# United States Patent

[11] 3,554,084

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[54] **REDUNDANT FORCE SUMMING SERVO UNIT**
11 Claims, 6 Drawing Figs.

[52] U.S. Cl. .......... 91/3, 91/387, 91/411, 91/433, 91/448
[51] Int. Cl. .......... F15b 13/02, F15b 13/16
[50] Field of Search .......... 91/3, 387(Cursory), 363A, 388, (Cursory); 91/411, 446, 448, (Cursory); 91/384, (Cursory)

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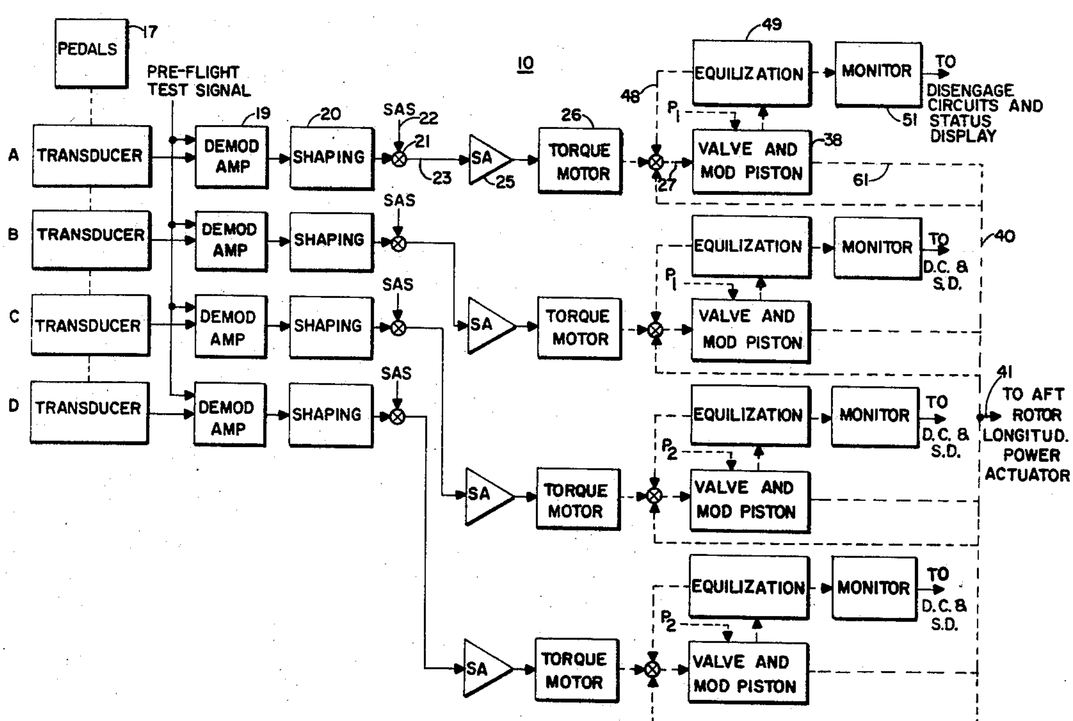
ABSTRACT: The servosystem which will tolerate failure in two of its four redundant channels and still be operational consists of four identical redundant units. The force outputs of the four redundant units are summed together, by means of a summing link or member, with the resultant force on the member being applied to a control valve of a fluid operated motor which is a low resistive load.

The input signal to each redundant unit is an electrical signal which energizes a torque motor. The torque motor responds by developing a force on its armature, which force is transmitted to a jet pipe valve. The jet pipe is deflected thereby causing a fluid pressure differential at the receiving holes coacting with the jet pipe nozzle. This pressure differential is transmitted to both a mod piston (which is the drive piston of the redundant unit) and the pressure sensor piston in parallel with the mod piston. The motion of both pistons is force fed back to the jet pipe by means of yieldable, impositive members as flat springs, thus closing two independent loops.



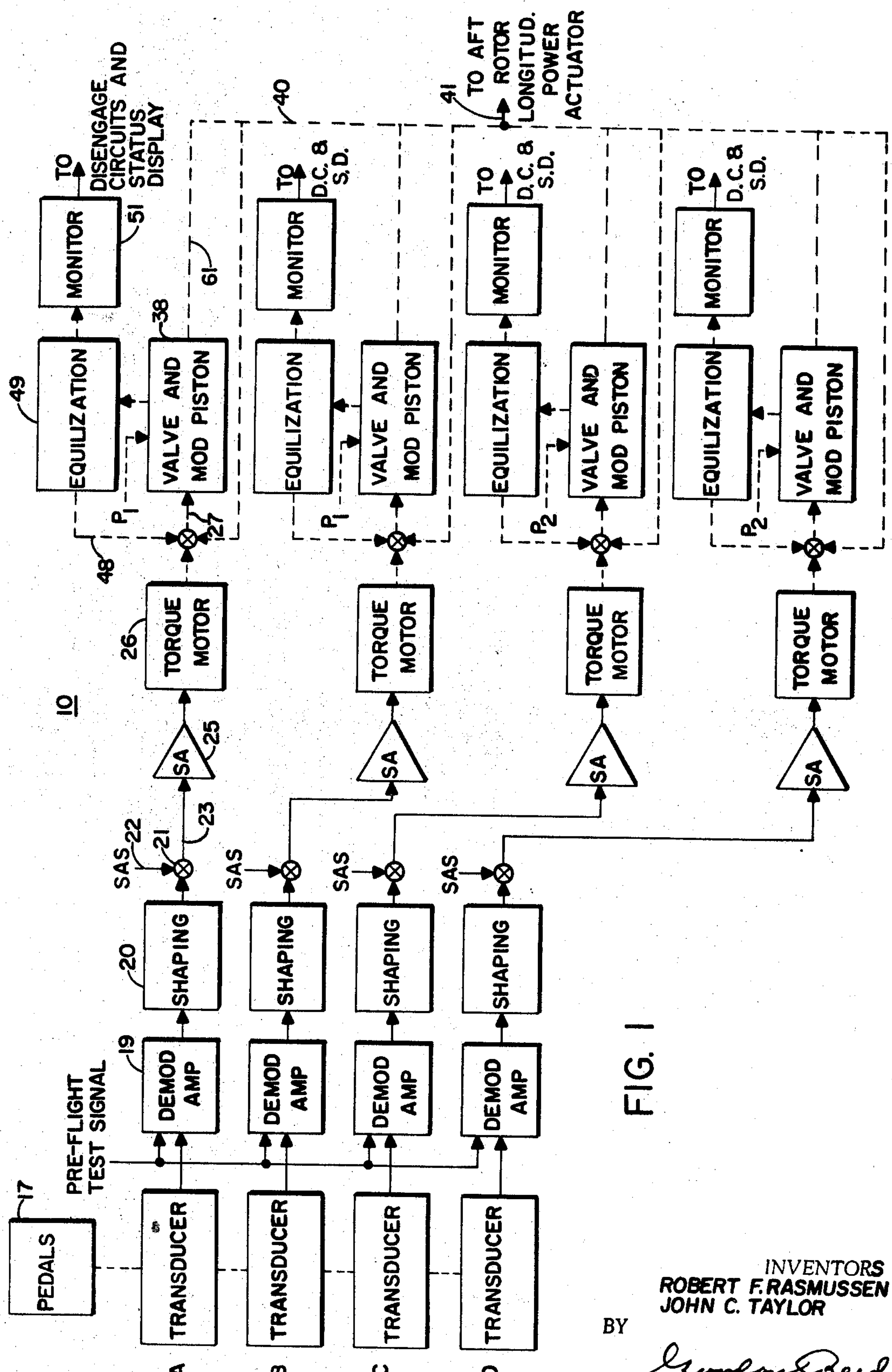



FIG. 1

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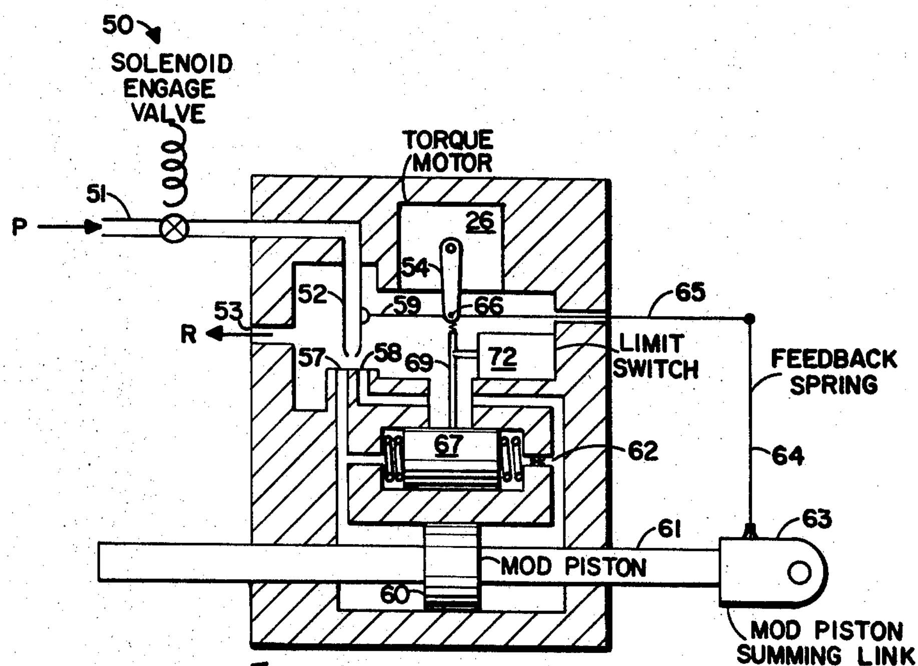


FIG. 2

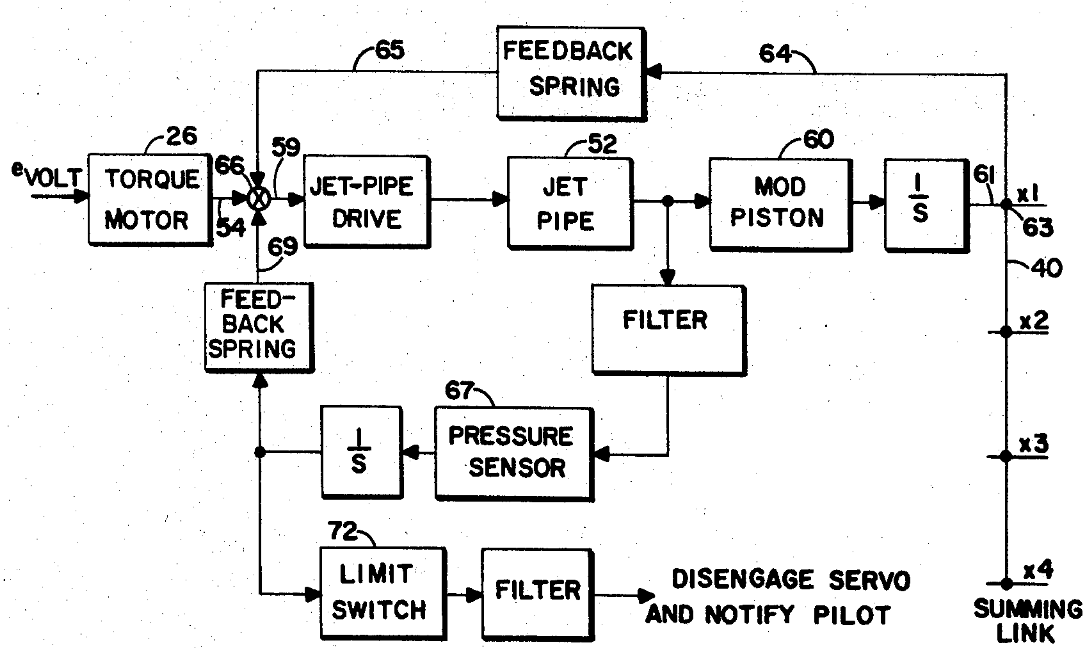


FIG. 3

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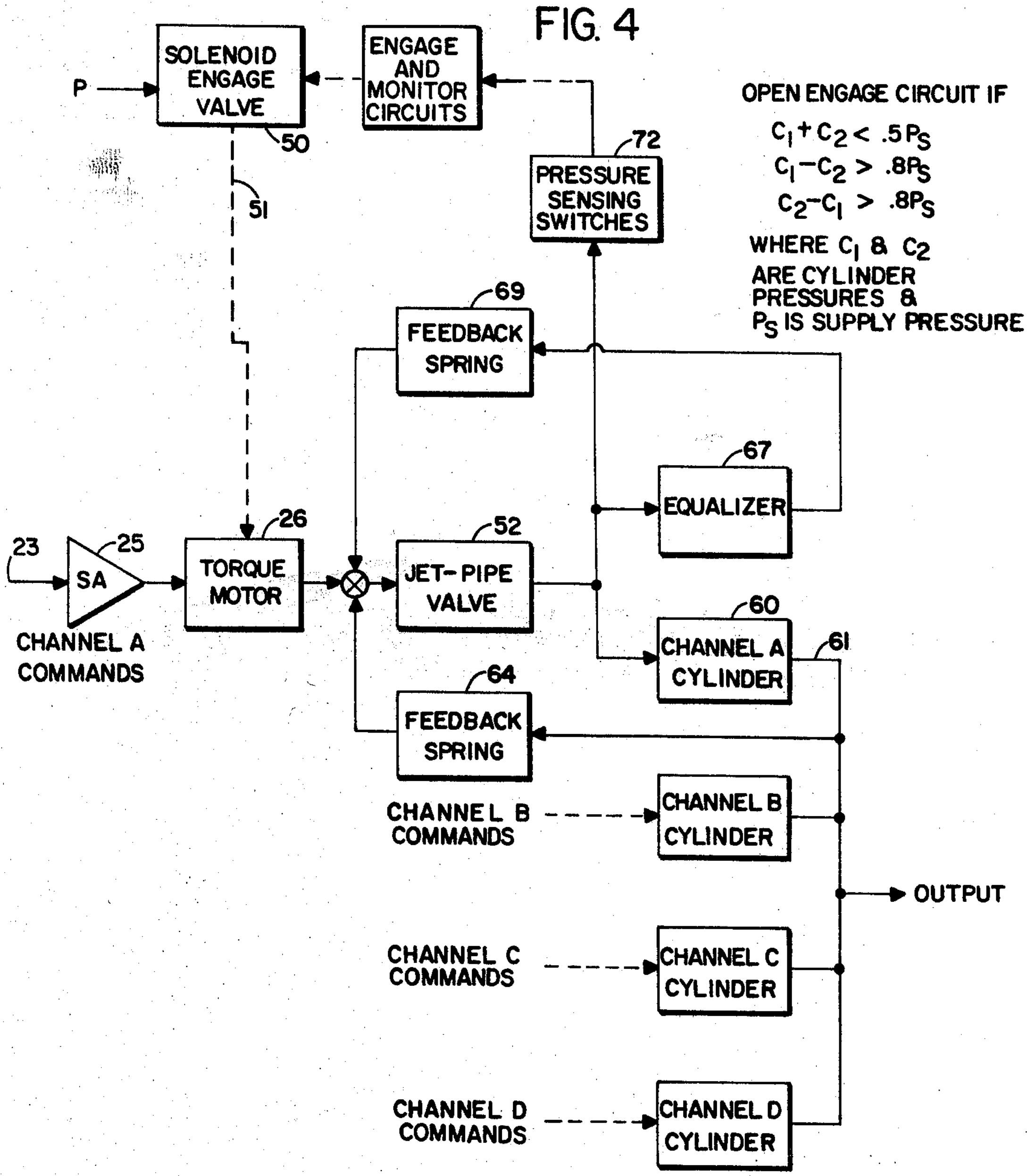
FIG. 4

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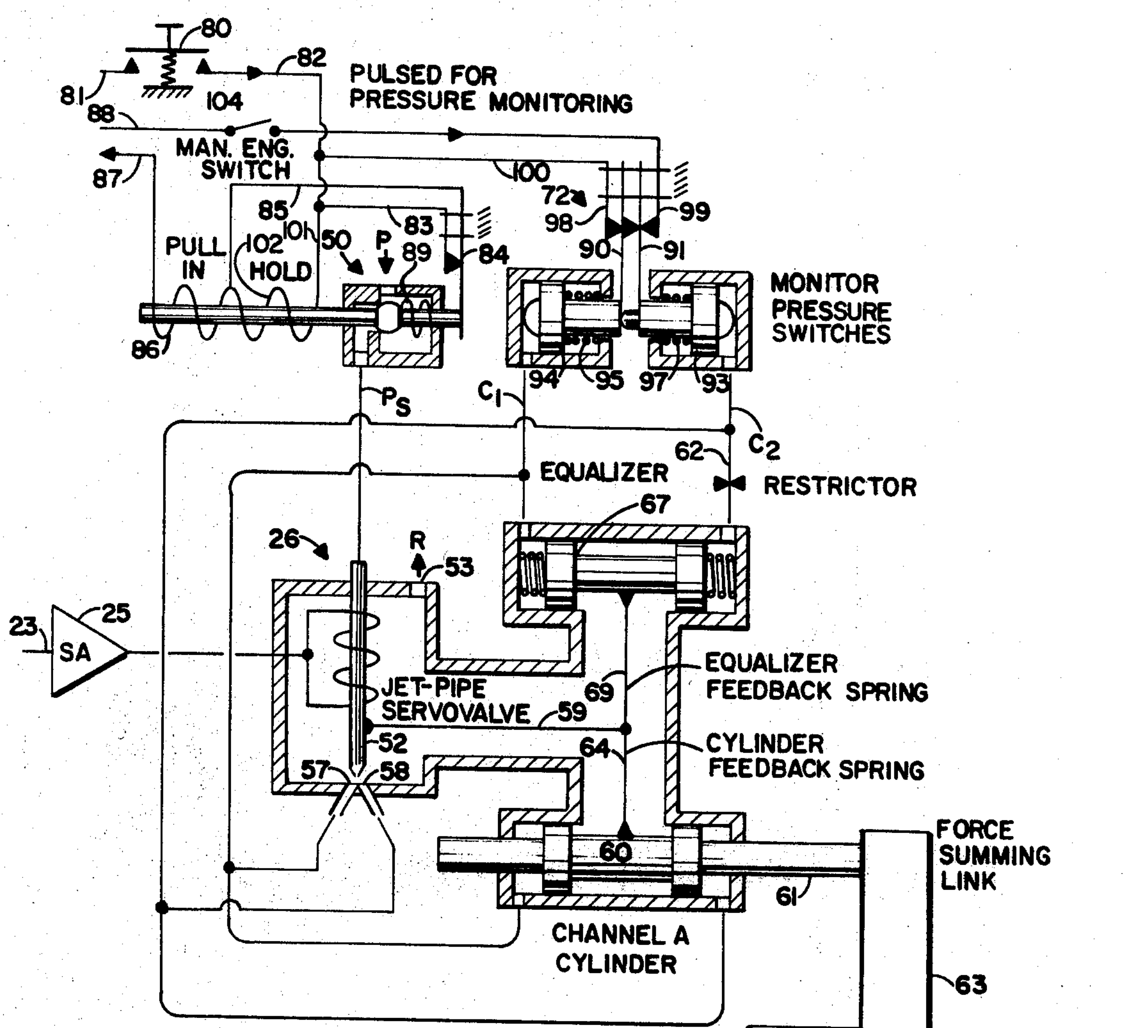


FIG. 5

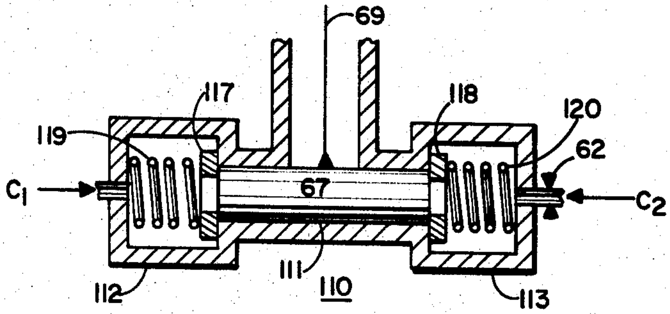


FIG. 6

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## REDUNDANT FORCE SUMMING SERVO UNIT

The purpose of the pressure sensor piston is to provide long-term hydraulic pressure equalization on all of the jet pipe receiving holes. For example, ideally if each of the four redundant units was identical and received the same input signal, the differential force on each mod piston because of the feedback, in the steady state, would be very low. However, this ideal condition can never be exactly attained because of differences during manufacturing resulting from tolerances and driving signal mismatches so it is possible for steady state extraneous forces to build up on the mod pistons and summing link. The forces are in balance and do no useful work. Some forces would be directed to the right, and others to the left, with a net force on the summing link being zero.

Since a pressure sensor piston receives the same pressure differential as its respective mod piston, it is possible to effectively measure the force on each mod piston by noting the displacement of its pressure sensor piston over a long period. The possibility obviously exists then of feeding back negative pressure sensor displacement to the jet pipe. By so doing, the differential pressure forces on the mod pistons could be relieved by repositioning the jet pipes.

Each redundant channel of the servo in one embodiment additionally includes a third piston arrangement which acts as a failure monitor. If a hardover failure (or a loss of input or feedback) occurred in any individual redundant channel, the pressure differential on both its mod and sensor pistons would build up to a large value. Both pistons would move in response to the pressure. However, the mod piston is constrained by the summing link, which is being held in place (within a small error) by the other redundant units; whereas the monitor pressure sensor piston is free to move, and does so, until it trips a failure switch which disengages the particular redundant channel or unit as by removing hydraulic pressure.

This invention relates to a novel "fly-by-wire control apparatus" incorporating duplicate or redundant units. The term "fly-by-wire"may be visualized as a structural departure with respect to a conventional aircraft having force transmitting control cables extending between the control stick operated by the pilot and the control surface or control surface actuator. In the "fly-by-wire apparatus," such cables are omitted and signals are taken directly from the pilot's stick or other signal sources for operating a motor to position the actuator.

## CONVENTIONAL CONTROL SYSTEMS

Existing conventional mechanical primary control systems, wherein the pilot through his control wheel and cables or linkage operates a control surface of an aircraft or controls the main actuator for the control surface, have reached a limit of performance due to the basic operation limitation of mechanical elements in such system. For example, such primary control system has mechanical elements wherein there is lost motion between the moving parts all whereby there is lack of concurrence in time or similarity between the movement of the control wheel and the operation of the surface. While the above lost motion effect has been referred to specifically there are other limitations on such a mechanical element system.

As the primary control system complexity multiplies to meet the increasingly stringent requirements imposed by high performance and variable geometry aircraft, sensitivity of the system to maintenance errors is magnified in such system, and the safety and reliability of the system decreases.

Additionally, as the mechanical linkage system is basically a single channel system with considerable cross-sectional area from the arrangement of its operable elements, a high vulnerability to combat damage exists.

## REDUNDANT FLIGHT CONTROL LINKAGE APPARATUS

The above disadvantages of conventional mechanical control linkages in an aircraft control surface positioning apparatus can be eliminated by use of electrical and electronic techniques. Thus, an electrohydraulic mechanization of the primary flight control system will provide an improved level of performance through elimination of the mechanical linkage nonlinearities. However, as an electronic or hydraulic element is more prone to failure than a comparable mechanical device multiple channels also known as redundant channels of the electrohydraulic system must preferrably be provided. Use of properly isolated redundant channels magnifies many times the probability that satisfactory system operation will remain after failures in one or more channels have occurred; therefore, a redundant automatic flight control system can provide higher reliability and safety as well as improved performance over the above type of primary control system.

One object of this invention therefore is to provide an improved apparatus with channel redundancy and with a minimum requirement for cross channel monitoring.

Another object of this invention is to provide for channel redundancy monitoring with a simple "in-line" monitor configuration eliminating requirements for cross channel comparisons and voting logic.

A further object of this invention is to provide redundant channels each having a separate force summed servo which servos coact on a common output member to minimize movement of the output member upon occurrence of a failure.

A further object of the invention is to provide in each of multiple redundant channels a hydraulic or fluid pressure equalization monitor arrangement to minimize channel mistracking.

A further object of the invention is to provide for each of the redundant channels an individual improved malfunction or failure monitor with provisions for disengaging a malfunctioning channel.

The above and other objects and advantages of the invention will appear upon consideration of the accompanying specification along with the subjoined drawings showing various embodiments of the invention.

In the drawings:

FIG. 1 is a block diagram of the novel four channel redundancy control apparatus;

FIG. 2 is a schematic of one modification of a pressure equalization disengage monitor arrangement;

FIG. 3 is a block diagram of a redundant channel along with the summing link or summing member for the four channels;

FIG. 4 is a block diagram of a second form of jet pipe equalization pressure arrangement and monitor failure control;

FIG. 5 is a schematic diagram of the arrangement of FIG. 4; and

FIG. 6 is a schematic of a modified form of equalizer piston-cylinder configuration.

In the apparatus, four identical channels, which receive like control signals, each include a force summed fluid servo. The servos may have separate fluid sources or there may be a lesser number of fluid sources than servos with suitable pressure operated switching means for each source for switching out a failed source from a servo and switching in an unfailed source to the servo, for example. The four servos are utilized to drive a common output member with each servo driven directly from operation of a member of one of the redundant channels. Multiple force summing minimizes the output link response to a failure of one channel thus allowing a simple failure monitoring arrangement to disengage a failed channel.

Also a simple pressure equalization arrangement for a fluid servo is used in each channel to minimize the mistracking due to mismatching of inputs or tolerances within the servo.

Response requirements of the servo failure monitor are not as critical with force summed servos (relative to displacement summed servos) as slowness or even failure of the monitor to disengage a hardover servo, for example results in only a small movement of the output link.

In each channel, a jet pipe valve utilizing an electrically driven moveable orifice is used to provide proportional control and also to minimize hydraulic contamination effects which sometimes occur in other arrangements.

A simple hydraulic pressure sensor piston monitors the differential pressure across each individual servo power piston and feeds back a time integrated and limited signal to the torque motor output which reduces the servo differential pressure and thereby channel mistracking. The primary positional feedback to the torque motor linkage that positions the jet pipe is obtained from the common multiple force summing member that is positioned by the four redundant servos.

Referring to FIG. 1, a redundant control system 10 includes a plurality of redundant channels A, B, C, and D. Since the channels are similar, a description of one will suffice for a description of all. Thus referring to channel A, an electrical variable signal source such as control transducer 17 operated by movement of a member such as the control stick of an aircraft supplies the control signal in each case to a demodulator amplifier 19 which in turn through a shaping network 20 supplies a control signal to summing device 21. The summing device also receives control signals from stabilization and control augmentation systems (not shown) through conductor 22. The signal from summing device 21 is transmitted by conductor 23 to a servoamplifier 25. The output of the amplifier 25 reversibly controls the operation of a conventional torque motor 26 which through motion transmission means 27 controls the operation of a moveable servo control member that differentially ports fluid to the ends of a piston of a servo 38. The piston under control of the fluid applied thereto operates through piston rod 61 a common member 40 of the four channels, and the common member 40 has an output 41 that operates for example the control valve of a main actuator for a control surface.

The movement of the servopiston and its rod 61 to force summing member 40 is also supplied in a feedback arrangement of the force transmission type to a force summing arrangement within servo 38. The output from the force summing arrangement within servo 38 is transmitted to the moveable servocontrol member.

Additionally, the force summing arrangement within servo 38 receives an input over transmission means 48, of the force type, from a pressure equalization arrangement 49.

The equalization arrangement 49 and a monitor 51 receives as inputs thereto the pressure across the mod piston in servo arrangement 38 to be described.

FIG. 2 shows a force summed fluid-type servomotor 38 for one channel. Servo 38 includes a torque motor 26 (that receives the variable transducer signal) having an output arm 54 which through a link 59 operates to variably displace a jet pipe or moveable servocontrol member 52. The jet pipe 52 receives pressure fluid through a solenoid operated engage valve 50 and conduit 51 from a fluid pressure source P. A return 53 is provided from the discharge of jet pipe 52. The jet pipe 52 coacts with two holes or ports 57, 58 which normally receive equal fluid pressure discharge from the jet pipe 52. The holes 57, 58 communicate by suitable conduits to opposed sides of a mod piston 60. Operation of the torque motor arm 54 and its connected link 59 in either direction according to the transducer signal displaces the jet pipe 52 to cause reversible movement of the mod piston 60. An output rod 61 of the mod piston 60 connects to a mod piston summing link or summing member 63 which may be used to position a control valve of a main actuator, now shown. The jet pipe 52 is repositioned toward its normal position by a feedback arrangement comprising summing member 63 on common member 40, a feedback spring 64 and link 65 connected to torque motor arm 54. The redundant servos operated member 63 thus exerts supervision of the feedback to jet pipe 52.

A hydraulic fluid cylinder-piston type monitor having a piston 67 has applied to opposite ends thereof pressure. The fluid is conducted to the ends through suitable orifices 62 or restrictors in subconduits connected with main conduits extending to opposite sides of the mod piston 60. The orifices provide a time delay to operation of the piston 67 following initial displacement of mod piston 60. Movement of the monitor piston 67 is transmitted through a feedback spring 69 (thus through a force or impositive feedback member) to the torque motor arm 54 to reposition jet pipe 52 to thereby provide pressure equalization from jet pipe 52 on the holes or ports 57, 58.

In addition, although a separate monitor may be provided, the hydraulic fluid clinder-piston monitor having piston 67 operates through a suitable connection a disengage limit switch 72 for opening the electrical circuit through a solenoid operated engage valve 50 permitting the valve to close by suitable means such as a spring to interrupt the flow of fluid to the adjustable jet pipe 52, thus indicating a failure in the particular channel involved.

FIG. 3 is an analysis block diagram of the force summed servo 38 of FIG. 2 wherein the electrical torque motor 26 receives a variable voltage signal over a conductor from amplifier 25 FIG. 1. The torque motor as a transducer converts the electrical input to a mechanical torque output proportional to its electrical input. Through its arm 54 and connecting link 59, it initially positions the jet pipe 52 a distance proportional to the difference between the torque motor output and the two feedback spring (64 and 69) forces.

Due to its displacement, the jet pipe 52 varies the fluid pressure in receiving holes 57 and 58, proportional to its displacement, whereby the mod piston 60 is positioned in inches per second in accordance with the flow resulting from that pressure differential which is applied to its opposite sides. The piston 60 moves in its cylinder in accordance with the integral of the time period of the differential pressure and resulting rate of flow, and through its rod 61 positions the mod piston summing link 63 and member 40 which operates the feedback spring 64 to provide a force feedback proportional to the displacement of the summing link 63 from the neutral position. The force on spring 64 varies the normal dimension or shortest distance between its output-input points. This force is transmitted through the connecting link 65 to the force summing point 66 which may be considered the remote end of arm 54.

While the servoloop has been described, there is also included a second loop for repositioning the jet pipe 52 through the pressure sensor monitor piston 67. The differential pressure on ends of the mod piston 60 resulting from any load on said mod piston because of disagreements with other mod pistons connected to summing link 40 is also applied through the suitable filters or orifices to the pressure sensor piston 67 which has a total displacement or movement dependent upon the time integral of the differential pressure supplied to the opposite sides of piston 67. This displacement is converted through the spring member 69 to pounds per inch of displacement of piston 67 and is also applied to the force summing point 66.

Additionally, in the embodiment of FIG. 2 the displacement of pressure sensor monitor piston 67 is supplied to the disengage on-off limit switch 72 which operates as will be described hereinafter to break the circuit through the operating solenoid of engage valve 50 thereby terminating the flow of fluid from the pressure source P to the jet pipe 52. The switch 72 also controls the energization of an indicator for notification to the pilot of such failure in the channel.

As indicated in FIG. 3, X2, X3, X4, the mod pistons of the three other redundant, force servos are connected to the summing member 40. It will be understood that each channel torque has similar feedback provisions as that in channel A.

While mechanical feedback has been described specifically to apply impositive torques to the jet pipe 52, it is contemplated that the feedbacks from summing member 40 and pressure sensor piston 67 could be electrical signals that would, instead of being applied to torque arm 54 as the mechanical feedback are, be applied to amplifier 25.

Concerning a second embodiment, and advancing from FIG. 3 the arrangement in FIG. 4 resembles FIG. 3 except for the fact that the differential pressure applied to the mod piston 60 is not only applied to the mod piston pressure equalizer 67 but is separately applied to the engage valve control circuit, switch 72 having a separate pressure responsive piston operating means. FIG. 4 thus has reference characters corresponding to those in FIG. 3.

In addition to the mod piston pressure sensing or responsive switches 72 controlling the solenoid engage valve 50, the valve may additionally be controlled from engage and monitor circuits to be described. FIG. 4 consequently includes logic statements which will hereinafter be reviewed.

FIG. 5 shows the force summed servo with the mechanical feedback of FIG. 4 shown in detail. The features common with FIG. 2 have the same reference characters.

FIG. 5, since it presents merely novel electrical circuits over material previously considered, will be described concurrently with its operation, thus momentary closing of a manually operable start monitor switch 80 upper left closes the circuit from a DC voltage source (not shown), conductor 81, switch 80, conductor 82, conductor 83, presently closed solenoid operated switch 84, conductor 85, through a pull-in electrical winding 86 of a push-type solenoid, conductor 87 to DC return. Previous to operation of switch 80, the engage valve 50 as shown is in closed position whereby pressure from the source P cannot be transmitted to the jet pipe 52. Following closing of switch 80 and energization of winding 86, the valve moves rightward in the FIG. thus permitting pressure fluid to be supplied to the jet pipe 52 and concurrently opening the circuit through switch 84 by an extension of the valve and valve 50 remains open by the following arrangement.

Pressure monitor disengage switch 72 as shown consists of four relatively moveable end mounted members two of which, 90, 91 are elongated to engage with shoulders on moveable plungers 93, 94, of the mod piston pressure monitor. Each of the separate plungers 93, 94 have engaged therewith suitable outwardly biasing springs 95, 97. It is evident that when pressure fluid through suitable conduits shown is admitted to one end of each of the plungers 93 and 94 that the normal pressure in ports 57, 58 as transmitted by the conduits compress the biasing springs 95, 97 causing the operable inner switch members 90, 91 to contact each other. The outer switch contacts 98, 99 which normally engage their respective inner contacts 90, 91 remain so engaged.

Thus, as involving switch 72 and its closed contacts, upon opening of engage valve 50 by operation of switch 80, a holding circuit for the engage valve solenoid is established from an electrical supply as a DC source (not shown), conductor 88 closed engage switch 104, outer contact 99, contacts 91, 90, 98, conductor 100, conductor 101, hold winding 102, pull-in winding 86, conductor 87 to DC return.

Concerning the operation of the pressure monitor disengage switch 72, in the event that there has been a failed operation malfunction in signal operated amplifier 25 of channel A for example whereas the remaining channels have operated, motion is applied in channel A through the force summing link 40 and rod 61 to piston 60 moving the cylinder feedback flat spring 64 and displacing the jet pipe 52, which has no torque applied thereto from torque 26, in accordance with the displacement of the output member 40 or summing link 63. This unrestricted displacement of pipe 52 results in a large pressure in one or the other of holes or ports 57, 58 which can approximate 80 percent of the total pressure $P_s$ resulting in a large displacement say of plunger 94 relative to plunger 93 or vice versa and as to plunger 94 engaging switch arm 90 thereby opening the electrical circuit between outer contact 98 and inner contact 90 or as to plunger 93 being operated disengaging outer contact 99 and inner contact 91 thereby opening the electrical circuit between conductors 88 and 100 thus deenergizing the windings 102, 86 of the solenoid causing the engage valve 50 to move to its closed position as shown in FIG. 5. The closing movement of the valve 50 may be obtained by a suitable auxiliary spring means 89 connected thereto aided by the pressure from the hydraulic power source.

As in FIG. 2, in FIG. 5 the spring biased, mod piston pressure equalizer piston 67 is supplied with fluid through a restrictor 62 to limit the flow rate thereto so that it operates on a long term basis. In one mode of operation of the pressure equalizer piston 67, in the event that the A channel mod piston 60 is jammed and will not move, a differential pressure on the opposite ends of the mod piston 60 for an extended time period occurs. The equalizer piston 67, because of the presence over the time period of the differential pressure, is displaced and through its spring connection 69 will return the jet pipe 52 to normal position where it applies equal pressure to the two openings 57, 58 and thus to both ends of the mod piston.

The circuit between conductor 88 and outer contact 99 of limit switch 72 as stated includes a manually operable engage-disengage switch 104 so that the channel may be also manually disengaged. It is moved to closed position following momentary operation of switch 80 to hold the solenoid valve 50 in the open or engaged position.

With respect to the logic statements of FIG. 4, note that in FIG. 5 the pressure line, to one end of monitor pressure switch plunger 93 remote from its biasing means, has been labeled $c_2$ and the line conveying pressure to one end of plunger 94 remote from its biasing spring is labeled $c_1$. Additionally the line conveying the pressure from the source to the jet pipe 50 is labeled $P_s$. In FIG. 4, the disengage logic is tabulated. In other words, the electrical engage circuit for valve 50 is opened between contacts 90, 91, if $c_1 + c_2$ be less than .5 $P_s$. Also the circuit is opened between contacts 90, 98 of the limit switch 72 if $c_1 - c_2$ be greater than .8 $P_s$. Finally, if $c_2 - c_1$ be greater than .8 $P_s$ the engage circuit is opened at contacts 91, 99. In other words, for the first condition, the inner contacts 90, 91, will become disengaged or are not closed. For the second condition, outer contact 98 and inner contact 90 will be disengaged. For the third condition or third logic statement, outer contact 99 and contact 91 will be disengaged. In other words, the solenoid winding for the engage valve 50 will be deenergized if there is very low pressure supplied to the mod piston 60 by the jet pipe 52. Also, if the pressure be excessive on one side of the mod piston relative to the other, either switch contacts 98, 90 or 99, 91 will be disengaged.

FIG. 6 shows a modified form of cylinder-piston equalizer 110, where the cylinder has a small diameter portion 111 housing the piston 67 and two enlarged cylindrical end sections 112, 113. The end sections of chambers 112, 113 are connected to the lines $c_1$, $c_2$ FIG. 5, with the chamber 113 including in the supply line thereto a fluid flow restrictor 62. End chamber 112 includes a washer 117 which abuts one end of chamber 112 while subject to the force of a prestressed coil spring 119. Similarly, the chamber 113 includes a washer 118 biased by a preloaded coil spring 120. The arrangement is such that the washers 117 and 118 with no pressure applied to conduits $c_1$ and $c_2$ apply no force to the piston 67. However, when a differential pressure is applied to the conduits $c_1$, $c_2$ the pressure must overcome either one of the preloaded springs 119, 120 before any movement is applied to the piston 67 for moving the jet pipe 52.

The arrangement in FIG. 6 has its advantages over that in FIG. 5 during the application of small static loads or small steady state loads on the summing member 40. For example, the apparatus including the summing member 40 may be associated with the primary controls such as the control stick and feel system in an aircraft whereby static loads from the feel system may be applied to the member 40 to apply a steady static load thereto. By the arrangement of FIG. 6, there will be no action or movement of the equalizer piston 67 until the static load exceeds the preloading of either one of the springs 119, 120, in other words by use of the springs 119, 120 and the arrangement of FIG. 6 a "dead spot" is provided so that operation of the piston 67 does not occur until the differential pressure exceeds the preloading of either spring 119 or 120 thus static loads are tolerated.

## OPERATION

Each of the servo amplifiers, such as amplifier 25 in channel A, for the force summed fluid servos receive a similar electrical control signal. The multiple mod pistons are normally linked together to a common output member **63**. In accordance with the common control signals, the mod pistons normally or ideally have the same operation. In actual practice, when such a mechanical arrangement is made, the null of each of the servounits will vary in accordance with manufacturing tolerances, material variations, etc. To compensate for these effects and to equalize the load-carrying capacity of the individual mod pistons, a differential pressure sensor equalizer is connected to the corresponding output cylinder to have applied thereto the differential pressure on a mod piston. To permit only long term pressure variations to effect operation of this mod piston pressure equalizer, flow to its opposite ends is limited by orifices. This limitation—plus the effect of the pressure sensor displacement characteristics —permits the input signal to the jet pipe to be modified such that the loads imposed on all the mod pistons are equalized, or nearly so.

Should a failure occur in but one of the redundant units —for example a hardover failure due to loss of servo feedback —the pressure sensor (failure switch) of that unit because of high pressure due to displacement of pipe **52** from center will move off towards one extreme. The opposing load on the remaining redundant mod pistons will be a fraction of this pressure value since they oppose the failed channel collectively. With the failure switch **72** set to 80 percent of the available pressure, $P_s$, from the pressure source, only the failed mod piston or servo unit achieves this pressure level and, thereby, trips its failure switch **72** indicating failure and disabling the fluid supply to its jet pipe. The remaining channels continue engaged or operative.

This type of operation requires that the mod pistons handle little or no load as stated. This is accomplished by having the mod pistons drive a standard control valve for a main actuator. While there are some dynamic loads reflected to the mod pistons, the effect of the pressure sensor (equalizer) line orifices minimize these effects.

Either soft or hardover failures are detected directly by the failure switch **72**. Dead failures are detected due to the relatively high-pressure gain utilized, which, when a command appears to the good servos makes the dead servo appear to be hardover in the opposite direction.

The four channels may have individual fluid sources and their jet pipes may be engaged and will remain engaged with at least one fluid source through pressure responsive switching in the event its original pressure source fails.

It will be understood that the gains of the redundant channels are so designed that when the overall system calls for the operation of a pressure equalization monitor, such as **67** FIG. **5**, there is little probability of operation of a disengage switch **72**, in response to such call.

## FAILURE MONITORING

The force summed servos with hydraulic mod piston pressure equalization provide unique, straight-forward monitoring of channel performance. When channel mistracking reaches a predetermined level, it is, by definition, a failure of that channel and a switch is actuated. Essentially, this method of monitoring compares each servo against the average output for all servos in that axis. Thus, no complicated logic arrangement is required to sort out the failed channel. In the four channel embodiment herein, even after two channels fail and are disengaged, the system is fail safe since if a malfunction occurs in one remaining channel, any tendency for irregular operation of the output member **40** is compensated by the remaining properly operating channel.

It will now be evident that there has been provided a novel control system such as a "fly-by-wire system" consisting of redundant channels, having fluid operated servos. Such systems, because of the redundancy, provide a fail operational arrangement more reliable than the conventional primary control system. Further, the arrangement includes a self-equalization arrangement for equalizing the fluid pressure on the holes cooperating with a jet pipe in each redundant channel. Additionally a disengage monitoring arrangement has been included responsive to fluid pressure on a servo to disable a failed servounit.

We claim:

In a fluid type actuator having a power output section and a control section, receiving fluid under pressure, for porting fluid through suitable passages to said power section, said control section have a displaceable fluid conducting member and coacting receiving ports to vary the flow rate to the power section and thus the velocity of displacement thereof in combination:

- means for variably displacing said member from a normal position relative to said two coacting fluid receiving ports;
- yieldable feedback means between the member and power output section operated from said power section to modify the position of or return said member and ports to the normal position; and
- further fluid pressure monitor means including a controller having subpassages, at least one having flow restrictive means, connected to said passages to said power section thereby responsive by said restrictive means to time duration of fluid pressures applied to said power section, operating on the member to control the rate of flow to the power section.

**2.** The apparatus of claim **1** wherein operation of said controller in said further means modifies the flow rate by modifying the position of said member to provide long term equalization of fluid pressure in the passages to the power section.

**3.** In a fluid-type actuator having a power output section and a control section for porting fluid from a source of pressure fluid to said power section, said control section having a displaceable member to vary the flow rate to the power section and thus its displacement rate:

- first means for variably displacing said member from a normal position;
- second or feedback means operated from said power section to return said member to normal position;
- a pressure fluid conduit means in said control section and having an operable valve therein for transmitting fluid to said control section; and
- pressure fluid monitor means connected to said power section thus responsive to fluid pressure therein and controlling the operation of said valve.

**4.** The apparatus of claim **3** characterized by the valve being of the solenoid actuated type and the pressure fluid monitor means also includes a switch means having contacts controlling a circuit of said solenoid and responsive to pressure to said power means to effect opening of said valve to supply fluid to the conduit.

**5.** The apparatus of claim **4**, in including a pressure monitoring start switch for momentarily energizing said solenoid to momentarily open the valve to passage of fluid to apply pressure to said power means and switch means.

**6.** The apparatus of claim **3**, pluralized to provide at least three redundant actuators, and wherein each power section thereof is connected to a common force summing member, all power sections normally applying effects to an output of said summing member, whereby each feedback means is operated in accordance with the average position of the output from the summing member whereby failure to cause a first means in one section to initially displace a displaceable member in the section results in the feedback means repositioning said displaceable member in the section causing a larger than normal operating pressure thus a change in normal operating pressure to its power means with the accompanying operation of the pressure monitor means to terminate operation of said valve.

**7.** The apparatus of claim **6** wherein the first means displacing said member and the second feedback means thereof are structurally of the impositive type.

**8.** The apparatus of claim **7** wherein the power section includes a power piston and wherein the displaceable member controls the position of a nozzle or jet pipe which normally applies equal pressures to a pair of receiving ports or openings directly connected to opposite ends of the power piston of the power means to apply normally equal pressures thereto.

9. The apparatus of claim 7, including a pressure equalizer monitor responsive to time duration of fluid pressure applied to said power section and operatively connected to said displaceable member.

10. The apparatus of claim 3, characterized by the valve being of the solenoid actuated type and the pressure fluid monitor means also includes a switch means controlling a circuit of said solenoid and responsive to continued large displacements of said member from normal position opening said circuit to return the valve to closed position.

11. In control apparatus, means for operating a low resistive load output comprising:

a plurality of redundant channels each having a piston type proportional operated fluid servomotor, all servomotors through their pistons operating a force summing common member connectable for operation to said output;

first means displaceable for applying differential fluid pressure to said redundant servo motor pistons to effect operation thereof;

second means responsive to excessive differential pressures in each servomotor operating a failure switch connected to the first means for terminating application of fluid pressure to said servomotor thus terminating operation of said servomotor; and

a differential pressure responsive device in each channel connected through fluid flow restrictive conducting means to opposite sides of its associated piston to respond only to time duration differential pressures across said piston and connected to the first means to modify the displacement thereof.