylinder 270. The spring 274 then moves the piston upwardly, carrying with it the push rods 258 and 260. Push rods 260 acts through a strain relief connection 286 and a push rod 288 on the right end of a lever 290. A rod 292 extends upwardly from the left end of lever 198 and through an aperture in the lever 290. The rod 292 is flanged at its upper end as indicated at 294. When the temperature control moves the rod 288 upwardly, the lever 290 is carried upwardly, so that it engages the flange 294 and moves rod 292 and lever 198 upwardly, in a torque decreasing direction. At the same time, the upward movement of push rod 258 is transmitted through arm 256 to the fuel control mechanism to reduce the fuel supply. Both of these changes tend to reduce the engine temperature.

In emergency conditions, it is desired to increase the upper limit of power output by increasing the maximum permissible temperature. This is done by the temperature reset cam 250. When the manual control lever 88 is moved to its maximum power position, the cam 250 opens valve 282, thereby providing an additional supply of hydraulic fluid to the cylinder 270, and increasing the pressure therein. Under such conditions the valve 284 which is subject to the tail pipe temperature has to open wider in order to reduce the pressure in cylinder 270 to a point where the push rod 258 and the related temperature control mechanism are operated to reduce the torque load and the fuel supply.

Figure 3

Figure 3 illustrates in detail one form which the barometric control 120 of Figure 2 may take. As illustrated in Figure 3, this barometric control includes a pair of opposed bellows 300 and 302. The interior of bellows 300 is supplied with fluid at the compressor inlet pressure by means of a conduit 304. The two bellows are connected by a rod 306, on which is pivoted a lever 308. The left end of lever 308 is connected by means of a pin and slot connection to a valve 310 which controls the supply of hydraulic fluid to a fluid motor generally indicated at 312. The motor 312 includes a piston 314 moving in a cylinder 316. The piston 314 moves a rod 318 which is connected through another pin and slot connection to the right end of lever 308.

The exteriors of the bellows 300 and 302 are exposed to the same pressures. That pressure acts upward on bellows 300 and downward on bellows 302, so that its influence on the expansion and contraction of the bellows is balanced out. The two bellows therefore move the rod 306 up and down in accordance with the difference between the compressor inlet pressure inside bellows 300 and the substantially zero pressure inside the evacuated bellows 302. The compressor inlet pressure used is the static pressure, which is a measure of the barometric pressure, whose chief variation is in accordance with altitude, so far as aircraft are concerned.

It may therefore be seen that the two bellows 300 and 302 move the rod 306 up and down in accordance with variations in atmospheric pressure. This motion is transmitted through lever 308 to valve 310, which controls the piston 314 to cause a following movement of piston rod 318.

For example, if the bellows 300 and 302 move the rod 306 downwardly from the position shown in the drawing, then the lever 308 pivots about its right end, moving valve 310 downwardly and admitting hydraulic fluid under pressure from conduit 320, past valve 310 and through conduit 322 to the cylinder 316 above piston 314. At the same time, the space in cylinder 316 below piston 314 is connected through a conduit 324 and past valve 310 to a drain conduit 326. The piston 314 is accordingly moved downwardly by the different pressures acting on it. This downward movement of piston 314 rotates lever 308 clockwise about its pivot on the rod 306. The valve 310 is thereby raised to return to its neutral position. When the valve again reaches its neutral position, the motion of piston 314 stops. It will be readily understood that, in a similar manner, upward movement of rod 306 produces a following upward movement of piston 314.

The piston rod 318 carries an arm 328. Pivoted on the end of arm 328 is a link 330, which extends upwardly from the arm 328. At its upper end, the link 330 carries a roller 332 which rides on a cam 334. The roller 332 is maintained in engagement with the surface of cam 334 by means of a bracket 336 pivoted on the link 330 and carrying a roller 338 which engages the opposite surface of cam 334. A spring 340 biases the bracket 336 and the link 330 apart, and thereby maintains both rollers 332 and 338 in contact with the opposite surfaces of the cam. Pivotaly attached to the upper end of link 330 is another link 290 which carries at its end a bearing 122 that moves between opposed facing surfaces of levers 196 and 198. The link 200, bearing 122, and levers 196 and 198 are the same parts designed

nated by those reference numerals in Figure 2. It will be seen that the lateral position of bearing 122 between the levers 196 and 198 is determined by the pressure acting on the bellows 300 and 302 and by the contour of cam 334.

Attached to the upper end of rod 313 is another link 342, which carries at its upper end a roller 344 riding on an elongated cam 346. Roller 344 is maintained in engagement with cam 346 by means of a bracket 348, a roller 350, and a spring 352, which function in the same manner as the bracket 336, roller 338, and spring 340, just described. Pivotaly attached to the upper end of link 342 is another link 352 which carries at its end a bearing 354 engaging surfaces on opposed levers 356 and 358.

The lateral position of bearing 354 is determined by the pressures acting on bellows 300 and 302 and by the contour of cam 346. The bearing 354 and its related elements are parts of the fuel control system illustrated in Figure 7.

Figures 4 and 5

These two figures show alternative arrangements of bellows for operating the barometric control system of Figure 3. These two bellows systems differ from those of the prior art in that they add to the compressor inlet pressure the compressor discharge pressure so that the composite of these pressures, which is the same as the compressor pressure differential, becomes the operating force of the barometric control. The advantage of this arrangement, over using compressor inlet pressure alone, is that in the latter the overspeed governor has poor acceleration control and no temperature signal, whereas the former gets good acceleration control and full available horsepower at all airspeeds and air temperatures.

Figure 4 illustrates a casing 360, whose interior is subject to compressor inlet pressure by means of a conduit 362. Inside the housing 360 are mounted a pair of opposed bellows 364 and 366. Bellows 364 is evacuated, while the interior of bellows 366 is subjected to the compressor discharge pressure by means of a conduit 368. The two bellows 364 and 366 are connected by a rod 370, which carries a lever 308 corresponding in structure and function to the lever 308 of Figure 3. The bellows 364 has a larger area than the bellows 366. For the purpose of analyzing the operation of these two bellows, the bellows 364 may be considered as having one area equal to that of bellows 366 and a second area acting in the same direction as the first area, but unopposed by any area of bellows 366. Considering the first area, it will be seen that the compressor impact pressure acts in an opposite direction on it to that in which it acts on bellows 366, so that its influence is balanced out. The bellows 366 and the first area of bellows 364, acting together, provide a downward force on the rod 370 which varies with the absolute compressor discharge pressure. Considering the second area of bellows 364, it will be seen that it is subject to the compressor impact pressure acting downwardly. Therefore the rod 370 is moved downwardly by the sum of two forces, one of which varies with the absolute compressor discharge pressure and one which varies with the absolute compressor inlet pressure. The compressor discharge pressure is a measure of the rate of flow of combustion air to the engine. The compressor inlet pressure, on the other hand, is a measure of the air density at the altitude of flight.

In the apparatus of Figure 4, both these factors are considered in determining the torque output of the engine and the fuel flow to the engine.

Figure 5 illustrates an arrangement of bellows equivalent in effect to the arrangement shown in Figure 4. Figure 5 shows a casing 370 containing a lower bellows 372, which is evacuated, and a pair of concentric upper bellows 374 and 376. The inner bellows 376 is supplied with fluid at the compressor discharge pressure through a conduit 378. The outer bellows 374 is supplied with fluid at the compressor inlet pressure through a conduit 380. The pressure within the casing 370 outside the bellows may be any desired pressure. The three bellows are connected by a rod 382.

It is believed to be obvious without any elaborate analysis that the bellows system of Figure 5 produces a net force acting downward on rod 382, which is the sum of one force varying with the compressor inlet pressure and a second force varying with the compressor discharge pressure.

Figure 6

This figure illustrates a valve structure which may be used in place of the valve mechanism 106 of Figure 2. Like the valve mechanism 106, the valve arrangement shown in Figure 6 will produce a rate of movement of the motor controlled by it, which varies in accordance with the amount of unbalance between the controlling forces acting on opposite ends of the valve. In the structure of Figure 6, this result is accomplished by the use of a heavy cantilever spring which biases the valve toward its neutral position in such a manner that under normal operating conditions, the valve will not depart from its neutral position by more than a few thousandths of an inch. In this way, a much simpler valve structure may be used than in Figure 2, with a very similar result.

Referring to Figure 6, there is shown a piston valve 400 having a pair of spaced lands 402 and 404. These lands cover ports leading to the opposite ends of a fluid motor when the valve is in its neutral position, as shown in the drawing.

The central recess between the lands is connected to a source of fluid pressure (not shown) and the recesses on the other sides of the two lands are connected to suitable drains (not shown). The connections between the valve, the pressure source, the drains, and the fluid motor, are substantially the same as those of valve mechanism 106 of Figure 2. The upper end of the valve carries a gear 406 by which it is continuously rotated to prevent sticking. The lower end of the valve is exposed to the torque measuring fluid pressure in a chamber 410, which corresponds to the chamber 110 of Figure 2. The downward controlling force acting on the valve is transmitted through a bearing pin 412 in the end of a lever 414. The pin 412 corresponds to the rod 112 of Figure 2. Between the lower end of pin 412 and an abutment 416 on the upper end of the valve 400 lies the end of a leaf spring 418. The right end of the leaf spring 418 is attached to a fixed support 420. A central portion of the leaf spring 418 is attached to an adjustable support generally including a pin 422 which extends through a bushing 424 in the casting 426. The upper end of pin 422 is flanged to engage the upper surface of leaf spring 418. The pin 422 is threaded immediately below the leaf spring 418 to receive a pair of lock nuts 428 to clamp the spring 418 between those nuts and the

flange on the upper end of pin 422. The sleeve 424 is internally threaded to engage the threads on the pin 422. The lower end of pin 422 extends beyond the sleeve 424 and is slotted so that it may be rotated by a screw driver or other suitable tool to adjust the vertical position of the central portion of leaf spring 418, thereby adjusting the neutral position of valve 400. A lock nut 430 and a cap 432 are provided on the lower end of pin 422 to hold it in any desired adjusted position.

Figure 7

Figure 7 shows the fuel flow control system which together with the apparatus shown in Figures 2 and 3, completes the control apparatus schematically indicated at 54 in Figure 1. This fuel control apparatus regulates the hydraulic pressure transmitted through conduit 56 of Figure 1 to determine the delivery of the pressure responsive variable delivery fuel pump 26. This variable fuel controlling pressure is established in accordance with (1) the engine speed, (2) the operation of the barometric control, and (3) the position of the manually operated control lever 88.

There is shown in Figure 7 a valve 440 which regulates the pressure acting on the under side of a piston 442, which pressure is transmitted to conduit 56. The valve 440 moves in a bore in a casting 444. When in the position shown in the drawing, the valve 440 traps the fluid in the chamber under piston 442. When the valve 440 is moved downward from that position, it admits fluid from a conduit 446 through a conduit 448 to the chamber 450. When the valve 440 is moved upward from the position shown in the drawing, it permits the fluid in chamber 450 to drain through the recess in the valve out through the upper end of the valve bore.

A rod 454 having its lower end rounded rests in a spherical seat in the upper surface of piston 442. The upper end of valve 440 carries a lever 452, whose left end is connected by a pin and slot connection to the rod 454. The upper end of rod 454 is connected to a lever 358 of a variable ratio lever mechanism 456, parts of which were shown and described in connection with Figure 3. The lever mechanism 456 varies the force applied by a spring 458 to the rod 454 in accordance with the operation of the barometric control.

The right end of lever 452 has a screw 460 adjustably attached thereto. The lower end of screw 460 engages or may engage an arm 256 carried by a follower 254 which cooperates with the fuel cam 252. The extreme right end of lever 452 rides on an arm 462 operated by governor 130.

When the engine speed falls below the desired value, the governor lowers arm 462, thereby turning lever 452 clockwise about its left end, and lowering valve 440. This admits fluid under pressure through conduit 448 into chamber 450, thereby increasing the fuel flow to the engine and moving piston 442 upwardly to restore valve 440 to its neutral position.

The cam 252 and its related elements affects the fuel flow only during conditions of acceleration when the engine speed is considerably below that indicated as desired by the governor. Under such conditions, the arm 256 stops the downward movement of screw 460, and thereby limits the rate of acceleration of the engine.

The rate of flow of fuel to the engine is modified by the barometric control acting through bearing 354 in the same manner as that the

torque is modified by the barometric control acting through bearing 122.

The coordination of the torque, speed and fuel systems are discussed in great detail in my co-pending application No. 762,038, filed July 19, 1947, previously referred to. There are also shown in that application the structural details of many of the parts of my control system which are shown in the present application only diagrammatically.

The terms and expressions used herein are employed for purposes of description and not of limitation, and I have no intention, in the use of such terms and expressions, of excluding any equivalents of the features shown and described or portions thereof, but recognize that various modifications are possible within the scope of the invention claimed.

I claim as my invention:

1. Control apparatus for an internal combustion engine and a variable torque load driven thereby, comprising means for varying the torque of said load, manually operable means for varying the action of said torque varying means, means for supplying a combustion modifying liquid to said engine, and means operable coactionally with said liquid supply means for modifying the relationship between said manually operable means and said torque varying means.

2. Control apparatus for an internal combustion engine and a variable torque load driven thereby, comprising means for regulating the torque of said load, control means for varying the torque value to be maintained by said regulating means, means for supplying a combustion modifying fluid to said engine, and means effective upon operation of said fluid supplying means to modify the relationship between said control means and said torque regulating means.

3. Control apparatus as in claim 2, in which said torque regulating means includes a control element, means responsive to the position of said element for varying said torque, and means responsive to the torque load for positioning said element, said control means includes a manually movable lever, a cam moved by said lever, and follower means connecting said cam and said control element so that movement of said lever varies a force acting on said element in opposition to said torque responsive means, said cam having a pair of parallel cam surfaces and being movable laterally to present selectively said surfaces to said follower means, and said modifying means includes means for moving said cam laterally.

4. Control apparatus as in claim 2, in which said combustion modifying fluid is a liquid, said supplying means delivers said liquid under pressure, and said modifying means includes an expandible bellows subject to said liquid pressure.

5. Control apparatus as in claim 2, including means responsive to the speed of said engine for controlling the supply of fuel thereto to regulate said speed at a selected value, and means movable concurrently with said control means for varying said selected speed value.

6. Control apparatus for an internal combustion engine, comprising means for regulating the power output of said engine, means for varying the setting of said regulating means including a manually movable lever, a cam moved by said lever, and follower means connecting said cam and said setting varying means, said cam having a pair of parallel cam surfaces and being movable laterally to present selectively said surfaces to said follower means, means for supplying a com-

bustion modifying substance to said engine, and means effective upon operation of said substance supplying means to move said cam laterally.

7. Control apparatus for an internal combustion engine and a variable torque load driven thereby, comprising means for varying the torque of said load, motor means for driving said torque varying means, control means for said motor means including a control element having a neutral position and effective upon opposite movements from said position to cause operation of said motor means in opposite directions, means for applying to said element a force acting in a torque decreasing direction and varying in accordance with the output torque of said engine, means for applying to said element an opposing force acting in a torque increasing direction, said last-named means comprising a pair of levers extending generally parallel to one another and each having a surface facing a similar surface on the other, a bearing located between and contacting both of said surfaces, means responsive to the rate of combustion air flow to said engine for moving said bearing along said surfaces to vary the mechanical advantage of said levers, a connection between one of said levers and said control element, spring means acting on the other of said levers in a torque increasing direction, manually operable means for varying the force of said spring means, and means responsive to the speed of said engine for moving said other lever in a torque decreasing direction when said speed exceeds a predetermined value.

8. Control apparatus as in claim 7, including means responsive to a temperature in said engine for moving said other lever in a torque decreasing direction when said temperature exceeds predetermined value.

9. Control apparatus as in claim 2, including means responsive to the speed of said engine for controlling the supply of fuel thereto to regulate said speed at a selected value, means movable concurrently with said speed-responsive control means for varying said selected speed value, said movable means being effective to increase said selected value from minimum to maximum upon movement over substantially one-half of its total range of travel, means responsive to a temperature in said engine for reducing the fuel supply when said temperature exceeds a predetermined value, and means movable concurrently with said control means and effective at a fixed point in the travel of said control means after said speed value varying means has reached its maximum speed position to increase said predetermined temperature value.

10. Control apparatus for a power plant including an internal combustion engine having a fuel supply thereto, a variable torque load driven thereby, and a compressor for supplying combustion air to said engine, said control apparatus comprising means subject to the pressure at the compressor inlet and the pressure at the compressor outlet and responsive to the composite, net force of said pressures, and means including said pressure responsive means for varying the torque load.

11. Control apparatus for a power plant including an internal combustion engine having a fuel supply thereto, a variable torque load driven thereby, and a compressor for supplying combustion air thereto, said control apparatus comprising: means subject to the pressure at the compressor inlet and the pressure at the compressor outlet and responsive to the differential of said pressures; means including said pressures differential responsive means for varying the torque load; and means operated by said pressure differential responsive means for varying the supply of fuel to said engine.

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