

United States Patent

[11] 3,561,322

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- [54] STABILITY AUGMENTATION SYSTEM**
14 Claims, 8 Drawing Figs.
- [52] U.S. Cl.**..... **91/1,**
91/363, 91/411, 91/437; 244/78
- [51] Int. Cl.**..... **F01b 25/26,**
F15b 11/16
- [50] Field of Search**..... **91/1, 411,**
411 (A), 413, 437, 438, 363 (A); 244/(Inquired),
78

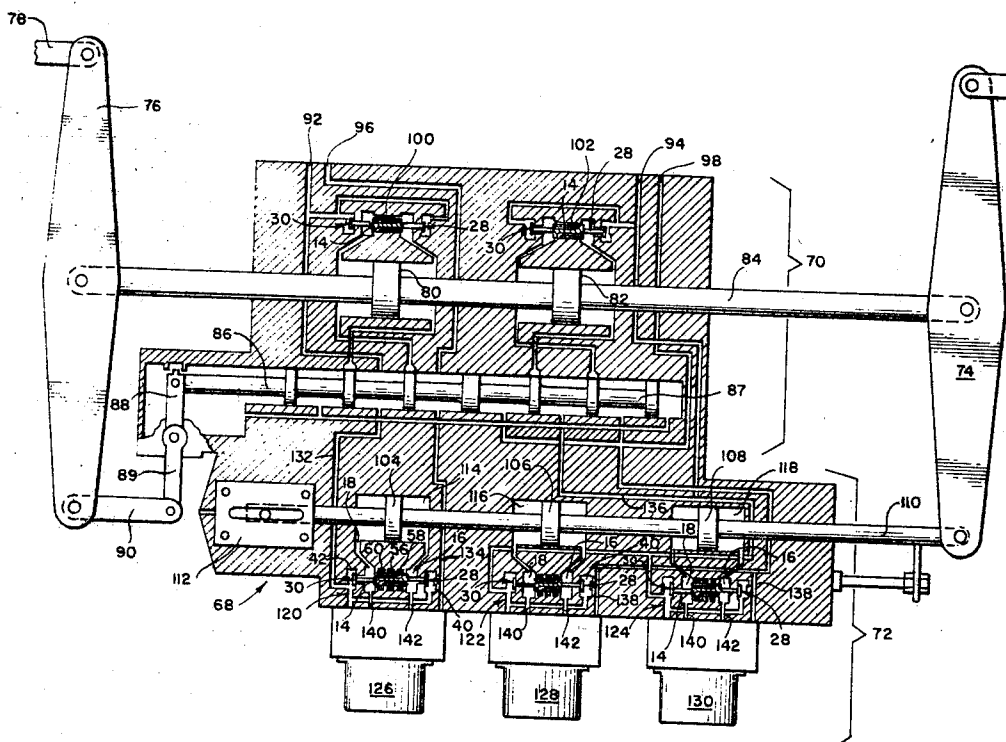
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ABSTRACT: A stability augmentation system is provided which has triple redundancy and is fail-functional. For each axis desired to be controlled, a hydraulic servocontrol mechanism is provided having three signal sources, and three actuator units. The actuator's pistons are connected together in tandem with a relief valve connected across each piston to prevent the piston of a failed unit from hampering the operation of the control. Means are provided to furnish a feedback signal to the servovalves controlling the pistons when pressure is being relieved across the piston to attempt to balance the output of the servovalve.

The hydraulic servocontrol mechanism may be combined with a hydraulic boost actuator comprising two tandem pistons having their pressure differential limited by boost relief valves and being controlled by a spool valve to provide an integrated hydraulic control actuator having a single output member.

The relief valve has a slider with a conduit therein positioned between a pair of fluid channels. The conduit is closed when the slider is centered in the valve and is open when the slider is not centered. Centering plungers, held in place by hydraulic pressure, normally hold the slider in its centered position. When the pressure differential across the slider exceeds the pressure of one of the plungers, the slider moves from its centered position.



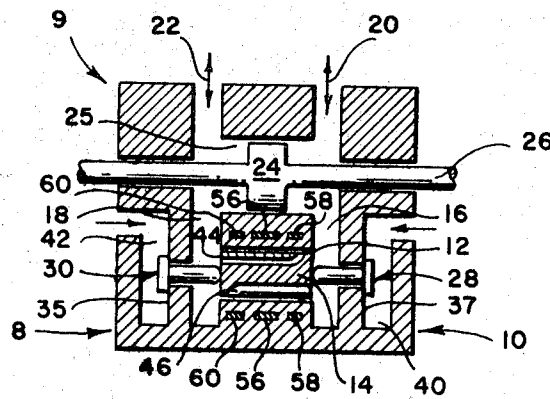


FIG. 1

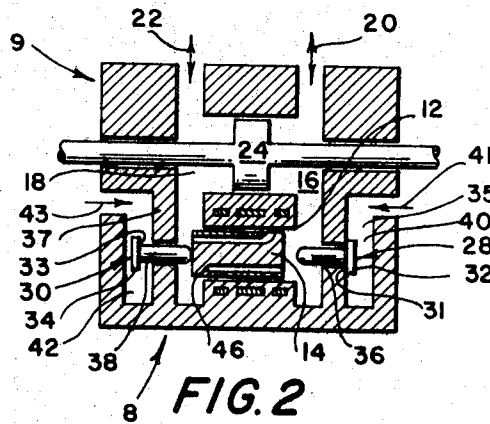


FIG. 2

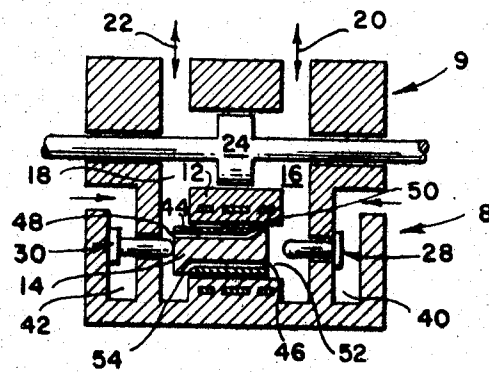


FIG. 3

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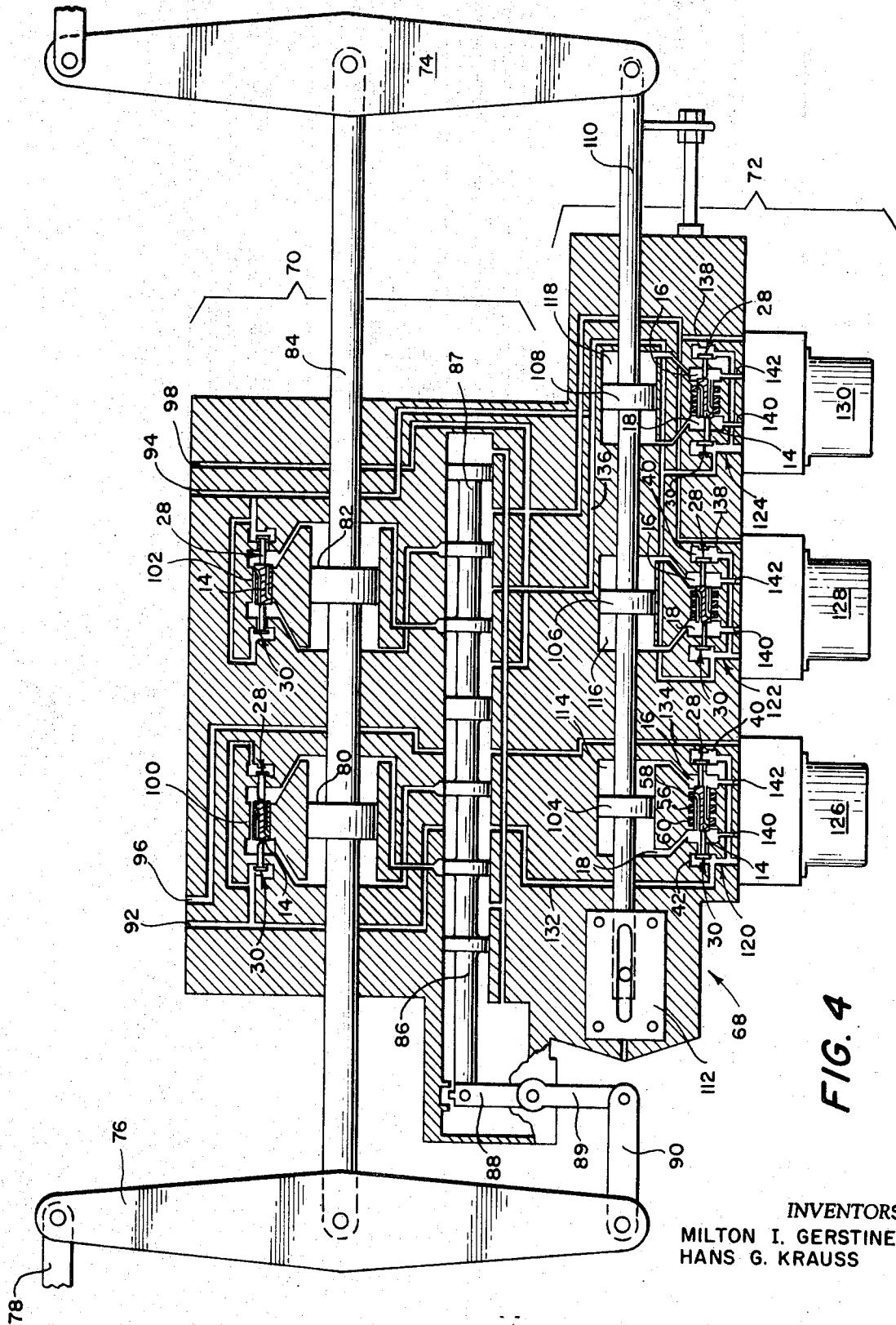


FIG. 4

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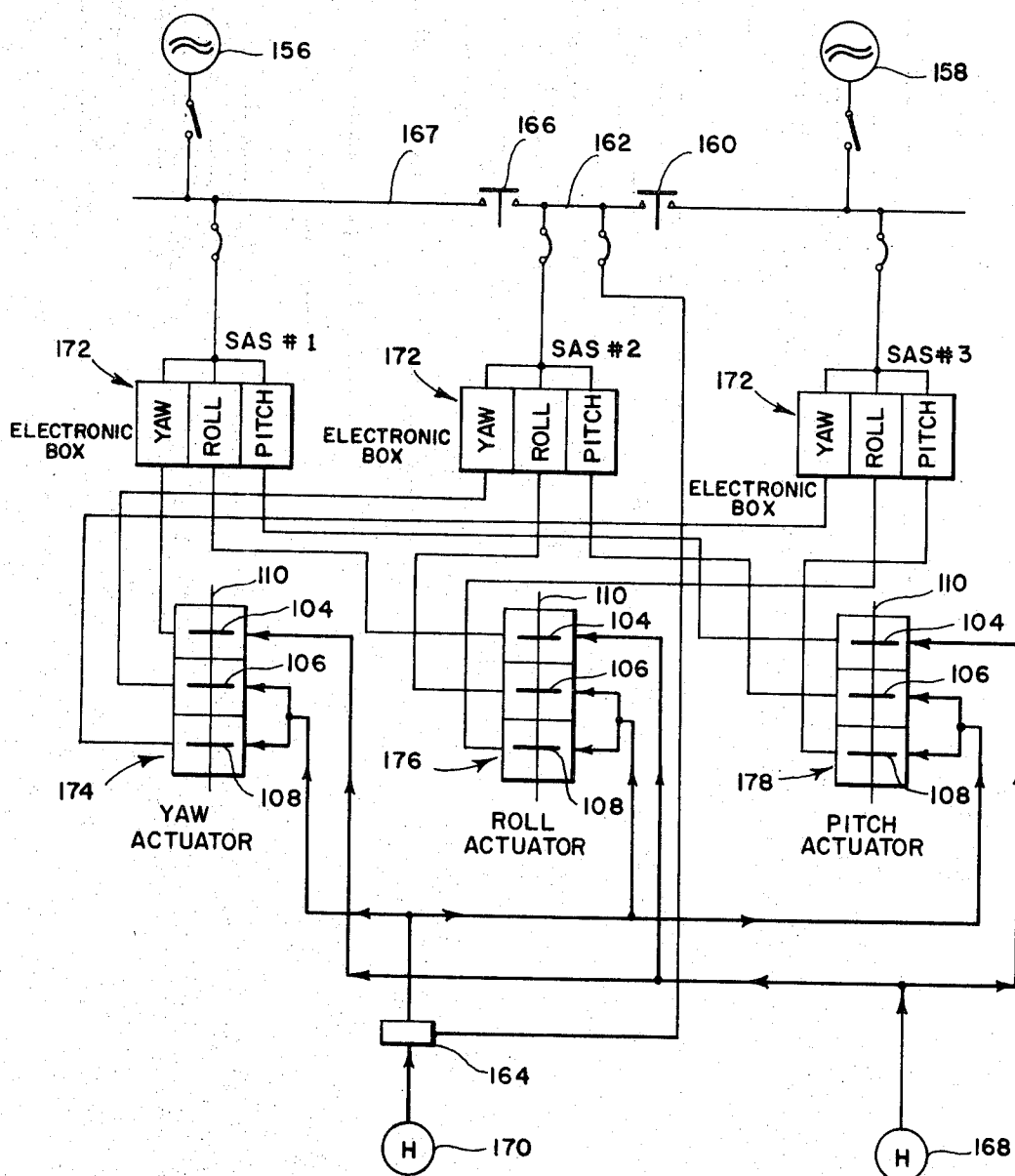


FIG. 5

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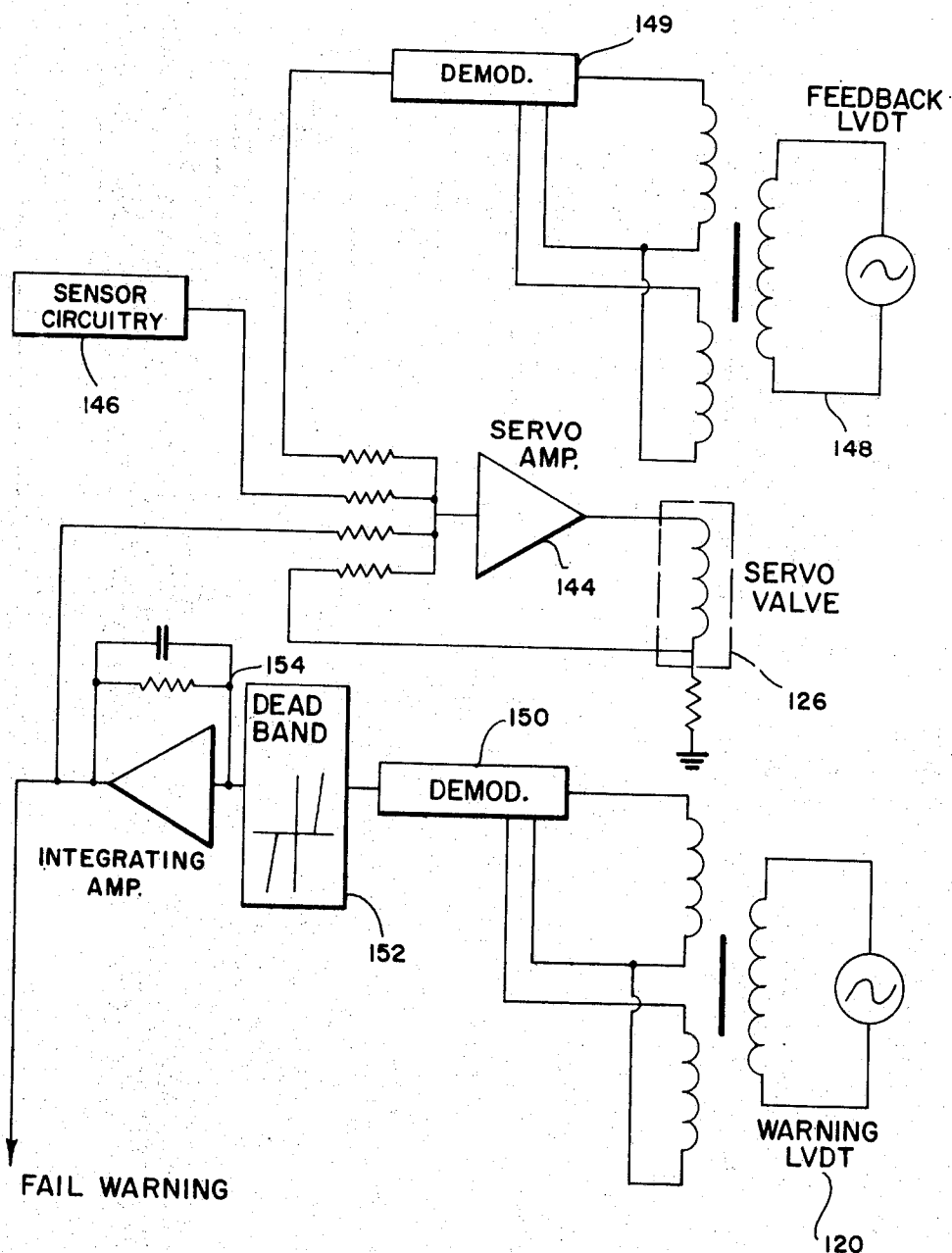


FIG. 6

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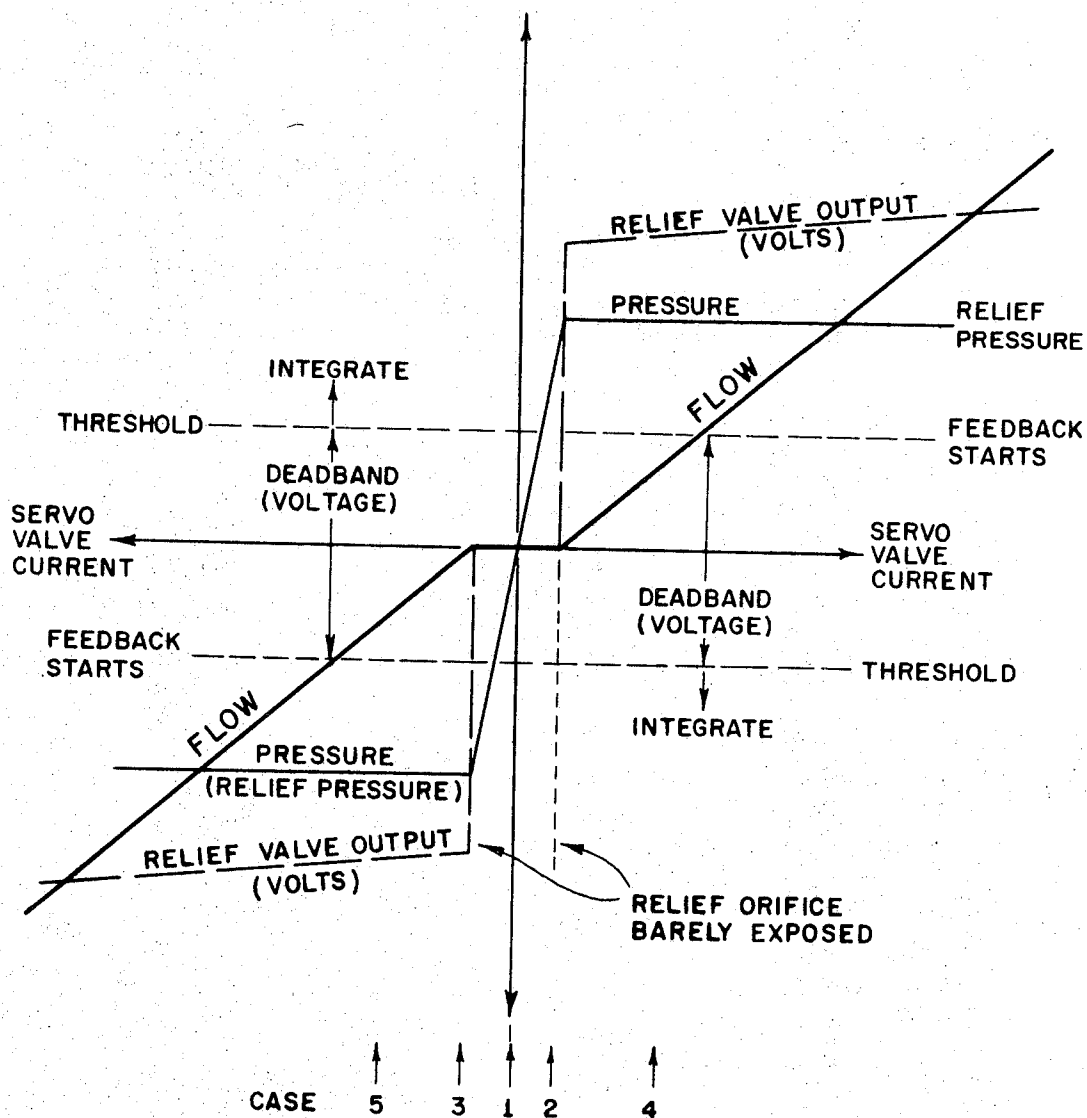


FIG. 7

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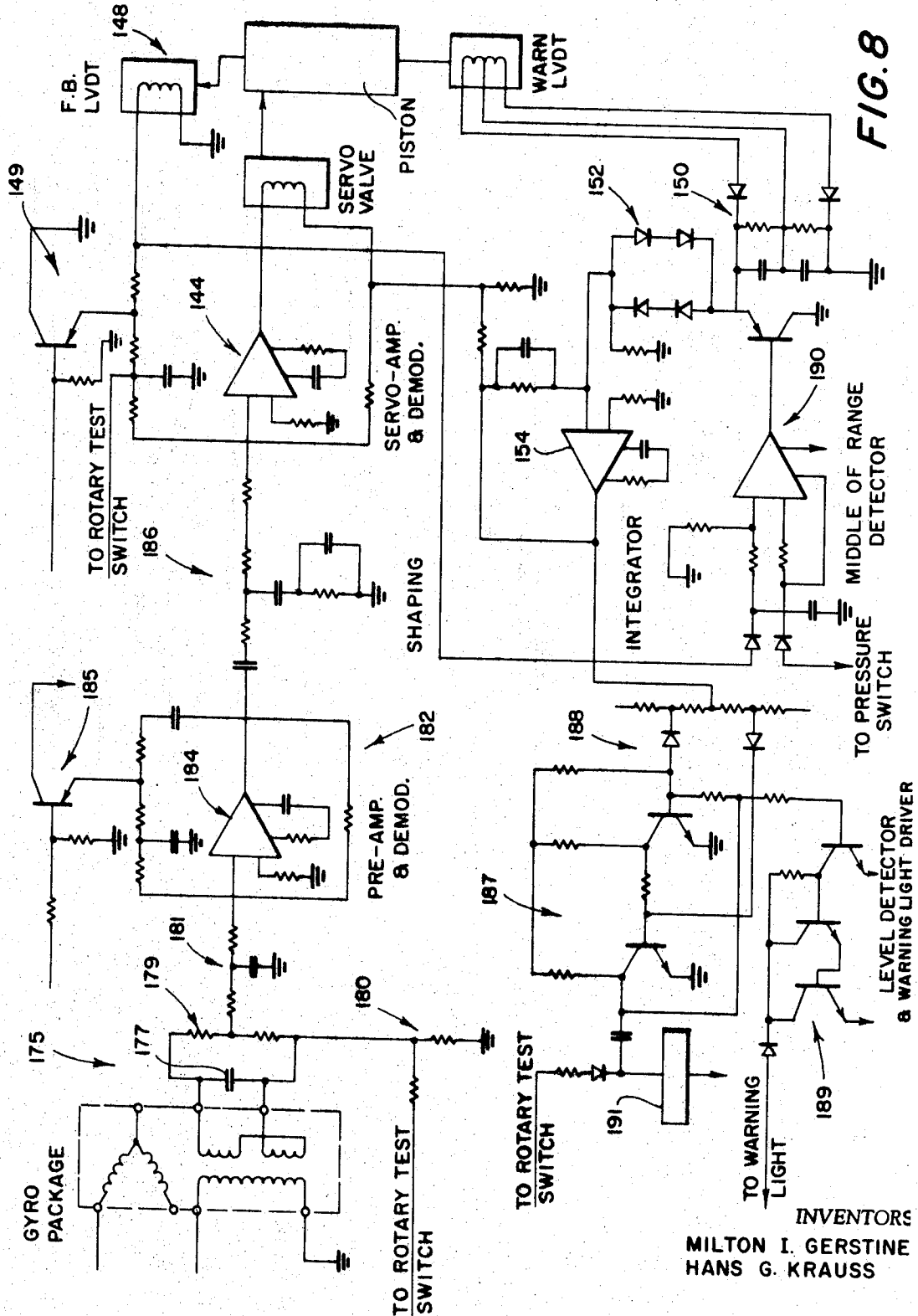


FIG. 8

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STABILITY AUGMENTATION SYSTEM

This invention relates to a stability augmentation system having improved reliability. More particularly this invention relates to an improved stability augmentation system having an improved hydraulic servocontrol mechanism, an improved relief valve, an improved integrated hydraulic actuator, and improved electronics.

The ever increasing demands for reliability and sensitivity in controls for dirigible crafts such as airplanes, helicopters, and the like had led in turn to increased demand for reliable stability augmentation systems.

It is therefore a primary object of this invention to provide an improved stability augmentation system.

It is an object of one aspect of this invention to provide an improved hydraulic servocontrol mechanism.

It is an object of another aspect of this invention to provide an improved hydraulic relief valve.

It is still another object of this invention to provide an improved integrated control actuator.

It is a further object of this invention to provide improved electronics for a stability augmentation system.

Additional objects and advantages of this invention will be set forth in part in the description which follows and in part will be obvious from the description or may be learned by the practice of the invention.

The objects and advantages are realized and attained by means of the instrumentalities and combinations particularly pointed out in the appended claims.

The hydraulic servocontrol mechanism of this invention comprises a plurality of signal sources, a hydraulic servoactuator having an output member responsive to signals from the signal sources, the actuator having a plurality of individual units with each unit including: a cylinder, a piston positioned in the cylinder and connected to the output member, a servovalve in communication with the cylinder for supplying hydraulic fluid to the cylinder, a relief valve in communication with the servovalve and the cylinder for limiting the differential pressure across the piston in the cylinder for limiting and for releasing the pressure differential by permitting the fluid flow between the two sides of the piston when the differential pressure exceeds a preselected maximum, each of the relief valves including sensing means for sensing when the pressure differential across the piston exceeds a preselected maximum, and means responsive to the sensing means for supplying a signal to the servovalve to attempt to cause the servovalve to decrease the pressure differential across the piston.

Preferably, each unit of the actuator has means responsive to the sensing means for providing a warning signal to indicate a failure when the pressure differential across the piston is greater than a preselected maximum.

The sensing means is preferably comprised of a linear variable differential transformer integrally formed in the relief valve.

The integrated hydraulic actuator of this invention includes the hydraulic servomechanism of this invention in combination with a hydraulic boost actuator which includes a plurality of individual units, each of said units having a cylinder, a piston mounted in the cylinder and connected to the output member of the servomechanism, and a spool valve in fluid communication with the cylinder for operating the piston and a common input member to said piston and said spool valve to permit movement of the input member to cause the spool valve to provide a boost to the piston to assist in movement of the output member.

In the preferred embodiment, the hydraulic servocontrol mechanism is a part of a stability augmentation system for a dirigible craft, the signal sources include means for supplying signals in response to a change in a condition of the craft, and the actuator acts to stabilize the craft in response to the signals from the signal source. In this embodiment it is preferred that the hydraulic servocontrol mechanism include at least three signal sources, and at least three actuator units to provide a fail-functional capability to this mechanism.

The relief valve of this invention comprises a valve body, a slider mounted in the valve body, a pair of fluid channels, one in communication with each end of the slider, at least one conduit in the slider with the conduit being closed when the slider is centered in the valve and the conduit being open to flow of fluid between the channels when the slider is moved from its centered position, a pair of centering plungers, one mounted at each end of the slider for normally holding the slider in its centered position, the centering plungers being held in position by hydraulic pressure acting on one end of each of the plungers whereby the centering plungers hold the slider in its centered position when the pressure differential across the slider is less than a preselected maximum and permit the slider to slide from its centered position toward the channel having the lesser pressure when the pressure differential across the slider is larger than the preselected maximum.

Preferably, the relief valve includes sensing means for sensing when the pressure differential across the slider exceeds a preselected maximum.

The sensing means is preferably a linear variable differential transformer integrally formed in the relief valve with a primary winding and two secondary windings positioned adjacent the slider with the slider being made of magnetic material and with the secondary windings being connected so that there is no output from the transformer when the slider is centered but an output is created when the slider is moved from its centered position.

In the preferred embodiment, the slider has two conduits with one being open when the slider is moved in one direction and the other being open when the slider is moved in the other direction.

The invention consists of the novel parts, constructions, arrangements, combinations, and improvements shown and described.

The accompanying drawings which are incorporated in and constitute a part of this specification illustrate one embodiment of the invention, and together with the description, serve to explain the principles of the invention.

Of the drawings:

FIG. 1 is a schematic representation partially in section of a relief valve constructed in accordance with the teachings of this invention, in association with a cylinder and piston;

FIG. 2 is the same as FIG. 1 with the slider of the relief valve moved slightly to the left;

FIG. 3 is the same as FIG. 1 with the slider moved entirely to the left;

FIG. 4 is a schematic representation partially in section of a hydraulic servocontrol mechanism constructed in accordance with the teachings of this invention;

FIG. 5 is a block diagram of a stability augmentation system constructed in accordance with the teachings of this invention illustrating the electrical and hydraulic power supplies and their distributions;

FIG. 6 is a single channel block diagram illustrating the circuitry associated with the servovalve and relief valve of one of the units of the hydraulic servocontrol mechanism of FIG. 4;

FIG. 7 is a graph showing the hydraulic relationships, as a function of servovalve current, existing in the relief valve when the slider is in the positions illustrated in FIGS. 1, 2, and 3 as a result of combined operation of the relief valve and servovalve; and

FIG. 8 is a circuitry schematic for a single channel of a stability augmentation system constructed in accordance with this invention.

It is to be understood that both the foregoing general description and the following detailed description are exemplary and explanatory but are not restrictive of the invention.

Reference will now be made in detail to the preferred embodiments of this invention, examples of which are illustrated in the accompanying drawings.

In FIGS. 1—3, the relief valve of this invention, generally 8, which may also be termed a limit valve, is shown in conjunction with a hydraulic device, generally 9. The parts of relief

valve 8 and hydraulic device 9 are identical in each of FIGS. 1—3. Though each element has a reference numeral in at least one of the figures, numerals are not shown for each element in each figure so that the numerals and lead lines will not obscure the figures. Relief valve 8 comprises a valve body, generally 10, a cavity 12 within the body, and a slider 14 slidably mounted in cavity 12.

The valve body 10 also contains a pair of fluid channels 16 and 18, respectively, one in communication with one end of slider 14 and one in communication with the other end of slider 14. Channels 16 and 18 are also in communication with a source of hydraulic fluid indicated by arrows 20 and 22. In the embodiment illustrated, relief valve 8 is to limit the pressure differential in hydraulic device 9. Thus, channels 16 and 18 are in communication with opposite sides of a piston or ram 24 mounted in a cylinder 25 of hydraulic device 9. Piston 24 is mounted on a rod 26 which may be connected to an element to be controlled by the source of hydraulic fluid.

In accordance with the invention, a pair of centering plungers 28 and 30 are mounted in valve body 10 at each end of slider 14 for normally holding the slider in its centered position with respect to channels 16 and 18. As here embodied and as best seen in FIG. 2, plungers 28 and 30 consist of head portions 32 and 34, respectively, and stem portions 36 and 38, respectively, and the inner ends of stems 36 and 38 normally abut the opposite ends of slider 14. (See FIG. 1). The shoulders 31 and 33 of plunger heads 32 and 34 rest against abutments 35 and 37 in the valve body 10, thus limiting the inward movement of the plungers.

In accordance with the invention the centering plungers are held in position against the valve slider by hydraulic pressure acting on one end of each of the plungers. If an appropriate hydraulic pressure is selected to give a desired force acting on the slider through the plungers, the plungers will hold the slider in its centered position when the pressure differential across the slider is less than a preselected maximum and will permit the slider to move from its centered position toward the channel having the lesser pressure when the pressure differential across the slider is larger than the preselected maximum. As here embodied, plungers 28 and 30 are positioned in fluid channels 40 and 42, respectively. Channels 40 and 42 lead to a second source of hydraulic fluid indicated by arrows 41 and 43, respectively.

In accordance with the invention, valve slider 14 has at least one conduit therein, with the conduit being closed when the slider is centered in the valve and being opened to flow fluid between channels 16 and 18 when the slider is moved from its centered position.

As here embodied, slider 14 has two curved conduits 44 and 46. As best seen in FIG. 3, conduit 44 has an opening 48 into channel 18 on the left end of slider 14 and an opening 50 in the side of slider 14 which, in the position illustrated in FIG. 1, is closed by the wall of cavity 12 in the valve body 10. Similarly, the conduit 46 has an opening 52 on the right end of slider 14 in communication with channel 16 and an opening 54 in the side of slider 14 which, when slider 14 is in the centered position, is closed by the wall of cavity 12.

In accordance with a preferred embodiment of the invention, the relief valve includes sensing means for sensing when the pressure differential across the slider exceeds a preselected maximum. As here embodied, the sensing means comprises a linear variable differential transformer integrally formed in the relief valve.

As best seen in FIG. 1, valve cavity 12 has a primary winding 56 centered between two secondary windings 58 and 60, and valve slider 14 is made of a magnetic material. The outputs of secondary windings 58 and 60 are connected together in opposition to each other so that when there is equal induced voltage in each winding there is no output from the transformer. When slider 14 moves from its centered position, the induced voltage in one of the secondary windings is reduced as the slider moves out of alignment with it. This reduction of voltage in one of the secondary windings in conjunction with

an increase in the induced voltage in the other of the secondary windings provides an output from the transformer. The further slider moves from its centered position, the greater is the output from the transformer. The transformer thus has no output when slider 14 is centered between channels 16 and 18 but does have an output, from secondary windings 58 or 60, which increases in magnitude as the slider moves progressively away from the centered position. The phase of the output voltage will depend on the direction in which the slider moves.

In operation of the valve, hydraulic fluid controlled by a servovalve, or the like, is passed into channels 16 and 18, as indicated by arrows 20 and 22. If the pressure is increased in channel 16, piston 24 of hydraulic device 9 will be driven to the left. This same pressure acts on the right end of valve slider 14 attempting to drive the slider to the left. However, the stem 38 of the plunger 30 is abutting against the surface of the left end of slider 14, and head 34 of plunger 30 is in communication with a constant source of hydraulic pressure in channel 42. Thus, as long as the force of the fluid pressure acting on the right end of slider 14 is less than the force created by the fluid pressure in channel 42 acting on the head of plunger 30, slider 14 will remain in its centered position.

In a similar manner, plunger 28 holds slider 14 in position against fluid pressure acting on the left end of slider 14 in channel 18. The centered position illustrated in FIG. 1 is the position of slider 14 when the pressure differential across the slider is within an acceptable limit. In this position there is no effective output from the secondary windings 58 and 60 of the linear variable differential transformer.

If there is excessive pressure on one side or the other of piston 24, as would result from a malfunction in the system supplying hydraulic fluid to the hydraulic device, the valve operates as shown in FIG. 3. Here, the pressure in channel 16 has built up to a point where the force exerted by the pressure acting on the right end of slider 14 is greater than the force exerted on slider 14 by plunger 30. Thus, the force of the hydraulic pressure acting on slider 14 has overcome the force of plunger 30, and slider 14 has moved to the left until its further leftward movement was restricted by the abutting of head 34 of plunger 30 against the wall of channel 42. In this position, opening 54 of conduit 46 is in communication with channel 18 and flow is permitted between channel 16 and channel 18 through conduit 46. Thus, the excessive pressure built up in channel 16 is relieved.

As will be apparent to those skilled in the art, if the excess pressure had been built up in channel 18 rather than channel 16, slider 14 would have slid to the right and conduit 44 of the slider 14 would have provided the communication between channels 16 and 18 to relieve the excess pressure.

With slider 14 moved from its centered position to the extreme left, the induced voltage in secondary winding 58 is reduced and the induced voltage in secondary winding 60 is increased. The amount that the voltage of secondary winding 60 exceeds the voltage of secondary winding 58, becomes the effective output of the transformer. The output from the transformer which varies linearly with the movement of the slider is at a maximum when the slider is at the extreme left, as illustrated in FIG. 3.

The output from the transformer may be utilized to actuate a warning signal, such as a light, to indicate the failure in the system, and may also be used as a feedback to the servovalve or the like to attempt to integrate the output of the servovalve to zero and thus to recenter the slider. In the case of a malfunction in the system, as indicated in FIG. 3, the output of the feedback to the servo would not be great enough to recenter the slider.

Referring now to the embodiment illustrated in FIG. 2, it may be seen that as in the case of the embodiment in FIG. 3, the pressure in channel 16 on the right end of slider 14 has caused slider 14 to move to the left overcoming the force acting on this left end of slider 14 through plunger 30. In this illustration, however, the pressure in channel 16 has not been so great as to drive slider 14 to the extreme left, but here the

pressure has only been sufficient to move slider 14 far enough to slightly open conduit 46. The slight opening of conduit 46 relieves the pressure differential between channels 16 and 18 sufficiently to establish an equilibrium between the forces acting on slider 14. In this circumstance, there may not be a failure of the system but rather a certain inaccuracy in the system which creates a pressure on one side of piston 24, which is slightly greater than the maximum desirable for hydraulic device 9, thus calling for pressure relief by the relief valve 8.

In this situation, as illustrated in FIG. 2, the movement of the slider to the left causes an output from the linear variable differential transformer. The output will not be as great as the output from the transformer when the slider is in the position illustrated in FIG. 3. However, the output, when used as a negative feedback to the servovalve controlling the supply of hydraulic fluid, has the effect of reducing the output of the servovalve and thus reducing the pressure differential across piston 24 (and slider 14) and returning slider 14 to its centered position.

Since the magnitude of the pressure differential across piston 24 was not great enough to cause slider 14 to slide to its extreme left position, the differential was not enough to be considered a failure. In this situation, it is desirable that the feedback to the servovalve be instituted, but it is not necessary to actuate the warning signal. Thus the warning signal is preferably connected to the transformer through means that will only actuate the signal when the slider has moved to the extreme left or right. This means will be described in more detail in conjunction with FIG. 6.

It should be understood that valve 8 may be used in many systems, and the hydraulic device 9 is shown in conjunction with valve 8 in the drawings only for the purposes of illustration.

The relief valve of this invention has particular utility in combination with a hydraulic servocontrol mechanism of the type used in stability augmentation systems for dirigible crafts such as helicopters and airplanes. By utilizing the relief valve in combination with a hydraulic servocontrol mechanism in a redundant type stability augmentation system, a stability augmentation system may be provided which has improved reliability and is fail-functional. Such a system is illustrated in FIGS. 4-8.

The hydraulic servocontrol mechanism of this invention includes a plurality of signal sources and a hydraulic servoactuator having an output responsive to signals from the signal sources. The signal sources will be described in more detail in the description of FIGS. 6 and 8. As embodied in FIG. 4, the hydraulic servoactuator is a part of an integrated control actuator. The integrated actuator of this invention includes a hydraulic boost actuator and a hydraulic servomechanism as described above with both having a common output member. As here embodied, the hydraulic boost actuator 70 is in the upper portion of the integrated actuator of FIG. 4 and the hydraulic servoactuator 72 is in the lower portion of the integrated actuator of FIG. 4. The displacement of the boost actuator and the hydraulic servo are summed by an output link 74 in the right-hand side of the FIG.

While the hydraulic servomechanism is advantageously used in conjunction with the boost actuator in an integrated actuator of this embodiment, it will be apparent to those skilled in the art that the hydraulic servoactuator may be used independently of the boost actuator without sacrificing its chief advantages.

The boost actuator of the integrated hydraulic actuator of this invention includes a plurality of individual units each having a cylinder, a piston mounted in the cylinder and connected to an output member, spool valve in communication with the cylinder for operating the pistons and a common input member connected to the piston and the spool valve to permit movement of the input member to cause the spool valve to provide a boost to the piston to assist in the movement of the output member.

As here embodied, the boost actuator 70 includes an input link 76 operated by a connector 78 from the pilot's cyclic control stick. Link 76 is connected to a pair of tandem pistons 80 and 82 mounted on a common shaft 84 which is connected at one end to input link 76 and at the other end to output link 74, which is connected to a control surface of the craft. Pistons 80 and 82 are driven by tandem spool valves 86 and 87 connected by links 88, 89 and 90 to input link 76. Two separate hydraulic supply channels 92 and 94 are connected to spool valves 86 and 87, respectively, and two separate hydraulic return channels 96 and 98 are connected to spool valves 86 and 87, respectively.

Two relief valves 100 and 102 are in fluid communication with pistons 80 and 82, respectively, to limit the pressure differential across piston 80 and 82, respectively. Relief valves 100 and 102 are virtually identical to the relief valves illustrated in FIGS. 1-3 and the numbers of the parts of the relief valves illustrated in FIGS. 1-3 have been retained. The only difference between relief valves 100 and 102 and the relief valves illustrated in FIGS. 1-3 is that in valves 100 and 102 the linear variable differential transformer is not necessary, and thus the electrical windings 56, 58, and 60 have been omitted.

The operation of the relief valve in the boost actuator is substantially as described above in the description of the operation of the relief valves themselves. When an excess pressure differential appears across the slider 14, the slider will push one or the other of centering plungers 28 and 30 out of the way and open a bypass conduit 44 or 46. The maximum force capability of the boost actuator is thereby limited to a predictable value to prevent excessive force from being exerted by the actuator which would damage the control system of the craft.

The lower portion of the integrated control actuator 68 is the hydraulic servoactuator 72 which as here embodied serves as an actuator for a triple redundant stability augmentation system.

In accordance with the invention, the hydraulic servoactuator has a plurality of individual units with each unit including a cylinder, a piston, a servovalve, a relief valve having means for sensing when the pressure differential across the piston exceeds a preselected maximum, and means responsive to the sensing means for supplying a signal to the servovalve to attempt to cause the servovalve to decrease the pressure differential across the piston.

As here embodied, there are three units and hence three pistons, 104, 106, and 108 connected on a common shaft 110. One end of shaft 110 is connected to output link 74, and the other end is mounted in an authority limiter 112. Authority limiter 112 limits the output of shaft 110 by physically limiting the distance that shaft 110 can move in either direction.

Pistons 104, 106, and 108 are mounted in three separate body cavities or cylinders 114, 116, and 118. In communication with each of the cylinders is a relief valve 120, 122, and 124, respectively, and a servovalve 126, 128, and 130, respectively. The servovalves may be of any standard type and preferably are jet pipe type servovalves.

As may be seen from the drawing, a hydraulic supply channel 132 extends between servovalve 126 and spool valve 86 at a point where spool valve 86 is connected to hydraulic supply channel 92. Similarly, a hydraulic return channel 134 extends between servovalve 126 and spool valve 86 at a point where spool valve 86 is connected to hydraulic return channel 96. In a similar manner, servovalves 128 and 130 are connected by hydraulic supply channel 136 to hydraulic supply channel 94 and by hydraulic return channel 138 to hydraulic return channel 98.

Relief valves 120, 122, and 124 are substantially identical to the warning relief valves illustrated in FIGS. 1-3 of this application and the corresponding parts of the valves are identical with the same numbers as the corresponding parts in FIGS. 1-3.

Since valves 120, 122, and 124 are identical, all of the reference numbers have not been shown on each of the valves so that the drawings may be more readily understood. Each of the valves has a slider 14, a pair of plungers 28 and 30, primary windings 56 and secondary windings 58 and 60.

Channels 16 connect the right end of sliders 14 to the right end of pistons 104, 106, and 108, respectively, and channels 18 connect the left end of sliders 14 to the left end of pistons 104, 106, and 108, respectively.

The relief valves 120, 122, and 124 differ slightly from the valves illustrated in FIGS. 1—3 in that in FIGS. 1—3, the piston associated with the relief valve is intermediate the source of hydraulic fluid and the relief valve, whereas the relief valve is intermediate the piston and the servovalves in the embodiment illustrated in FIG. 4. This change in arrangement has no effect on the operation of the relief valve other than that it requires additional channels to be connected between the relief valve and the servovalve.

As may be seen in FIG. 4, channels 140 and 142 connect servovalves 126, 128, and 130 to relief valves 120, 122, and 124, respectively. As may also be seen in FIG. 4, channels 40 and 42 which are in communication with the heads of plungers 28 and 30 are connected to hydraulic supply channel 92 through channel 132 in the case of relief valve 120 and are connected to hydraulic supply channel 94 through channel 136 in the case of relief valves 122 and 124.

As here embodied, relief valves 120, 122, and 124 act as warning relief valves and have four functions:

(1) In the event of a loss of system pressure, slider 14 is no longer held centered by centering plungers 28 and 30 and is free to bypass fluid around the associated piston 104, 106, or 108 and prevent a given piston from impeding the motion of the two operating pistons; (2) In the event of a malfunction, which incorrectly causes a large differential pressure across one of the pistons, the force developed across slider 14 will be sufficient to push one of centering plungers 28 or 30 out of the way and relieve pressure across the piston. As in the case of force limiters 100 and 102, this arrangement sets rather accurately the force capability of a given piston;

(3) In contrast with the force limiter, the warning relief valves have electrical windings surrounding the slider which sense the position of the slider and provide a warning to the operator any time the valve is sufficiently displaced from its center to indicate a failure somewhere in the system; and

(4) A signal from the electrical windings is also used, through negative feedback to the electrical network controlling the servovalve, to integrate the output of the servovalve to zero and recenter slider 14. Thus the warning relief valve tends to apply a balancing effect to the servovalve outputs.

In normal operation of the hydraulic servocontrol mechanism, one of the three pistons 104, 106, and 108 will be the controlling piston and the other two pistons will be opposing each other. The controlling piston will then supply whatever force is necessary to handle the external load (i.e., the control surface operated by link 74) plus any unbalance between the two opposing pistons.

The three operating positions of the relief valve as shown in FIGS. 1—3 occur when the servovalve current is at the corresponding points illustrated in FIG. 7. The condition of the relief valve illustrated in FIG. 1 is referred to as Case 1 in FIG. 7. It may be seen that Case 1 represents the controlling piston, and the servovalve current for this piston is essentially zero. The flow is essentially zero and is only sufficient to supply the necessary piston motion to the control surface and the pressure is relatively small, only enough to move the external load and provide a small force to round out any inequity of the forces produced by the two opposing pistons.

Since the piston just described is the controlling piston, then the other two pistons are the opposing pistons. The condition of these relief valves is referred to as Case 2 and Case 3 in FIG. 7. Case 2 is illustrated in FIG. 2. Thus one of the opposing pistons has a servovalve current corresponding to the plot

shown for Case 2 in FIG. 7, and the differential pressure across the warning relief valve associated with that piston is just sufficient to overcome the centering force of the pressure of centering plunger 30. As may be seen in FIG. 7, a voltage output from the relief valve has been created and at this position it will exceed the deadband if the servovalve does not recenter the slider; there has been a limited amount of servovalve current; the relief orifice is barely exposed; there has been very little fluid flow; and the pressure is increased to the relief pressure valve.

The third valve which is Case 3 will be exactly the opposite of Case 2, with the differential pressure being just sufficient to overcome the centering force of the pressure of centering plunger 28 to move slider 14 to the right rather than to the left as in Case 2. Thus, here again, a voltage output from the relief valve has been created, and at this position, it will exceed the deadband if the servovalve does not recenter the slider, there has been a limited amount of servovalve current (having a phase opposite to that of Case 2), the relief orifice is barely exposed, there has been very little fluid flow, and the pressure has increased to the relief pressure valve (but it is in the opposite direction of that of Case 2). Since Case 3 is merely the opposite of Case 2, there is no figure specifically illustrating the valve of Case 3.

The current to the servovalve controlling the piston in Case 3 will be essentially equal and opposite to the current to the servovalve controlling the piston in Case 2. The flow through both relief valves will be small and only enough to keep the pressure constant across their respective pistons.

It is to be understood that in Cases 2 and 3, the system is not considered to be in a failure condition, but rather this is the normal condition with two opposing pistons and one controlling piston. The reason that all three pistons are not operating identically is due to slight differences caused in the sensing means, the circuitry, the hydraulics, or the servovalves. Thus at a given time the output from any one servovalve will be slightly different from that of the others. Since there are three servovalves all having different outputs but with each being in the normal range of operation, it is obvious that shaft 110 cannot be driven to three separate positions. Accordingly, the shaft will be moved to the position demanded by the servovalve which has an output between the output of the other two servovalves.

To take an example for illustration purposes only, assume that the output from servovalve 126 at a given instance would tend to drive piston 104 one unit to the right, the output from servovalve 128 would tend to drive piston 106 one unit to the left, and the output from servovalve 130 would tend to drive piston 108 one-half unit to the left. Since the output of servovalve 130 is between the outputs of servovalves 126 and 128, piston 108 will become the controlling piston and shaft 110 will move one-half unit to the left. However, since servovalve 126 is still attempting to drive piston 104 one unit to the right of its original position and thus $1\frac{1}{2}$ units to the right of its ultimate position, pressure will continue to build up on the left end of piston 104, and also on the left end of slider 14 of warning relief valve 120 until the pressure of plunger 28 is overcome sufficiently to open conduit 44, permitting fluid to flow sufficiently to keep the pressure constant across piston 104. Warning relief valve 120 is then in the position of Case 3 in FIG. 7.

Since servovalve 128 is attempting to drive piston 106 one unit to the left of its original position and a half unit to the left of its ultimate position, pressure will continue to build up on the right end of piston 106 and slider 14 of warning relief valve 122. When the pressure is sufficient to overcome the pressure of centering plunger 30, slider 14 of warning relief valve 122 will slide to the left far enough to open conduit 46, and permit flow of sufficient fluid to keep the pressure constant across the piston 106. Warning relief valve 122 will then be in the position of Case 2, as illustrated in FIG. 7.

It is to be further understood that since the signals coming from the signal sources to the servovalves will be constantly

changing, the output from the servovalves will be constantly changing, and the piston which is controlling may be constantly shifting from one piston to the others. Thus the controlling piston may be 106 at one instance, it may be 108 at the next instance, and still at another instance 104. But in each case the remaining two pistons will be opposing each other.

In the case of a failure in either the hydraulics, the electronics controlling a given servovalve, the servovalve, or the actuator itself which causes an excessive buildup of hydraulic pressure across a given piston, a warning relief valve will operate as illustrated in FIG. 3, which is referred to as Case 4 in FIG. 7.

As may be seen, slider 14 has been moved to the extreme left completely opening conduit 46. Under this condition there will be considerable flow through the relief valve, but the pressure differential across piston 24 will be reduced to an amount that can be substantially offset by one of the other pistons. The signal from the output of the electrical pickoff will try to balance the output of the servoamplifier, but will not have sufficient capacity. This inability of the servovalve to be balanced by the signal from the output of the warning relief valve will cause a failure indication to be presented to the operator in the form of a warning signal.

Referring to Case 4 in FIG. 7, it can be seen that there has been considerable servovalve current, the voltage from the relief valve has exceeded the deadband and is in the gradually increasing area to the right of the maximum voltage for Case 2. In this position, there is considerable fluid flow, but the relief orifice has prevented the pressure from building up beyond the relief pressure value.

It is to be understood that a failure could also occur causing slider 14 to move to the right rather than to the left. However, other than the location of the slider and the use of the other conduit in the slider for pressure release, the result will be the same as illustrated in FIG. 3. This condition is illustrated as Case 5 in FIG. 7 and is simply the inverse of Case 4.

With one of the warning relief valves in the condition illustrated in FIG. 3 and shown as Case 4 in FIG. 7, the remaining operational pistons can still control the output of shaft 110 without interference from the failed unit. There are now only two operational pistons, and one piston will be offsetting the failed piston while the other will be the controlling piston. Since the warning relief valve has restricted the pressure differential across the failed piston to a preselected maximum, one of the remaining operating units will be able to substantially completely offset the pressure of the piston of the failed unit with the piston of the remaining unit acting as the controlling piston.

FIG. 6 is a single channel block diagram typical of the three channels controlling the hydraulic servoactuator 72 and illustrates the circuitry associated with the servoloop. For purposes of illustration it may be presumed that this servoloop operates the unit containing servovalve 126.

The four inputs to the servoamplifier 144 controlling servovalve 126 are (1) the signal from the sensor circuitry 146 which has been derived from a sensor such as a rate gyro and passed through appropriate shaping network; (2) the signal from a feedback linear variable differential transformer (LVDT) 148 which has passed through a demodulator 149 and provides information regarding the displacement of shaft 110; (3) a voltage proportional to the current through servovalve 126 for electronic stabilization of servoamplifier 144; and (4) a signal which has been derived from the warning LVDT in relief valve 120 and passed through a demodulator 150, a threshold circuit 152 and an integrating amplifier 154. For the controlling piston, the signal derived in the warning LVDT output is 0, and for the two opposing pistons (Cases 2 and 3) this voltage is just enough to balance out any unbalance in the other three inputs as long as their sum does not exceed a permissible unbalance of the channel.

The warning LVDT signal is passed through demodulator 150 and then put through a threshold circuit 152 which creates a deadband so that no current from the demodulator

will enter into the integrating amplifier 154, unless the signal from the warning LVDT is of a certain minimum value. This minimum value is set to require that slider 14 of the warning relief valve must move a set distance away from its center position. As can be seen in FIG. 7, slider 14 moves sufficiently to create a signal which will exceed the threshold any time that pressure is relieved by flow of fluid through a conduit in slider 14.

The current constants are such that any time the threshold is exceeded, integrating amplifier 154 will integrate very rapidly to balance out servoamplifier 144 to thereby balance out the differential pressure across the warning relief valve and to return slider 14 to its center position. Once slider 14 has returned to a position where the output of the warning LVDT no longer exceeds the threshold, integrator 154 balances very slowly until the servoamplifier unbalance again is sufficient to cause the warning relief valve slider 14 to depart from its center position and the output of LVDT to again exceed the threshold.

In normal operations, therefore, the output of the warning LVDT will be oscillating in and out of the deadband and the output of the integrating amplifier 144, although varying at the rate of oscillation of the warning relief valve slides will be relatively constant and will be an indication of the total unbalance of the servoloop. When this unbalance exceeds what is defined as the maximum permissible unbalance for normal operating channels, the failure warning circuit will be actuated.

As described above, the system is fail-functional and if any one channel should fail, the other two channels will continue to operate. In order to achieve fail-functional operation a guaranteed source of electrical power and hydraulic pressure are required. A power schematic for the triple redundant stability augmentation system (SAS) is illustrated in FIG. 5.

As can be seen in FIG. 5, each of the three SAS has an electronic package made of three separate channels, one each for the yaw, roll, and pitch axes. Three actuators 174, 176, and 178 are also provided, one for each of the yaw, roll, and pitch axes. Each actuator has three pistons connected to a common piston rod. These pistons correspond to pistons 104, 106, and 108 in the actuator illustrated in FIG. 4, and the piston rod corresponds to shaft 110.

One electronic package controls one piston in each of the three actuators. Thus the yaw channel of SAS 01 controls piston 104 of actuator 174, the roll channel of SAS 01 controls piston 104 of roll actuator 176, and the pitch channel of SAS 01 controls piston 104 of pitch actuator 178. In a similar manner, the yaw roll and pitch channels of SAS 02 control pistons 106 of the yaw, roll, and pitch actuators, respectively, and the yaw, roll, and pitch channels of SAS 03 control pistons 108 of the yaw, roll, and pitch actuators, respectively.

A first generator 156 feeds SAS 01. A second generator 158 feeds SAS 03 directly and, through relay 160 and essential bus 162, feeds SAS 02. If generator 156 fails, only SAS 01 loses power; if generator 158 fails, power would be instantaneously lost in both SAS 02 and SAS 03, but the loss of power in essential bus 162 would cause a shutoff valve 164, also connected to essential bus 162, to remove pressure from the sections of the actuators which were being supplied by SAS 02 and SAS 03. If the loss of power from generator 158 lasts for more than several seconds, a changeover relay 166 ties essential bus 162 to essential bus 167 and generator 156 to supply power to SAS 02 from generator 156 and also to supply power to shutoff valve 164, causing the shutoff valve to reopen.

The section of each hydraulic actuator which is part of SAS 01 is supplied from a hydraulic source 168 and the sections of each hydraulic actuator which are part of SAS 02 or 03 are supplied from a second hydraulic source 170 through shutoff valve 164.

It should be understood that if one of the hydraulic sources fails, the pressure on the centering plungers of the relief valves connected to the source will be eliminated and the valve sliders will be free to relieve any differential pressure across

the associated piston. Thus even if source 170 should fail, eliminating the pressure from two of the SAS, the remaining SAS can still operate the actuators since in each SAS the pistons controlled by the failed SAS cannot interfere with the piston from the operable SAS.

The preferred channel circuitry for a single channel is shown in FIG. 8. It consists of a gyro package 175 which includes the gyro itself with a capacitor 177 loading the output of the pickoff to provide AC power frequency compensation, a resistor network 179 to provide a temperature compensation and gain adjustment, and a resistor network 180 to inject a signal through the gyro for test purposes. The gyro signal is passed through a gain and phase shift network 181 and is then presented to a preamp demodulator 182. Preamp demodulator 182 utilizes an integrated amplifier 184 to amplify the signal first as an AC signal and then after the signal is demodulated with a chopper demodulator 185 to amplify the signal again in a reflex fashion as a demodulated or DC signal. The signal goes then to the shaping network 186.

The shaped signal is presented to another integrated amplifier which corresponds to amplifier 144 in FIG. 6. The function of the servoloop has already been described in connection with FIG. 6 and will not be repeated here.

As seen in both FIGS. 6 and 8, integrator 154 provides a signal to servoamplifier 144. Although integrator 154 introduces a signal to the servoamplifier, the effect of the signal can be no more than to cause a slight unbalance to the channel even if a malfunction should cause integrator 154 to go hardover. Since integrator 154 saturates a little beyond the voltage required to balance the maximum permissible channel unbalance, the effect of an integrator malfunction would be at most to cause an unbalance to a normal channel only slightly in excess of the maximum permissible under normal circumstances.

It can be seen that only the circuitry across the top of FIG. 8, namely the gyro package 175, the preamp demodulator 182, the shaping network 186, the servoamplifier 144, and the feedback demodulator 149 are required for the normal fail-functional operation of a triple SAS. The other circuits shown in FIG. 8 are only for the purpose of balancing, pilot warning, or maintenance guidance. Although they could cause a false warning in the event of a malfunction, no failure of this other circuitry could cause a malfunction of the type that would contribute to unacceptable flight characteristics.

The operation of the balance and warning circuits is as follows. The signal from integrator 154, besides being fed back through servoamplifier 144 for balancing the input voltages to the servovalve, is also fed to a magnitude sensing circuit 188 which operates an indicator flag on the front of the built-in test equipment (BITE) 191 unit for informing the maintenance mechanic of channel failure. Operation of this circuit is such that if the output of the integrator 154, either positive or negative, exceeds a certain value, a bistable circuit 187 will suddenly change its state and cause a flag to appear to identify the malfunctioning channel. At the same time another circuit, namely the warning light driver circuit 189, will light a warning light on the pilot's master warning control.

Any malfunction, whether in the electronic power supply, hydraulic power supply, or in the electronic packages themselves, will cause the warning light to go on. However, only malfunctions within the electronic unit or the associated hydraulic actuator section will cause the warning flag to appear. In order to prevent operation of the warning flag in the event of hydraulic pressure failure, a grounding signal is sent from a hydraulic pressure switch to the middle of the range detector 190.

The primary function of the middle of the range detector is, however, to inhibit a failure warning when a large signal from the gyro demands that the actuator go hardover as part of its normal function. Should this occur, all three channels in a given axis would display warning flags and light the warning light unless special means are provided to prevent this.

Since hardovers are quite possible during normal operations, the middle of the range detector 190 is used as a means of preventing integration by integrator 154 unless the actuator is a predetermined distance from its center of operation. Whenever the displacement of the actuator exceeds this predetermined distance, as indicated by the voltage from feedback LVDT 148, middle of range detector 190 grounds out the signal which is coming from the warning LVDT demodulator 150 and being presented to the threshold circuit 152, and prevents integration thereby preventing the warning flag from appearing and the warning light from going on.

In accordance with this invention, a new and improved hydraulic servocontrol mechanism has been provided which includes a hydraulic servoactuator having a plurality of pistons controlled by an equal number of electronic channels with each of the pistons being operated by hydraulic fluid passing through a control such as a servovalve and having an associated relief valve connected thereto to provide bypass of fluid when the pressure differential across the piston exceeds a preselected maximum value.

In this system, the warning relief valve also acts to provide a feedback signal to the servovalve to attempt to balance the output of the servovalve, and thus return the pressure differential across the piston to within the normal operating range. The warning relief valve may also provide a signal to the operator of the craft utilizing the stability augmentation system to indicate when the pressure differential has exceeded the maximum allowed value.

The hydraulic servocontrol mechanism of this invention has particular utility in a fail-functional stability augmentation system.

Another aspect of this invention relates to a new and improved relief valve which can provide an equalizing of pressure across an associated piston when the pressure differential across the piston has exceeded a predetermined maximum. In one embodiment of the invention, the relief valve provides a feedback signal to the source of hydraulic fluid to reduce the excessive pressure differential across the piston. The pressure relief valve may also act as a warning relief valve by supplying a signal indicating the existence of excessive pressure differential across the piston.

A further aspect of this invention is the provision of a new and improved integrated hydraulic actuator including both a boost actuator system to provide a hydraulic boost to the operation of the output member, and a servocontrol mechanism to provide operation of the output member by a stability augmentation system.

We claim:

1. A hydraulic servocontrol mechanism comprising:

- a. a plurality of signal sources;
- b. a hydraulic servoactuator having an output member responsive to signals from said signal sources, said servo actuator having a plurality of individual units with each unit including:
 1. a cylinder;
 2. a piston and a piston rod positioned in said cylinder, said piston connected to said output member by said rod;
 3. a servovalve in communication with said cylinder by means of channels for supplying hydraulic fluid to said cylinder;
 4. a differential pressure responsive relief valve in communication with said servovalve and said cylinder for limiting the differential pressure across said piston in said cylinder said relief valve having passage means therein for permitting fluid flow between the two sides of said piston when the differential pressure exceeds a preselected maximum, each of said relief valves including sensing means for sensing the position of said relief valve when the pressure differential across the piston exceeds a preselected maximum; and
 5. means responsive to said sensing means for supplying a signal to said servovalve to cause said servovalve to decrease the pressure differential across said piston.

2. The servocontrol of claim 1 including means responsive to said sensing means for providing a warning signal to indicate a failure when said pressure differential across said piston is greater than a preselected maximum.

3. The servocontrol of claim 1 wherein said relief valve comprises a valve body, a slider mounted in said valve body, a pair of fluid channels, each providing communication between one end of said slider and one end of said piston, said slider having at least one conduit therein which is closed when the slider is centered and is open to flow of fluid between said channels when said slider is moved from its centered position, a pair of centering plungers, one mounted at each end of said slider for normally holding said slider in its centered position, said plungers being held in position by a constant hydraulic pressure acting on one end of each of said plungers whereby when the pressure differential across said piston exceeds a preselected maximum, the force acting on said slider overcomes the centering pressure of said plungers and moves said slider in the direction of the channel having the lesser pressure.

4. The servocontrol of claim 3 wherein said sensing means is comprised of a linear variable differential transformer integrally formed in said relief valve.

5. The servocontrol of claim 4 wherein said transformer includes a primary winding and two secondary windings positioned adjacent said slider of said relief valve, said slider is made of a magnetic material and said secondary windings are connected so that signals do not issue from said transformer when said slider is centered but a signal issues from said transformer when said slider is moved from its centered position.

6. The servocontrol of claim 4 wherein said signal from said transformer is in the form of a negative feedback signal, said means which is responsive to said transformer signal comprises a demodulator, a threshold circuit, an integrating amplifier, and a servoamplifier, connected to said servovalve for supplying said signal thereto to integrate the output of said servovalve to zero and to thereby recenter said slider.

7. The servocontrol of claim 6 including a warning means and wherein said signal from said transformer energizes said warning means to indicate a failure.

8. The servocontrol of claim 7 wherein there are at least three signal sources, and at least three actuator units.

9. An integrated hydraulic actuator comprising:

A. a hydraulic servocontrol mechanism including:

1. a plurality of signal sources;
2. a hydraulic servoactuator having an output member responsive to signals from said signal sources, said servoactuator having a plurality of individual units with each unit including:
 - a. a cylinder;
 - b. a piston and a piston rod positioned in said cylinder, said piston connected to said output member by said rod;
 - c. a servovalve in communication with said cylinder by means of channels for supplying hydraulic fluid to said cylinder;
 - d. a differential pressure responsive relief valve in communication with said servovalve and said cylinder for limiting the differential pressure across said piston in said cylinder and for releasing the pressure differential said relief valve having passage means for permitting fluid flow between the two sides of said piston when the differential pressure exceeds a

preselected maximum, each of said relief valves including sensing means for sensing the position of said relief valve when the pressure differential across the piston exceeds a preselected maximum; and

e. means responsive to said sensing means for supplying a signal to said servovalve to cause said servovalve to decrease the pressure differential across said piston;

B. a hydraulic boost actuator including:

1. a plurality of individual units each of said units having:
 - a. a cylinder;
 - b. a boost piston mounted in said cylinder and connected to said output member;
 - c. a spool valve in communication with said cylinder for operating said boost piston; and
 - d. a common input member connected to said boost piston and said spool valve to permit movement of said input member to cause said spool valve to provide a boost to said boost piston to assist in movement of said output member.

10. The integrated hydraulic actuator of claim 9 including a relief valve operatively connected to each of said cylinders of said boost actuator for preventing excessive pressure buildups across said boost pistons.

11. The integrated hydraulic actuator of claim 10 wherein said relief valves in said hydraulic servocontrol mechanism and in said hydraulic boost actuator comprise a valve body, a slider mounted in said valve body, a pair of fluid channels, each providing communication between one end of said slider and one end of the piston associated therewith, said slider having at least one conduit therein which is closed when said slider is centered and is open to flow of fluid between said channels when said slider is moved from its centered position, a pair of centering plungers, one mounted at each end of said slider for normally holding said slider in its centered position, said plungers being held in position by a constant hydraulic pressure acting on one end of each of said plungers whereby when the pressure differential across said pistons exceeds a preselected maximum the force acting on said slider overcomes the centering pressure of one of said plungers and moves the slider in the direction of the channel having the lesser pressure.

12. The integrated hydraulic actuator of claim 11 wherein said sensing means of said servocontrol is comprised of a linear variable differential transformer integrally formed in said relief valves in said servocontrol and wherein said transformer includes primary windings and two sets of secondary windings positioned adjacent said slider of said relief valve, said slider is made of a magnetic material, and said secondary windings are connected so that no signals issue from said transformer when said slider is centered but a signal issues from said transformer when said slider is moved from its centered position.

13. The integrated hydraulic actuator of claim 12 wherein said signal from said transformer is in the form of a negative feedback signal, said means, which is responsive to said transformer signal comprises a demodulator, a threshold circuit, an integrating amplifier, and a servoamplifier, connected to said servovalve for supplying said signal thereto to integrate the output of said servovalve to zero and to thereby recenter said slider.

14. The integrated hydraulic actuator of claim 13 including a warning means and wherein said signal from said transformer energizes said warning means to indicate a failure.