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Description

This invention relates to a servo mechanism for use in a fluid servo system and, more particularly, to aircraft flight control servo systems. More specifically, the invention relates to a servo actuator control/damping mechanism which utilizes and combines the functions of an electro-mechanically driven servo valve to achieve fluid flow and actuator (ram) load control even after loss of fluid power.

Fluid servo systems are used for many purposes, one being to position the flight control surfaces of high performance aircraft. In such an application, the servo system desirably should provide for control and damping of flight control surface displacements or flutter after loss of fluid power. Otherwise, aircraft damage or loss of control may result.

It is generally known from US—A—4 351 357 to provide a hydraulic servo control system comprising a piston and cylinder device a mechanically operated servo valve, a compensator and check valves.

According to the present invention in one aspect there is provided a servo mechanism for use in a fluid servo system for controlling a fluid powered ram actuator having opposed pressure surfaces, comprising a servo valve including a valve member selectively positionable therein to provide variable fluid pressure and return flow orifices for metering fluid flow from a pressure port and pressure passage to either pressure surface of the ram and return flow from the other pressure surface of the ram to a return passage and return port through actuator passages between the servo valve and ram, a compensator between the return passage and return port, and additional passages between the return passage and actuator passages containing check valves, characterized by an electro-mechanical mechanism operative independently of fluid pressure to controllably position the valve member to effect controlled metering of high pressure fluid received from the pressure port through the variable fluid pressure orifices to either pressure surface of the ram for controlled actuation thereof and, in the even of a loss of such high pressure fluid, to effect regulated metering of bypass flow through the variable return flow orifices and across the ram for active damping control thereof, and a sensor for monitoring actuator load and providing actuator load feedback information to the electro-mechanical mechanism for implementing controlled modulation of the valve member in response to such actuator load feedback information during both normal operation and when there is a loss of such high fluid pressure to provide dynamic load feedback damping and overpressure relief functions.

An embodiment of the invention will now be described, by way of an example, with reference to the accompanying drawing, in which the single figure is a schematic illustration of a redundant servo system embodying a preferred form of servo actuator control/damping mechanism according to the invention.

Referring now in detail to the drawing, a dual hydraulic servo system is designated generally by reference numeral 10 and includes two similar hydraulic servo actuators 12 and 14. The actuators 12 and 14 are connected to a common output device such as a dual tandem cylinder actuator or ram 16 which in turn may be connected to a control member such as a flight control element of an aircraft. It will be seen below that the two servo actuators 12, 14 normally are operated simultaneously to effect position control of the actuator or ram 16 and hence the flight control element. However, each servo actuator 12, 14 preferably is capable of properly effecting such position control independently of the other so that the control is maintained even when one of the servo actuators 12, 14 fails or is shut down. Accordingly, the two servo actuators 12, 14 in the overall system provide a redundancy feature that increases safe operation of the aircraft.

The servo actuators 12 and 14 are similar and for ease in description, like reference numerals will be used to identify corresponding like elements of the two servo actuators.

Each servo actuator 12, 14 has an inlet port 20 for connection with a source of high pressure hydraulic fluid and a return port 22 for connection with a hydraulic reservoir. Preferably, the respective inlet and return ports 20, 22 of the servo actuators 12, 14 are connected to separate and independent hydraulic systems in the aircraft, so that in the event one of the hydraulic systems fails or shuts down, the servo actuator coupled to the other still functioning hydraulic system may be operated to effect the position control function. Hereinafter, the hydraulic systems associated with the servo actuators 12 and 14 will respectively be referred to as the forward and aft hydraulic systems.

In each of the servo actuators 12 and 14, an inlet passage 24 connects the inlet port 20 to a common main control servo valve designated generally by reference numeral 26. Each inlet passage 24 may be provided with a suitable filter 27 and a check valve 28 which blocks reverse flow through the inlet passage 24 from the servo valve to the inlet port. Each servo actuator 12, 14 also is provided with a return passage 30 which connects the return port 22 to the servo valve 26 via a damping mode accumulator or compensator 32 which serves to maintain pressure in the servo actuator sufficient to prevent cavitation across damping restrictions during damping mode operation as described hereinafter.

The main control servo valve 26 includes a plunger or spool 34 longitudinally shiftable in a cylindrical bore 36 which may be formed by a sleeve (not shown) in an overall system housing. The plunger 34 has two fluidically isolated valving sections indicated generally at 38 and 40, which valving sections 38, 40 are associated respectively with the actuators 12 and 14 and the passages 24 and 30 thereof. The plunger 34 may be selectively shifted from its illustrated neutral or centered position for selective connection of the passages 24 and 30 of each servo actuator 12, 14 to passages 42 and 44 in the same servo actuator.

The passages 42 and 44 of both servo actuators 12 and 14 are connected to the ram 16 which includes a pair of cylinders 46 having respective pistons 48 connected to ram output rod 50 for common movement therewith. More specifically, the passages 42 and 44 of each servo actuator 12, 14 are connected to a corresponding one of the cylinders 46 of opposite sides of the piston 48. The passages 42 and 44 also are connected by respective branch passages 52 and 54 to a common passage 56 which in turn is connected to the corresponding return passage 30. As shown, the branch passages 52 and 54 are respectively provided with anti cavitation check valves 58 and 60 which block fluid flow from the passages 42 and 44 to the common passage 56 but permit free flow from common passage 56 to passages 42 and 44.

With particular reference to the main control serve valve 26, each valving section 38, 40 of the plunger 34 has a pair of longitudinally (axially) spaced apart lands 60 and 62 which are locatable, as when the plunger 34 is in its neutral position, to block flow through respective metering ports 64 and 66 that respectively connect the passages 42 and 44 to the interior of the plunger bore 36. The lands 60 and 62 define therebetween a supply groove 68 which is in communication with the inlet passage 24 and outwardly thereof respective return grooves 70 and 72 which are inter-connected by passage 74 and in common communication with return passage 30. Accordingly, movement of the plunger 34 to either side of its neutral position will connect the inlet passage 24 to one of the passages 42 and 44 and the other of such passages to return passage 30 through respective metering orifices defined by the position of the lands 60 and 62 relative to respective port 64 and 66. Moreover, such metering orifices may be varied in size by selective positioning of the valve plunger 34 in the manner hereinafter described for controlled metering of flow to and from the passages 42 and 44.

From the foregoing, it will be apparent that selective movement of the plunger 34 simultaneously controls both valving sections 38 and 40 which selectively connect one side of each piston 48 to a high pressure hydraulic fluid source and the other side to fluid return for controlled metering of flow to and from the ram 16 which in turn effects controlled movement of the output rod 50 either to the right or left. In the event that one of the servo actuators 12, 14 fails or is shut down, the other servo actuator will maintain control responsive to selective movement of the plunger 34.

Controlled selective movement of the valve plunger 34 is desirably effected by an electric force motor 78 which may be located closely adjacent one end of the plunger. The force motor 78 may be of linear or rotary type and operative connection of the force motor 78 to the valve plunger 34 may be obtained by a link member 80.

The force motor 78 is responsive to command signals received from an electronic control or command system indicated at 82 which may be located, for example, in the aircraft cockpit, whereby the force motor 78 serves as a control input to the valve plunger 34. Also, the force motor 78 preferably has redundant multiple parallel coils so that if one coil or its associated electronics should fail, its counterpart channel or channels will maintain control. Moreover, suitable failure monitoring circuitry is preferably provided to detect when and which channel has failed, and to uncouple or render passive the failed channel.

Feed-back information to the command system 82 is obtained by position transducers or sensors 84 and 86 which are desirably operatively connected to an monitor the positions of the valve plunger 34 and ram output rod 50, respectively. In addition, electronic load sensors 88 or equivalent device are desirably operatively connected to respective cylinders 46 of the ram 16 for monitoring ram load and providing load feedback information to the command system 82 controlling the force motor 78. As shown, each load sensor 88 may be in the form of a differential pressure sensor including a position transducer 92 connected to a longitudinally shiftable spring centered piston 94. Opposite sides or pressure surfaces of the piston 94 are respectively connected by passages 96 and 98 to respective opposite sides or pressure surfaces of the piston 48 in the corresponding cylinder 46 of the ram whereby the position of the piston 94 and corresponding output of the transducer will be indicative of the direction and magnitude of differential pressure forces acting on the piston 48.

Operation

During normal operation of the servo system 10, high pressure fluid from the forward and aft hydraulic systems is supplied via respective inlet passages 24 to the main control servo valve 26. Through selective positioning of the valve plunger 34 in response to command signals received from the command system 82, high pressure fluid from each hydraulic system is controllably metered to either side of the respective piston 48 of the ram 16 to effect controlled movement of the ram output rod 50 with return flow from the opposite side of the piston 48 being simultaneously directed by the valve plunger 34 to return via the passage 30. Further, each electronic load sensor 88 may be used during normal operation to provide dynamic load feed-back information to the command system 82 for implementation of damping and over-pressure relief functions in conventional manner. Further, each electronic load sensor 88 may be used during normal operation to provide surface hinge moment control and hinge moment limiting. This would allow the servo to become a torque or force servo rather than a positional servo.

Should a loss of hydraulic power occur from both the forward and aft hydraulic systems, the command system 82 automatically implements damping mode operation. In the damping mode, the check valves 28 and compensators 32 serve to maintain positive pressure in the system 10 after such loss of hydraulic power by checking fluid loss through the inlet and return ports 20 and 22. The compensators' fluid storage volume can be selected such that damping may be met for a specified minimum period of time.

With positive pressure maintained in the system, active or regulated damping control of the ram 16 is effected by modulating the valve plunger 34 to provide variable orifices which direct and meter bypass flow across each ram piston 48. In this regard, it is noted that the electromechanically driven servo valve 26 is not dependent on hydraulic power for valve plunger positioning whereby the valve plunger 34 will continue to respond to system commands as long as at least one channel of the motor 78 and associated electronics survives and remains operative. Further, the valve plunger 34 is modulated in response to ram load feed-back information from the load sensors 88 which monitor the direction and amplitude of differential pressure across the pistons 48.

In an exemplary situation, over pressure existing or developed on the extend (left) side of each ram piston 48 may be bypassed across the ram 16 to the retract (right) side of each piston 48 by moving the valve plunger 34 to the right of its neutral position to provide a metering orifice connecting passage 42 to return passage 30. This establishes correspondingly metered bypass flow across each piston 48, such flow passing through passage 42 and the provided orifice to return passage 30 which directs the flow to the retract side of the piston via bypass passage 56 and branch passage 54. Conversely, moving the valve plunger 34 to the left of its neutral position will establish and meter bypass flow in the opposite direction across each piston 48. Moreover, the provided orifices may be controllably varied in size to provide desired damped bypass flow by selective positioning of the valve plunger in response to command signals dictated by sensed ram conditions, i.e., ram position monitored by position sensors 86 and ram load monitored by load sensors 88.

Such active or regulated damping control in response to ram load feed-back further may have associated therewith an overload relief function in the damping mode. When excessive load on the ram 16 is sensed by the load sensors 88, an appropriate command signal may be provided to position the valve plunger 34 at a location providing a desired orifice size sufficient to effect rapid relief of such overload condition in order to prevent damage to the actuator and the controlled element connected thereto.

From the foregoing, it can be seen that bypass flow across the ram 16 may be controlled by utilizing the existing flow metering pattern of the main control servo valve 26 and modulating the valve plunger 34 thereof to provide variable orifices for active damping and overload relief control. It also is noted that such active control is even more desirable in redundant systems as shown. If the ram 16 continues to be operated by high pressure fluid supplied to only one of the servo systems, the other servo system operates to effect by pass of the inactive portion of the ram 16. Therefore, the need in such instance for a separate bypass valve is eliminated by such implementation because the main control valve 26 is operated to permit fluid transfer across the respective piston 48 as in normal operation.

Claims

1. A servo mechanism (10) for use in a fluid servo system for controlling a fluid powered ram actuator (16) having opposed pressure surfaces, comprising a servo valve (26) including a valve member (34) selectively positionable therein to provide variable fluid pressure and return flow orifices (62, 64) for metering fluid flow from a pressure port (20) and pressure passage (24) to either pressure surface of the ram (16) and return flow from the other pressure surface of the ram (16) to a return passage (30) and return port (22) through actuator passages (42, 44) between the servo valve (26) and ram (16), a compensator (32) between the return passage (30) and return port (20), and additional passages (52, 54) between the return passage (30) and actuator passages (42, 44) containing check valves (58, 60), characterized by an electro-mechanical mechanism (78) operative independently of fluid pressure to controllably position the valve member (34) to effect controlled metering of high pressure fluid received from the pressure port (20) through the variable fluid pressure orifices (64, 66) to either pressure surface of the ram (16) for controlled actuation thereof and, in the event of a loss of such high pressure fluid, to effect regulated metering of bypass flow through the variable return flow orifices (64, 66) and across the ram (16) for active damping control thereof, and a sensor (88) for monitoring actuator load and providing actuator load feedback information to the electro-mechanical mechanism (78) for implementing controlled modulation of the valve member (34) in response to such actuator load feedback information during both normal operation and when there is a loss of such high fluid pressure to provide dynamic load feedback damping and over-pressure relief functions.

2. A servo mechanism as claimed in claim 1, further characterized in that said electro-mechanical mechanism (78) includes an electric force motor (78) directly mechanically connected to said valve member (34).

3. A servo mechanism as claimed in claim 2, further characterized by an electronic command system (82) for receiving such feedback information and controlling said force motor (78).

4. A servo mechanism as claimed in any preceding claim, further characterized by a position sensor (84) connected to said valve member (34) for providing valve position feedback information to said electro-mechanical mechanism (78) during both normal operation and when there is a loss of such high fluid pressure.

5. A servo mechanism as claimed in any preceding claim, further characterized by a position sensor (86) connectable to said ram (16) for providing ram position feedback information to said electro-mechanical mechanism (78) during both normal operation and when there is a loss of such high fluid pressure.

6. A servo mechanism as claimed in any preceding claim, further characterized in that said ram actuator (16) includes a pair of cylinders (46) and respective pistons (48) interconnected for common movement, a separate source of high pressure fluid for each of said cylinders (46), and said valve member (34) includes a pair of valving sections (38, 40) respectively associated with said pistons (48) for redundant operation, each said valving section (38, 40) including respective variable fluid pressure and return flow orifices (64, 66) for providing metered fluid flow to and from opposite sides of the respective pistons (48) in response to such selective positioning of said valve member (34) under normal operating conditions, and upon loss of one source of high pressure fluid for one of said cylinders (46), said electro-mechanical mechanism (78) being operable independently of the other source of high pressure fluid to selectively position said valve member (34) for controlled actuation of said ram (16) utilizing such other source of high pressure fluid acting on one of said pistons (48), and for controlled metering of bypass fluid across the other piston (48) associated with the lost source of high pressure fluid through said variable return flow orifices (64, 66) in the respective valving section (38, 40) for active controlled damping of said other piston (48).

Patentansprüche

1. Servomechanismus (10) für die Benutzung in einem hydraulischen Servosystem für die Steuerung eines hydraulisch betriebenen Rammenstellgliedes (16) mit gegenüberliegenden Druckoberflächen, mit einem Servoventil (26) mit einem Ventilkörper (34), der in diesem wahlweise positionierbar ist, um einen variablen Fluiddruck vorzusehen, und Rückflußmündungen (62, 64) zum Messen des Fluidstromes von einer Drucköffnung (20) und Druckdurchgang (24) zu jeder Druckoberfläche der Ramme (16) und Rückflusses von der anderen Druckoberfläche der Ramme (16) zu einem Rücklaufdurchgang (30) und Rücklauföffnung (22) durch Stellglieddurchgänge (42, 44) zwischen dem Servoventil (26) und der Ramme (16), einer Kompensiereinrichtung (32) zwischen dem Rücklaufdurchgang (30) und der Rücklauföffnung (20) und zusätzlichen Durchgängen (52, 54) zwischem dem Rücklaufdurchgang (30) und Stellglieddurchgängen (42, 44), welche Absperrventile (58, 60) enthalten, gekennzeichnet durch einen elektromechanischen Mechanismus (78), der betrieblich unabhängig vom Fluiddruck ist, um steuerbar den Ventilkörper (34) zu positionieren und ein gesteuertes Messen von Hochdruckfluid durchzuführen, welches von der Drucköffnung (20) durch die variablen Fluiddruckmündungen (64, 66) zur jeweiligen Druckoberfläche der Ramme (16) für ihre gesteuerte Betätigung aufgenommen ist, und um im Falle eines Abfalles eines solchen Hochdruckfluids das regulierte Messen des Bypass-Flusses durch die variablen Rückflußmündungen (64, 66) und quer über die Ramme (16) für ihre aktive Dämpfungssteuerung durchzuführen, und einen Sensor (88) zum Überwachen der Stellgliedlast und Schaffen einer Stellgliedlaströckkopplungsinformation zu dem elektromechanischen Mechanismus (78) für das Durchführen einer gesteuerten Modulation des Ventilkörpers (34) unter Ansprechen auf eine solche Stellgliedbelastungsrückkopplungsinformation sowohl während des normalen Betriebes als auch dann, wenn es einen Abfall eines solchem Hochfluiddruckes gibt, um eine dynamische Lastrückkopplungsdämpfung und Überdruckentlastungsfunktion vorzusehen.

2. Servomechanismus nach Anspruch 1, ferner dadurch gekennzeichnet, daß der elektromechanische Mechanismus (78) einen elektrischen Kraftmotor (78) aufweist, der mechanisch direkt mit dem Ventilkörper (34) verbunden ist.

3. Servomechanismus nach Anspruch 2, ferner gekennzeichnet durch ein elektronisches Befehlsystem (82) zur Aufnahme dieser Rückkopplungsinformation und Steuerung des Kraftmotors (78).

4. Servomechanismus nach einem der vorhergehenden Ansprüche, ferner gekennzeichnet durch einen Positionssensor (84), der mit dem Ventikörper (34) verbunden ist für die Schaffung einer Ventilpositionsrückkopplungsinformation zu dem elektromechanischen Mechanismus (78) sowohl während des normalen Betriebes als auch dann, wenn es einen Abfall des hydraulischen Hochdruckes gibt.

5. Servomechanismus nach einem der vorhergehenden Ansprüche, ferner gekennzeichnet durch einen Positionssensor (86), der mit der Ramme (16) verbindbar ist zur Schaffung einer Rammenpositionsrückkopplungsinformation zu dem elektromechnischen Mechanismus (78) sowohl während des Normalbetriebes als auch dann, wenn es einen Abfall des hohen Fluiddrukkes gibt.

6. Servomechanismus nach einem der vorhergehenden Ansprüche, ferner dadurch gekennzeichnet, daß das Rammenstellglied (16) ein Paar von Zylindern (46) und entprechende Kolben (48) aufweist, die für eine gemeinsame Bewegung dazwischen verbunden sind, eine getrennte Quelle des Hochdruckfluids aufweist für jeden der Zylinder (46) und der Ventilkörper (34) ein Paar von Ventilabschnitten (38, 40) aufweist, die jeweils dem Kolben (48) zugeordnet sind für redundanten Betrieb, wobei jeder Ventilabschnitt

(38, 40) jeweils variable Fluiddruck- und Rückflußmündungen (64, 66) aufweist zur Schaffung eines gemessenen Fluidstromes zu und von gegenüberliegenden Seiten der jeweiligen Kolben (48) in Antwort auf die wahlweise Positionierung des Ventilkörpers (34) unter normalen Betriebsbedingungen, und wobei nach Abfall eine Quelle von Hochdruckfluide für einen der Zylinder (46) der elektromechanische Mechanismus (78) unabhängig von der anderen Quelle des Hochdruckfluids betreibbar ist, um den Ventilkörper (34) wahlweise zu positionieren für eine gesteuerte Betätigung der Ramme (16), wobei die andere Quelle des Hochdruckfluids, welche auf einen der Kolben (48) wirkt, verwendet wird, und für das gesteuerte Messen des Bypass-Fluids quer über den anderen Kolben (48), welcher der abgefallenen Quelle des Hochdruckfluids über die variablen Rückflußmündungen (64, 66) in dem jeweiligen Ventilabschnitt (38, 40) für das aktive gesteuerte Dämpfen des anderen Kolbens (48) zugeordnet ist.

Revendications

1. Servo-mécanisme (10) pour utilisation dans un servo-système à fluide destiné à commander un actionneur (16) mû par fluide ayant des surfaces de pression opposées, comprenant une servo valve (26) comportant un élément de valve (34) déplaçable sélectivement à l'intérieur pour fournir une pression variable de fluide et des orifices d'écoulement de retour (62, 64) pour réguler l'écoulement de fluide d'un orifice de pression (20) et d'un conduit de pression (24) vers chaque surface de pression d'une pompe (16) et l'écoulement de retour de l'autre surface de pression de la pompe (16) à un conduit de retour (30) et à un orifice de retour (22) à travers des conduits d'actionneur (42, 44) entre la servo-valve (26) et la pompe (16), un compensateur (32) entre le conduit de retour (30) et l'orifice de retour (20), et des conduits additionnels (52, 54) entre le conduit de retour (30) et les conduits d'actionneurs (42, 44) contenant des valves de contrôle (58, 60), caractérisé par un mécanisme électro-mécanique (78) opérationnel indépendamment de la pression de fluide pour localiser de façon commandable l'élément de valve (34) afin d'effectuer une régulation commandée du fluide à haute pression reçue de l'orifice de pression (20) par les orifices variables (64, 66) de pression de fluide vers chaque surface de pression de la pompe (16) en vue de son fonctionnement commandé et, dans le cas d'une perte de ce fluide à haute pression, d'assurer un contrôle régulé de l'écoulement en dérivation par les orifices variables d'écoulement en retour (64, 66) et dans la pompe (16) en vue de sa commande active d'amortissement, et par un capteur (88) pour réguler la charge d'actionneur et fournir une information en retour de charge de l'actionneur au mécanisme (78) afin d'exécuter la modulation commandée de l'élément de valve (34) en réponse à ladite information pendant à la fois le fonctionnement normal et lorsqu'il y a une perte de cette haute pression de fluide de manière à permettre l'amortissement dynamique en retour de la charge et des fonctions de relâchement de la surpression.

2. Servo-mécanisme selon la revendication 1, caractérisé en ce que ledit mécanisme électro-mécanique (78) comprend un moteur électrique (78) relié mécaniquement directement audit élément de valve (34).

3. Servo-mécanisme selon la revendication 2, caractérisé par un système de commande électronique (82) pour la réception d'une telle information de retour et la commande du moteur (78).

4. Servo-mécanisme selon l'une des revendications précédentes, caractérisé par un capteur de position (84) relié audit élément de valve (34) pour la délivrance d'une information en retour de la position de la valve audit mécanisme électro-mécanique (78) pendant à la fois le fonctionnement normal et lorsqu'il existe une perte d'une telle haute pression de fluide.

5. Servo-mécanisme selon l'une des revendications précédentes, caractérisé par un capteur de position (86) qui peut être relié à ladite pompe (16) pour la fourniture d'une information en retour de la position de pompe audit mécanisme électro-mécanique (78) pendant à la fois le fonctionnement normal et lorsqu'il y a une perte d'une telle haute pression de fluide.

6. Servo-mécanisme selon l'une des revendications précédentes, caractérisé en ce que ledit actionneur de pompe (16) comprend une paire de cylindres (46) et des pistons respectifs (48) reliés entre eux par un mouvement commun, une source séparée de fluide à haute pression pour chacun desdits cylindres (46), et en ce que ledit élément de valve (34) comporte une paire de sections de valve (38, 40) associées respectivement auxdits pistons (48) pour un fonctionnement redondant, chaque section de valve (38, 40) comprenant une pression de fluide variable et des orifices d'écoulement en retour permettant un écoulement de fluide régulé vers et des faces opposées des pistons respectifs (48) en réponse au positionnement sélectif de l'élément de valve (34) sous les conditions de fonctionnement normal, et en cas de perte d'une source de fluide à haute pression sur l'un desdits cylindres (46), ledit mécanisme électro-mécanique (78) étant opérationnel indépendamment de l'autre source de fluide à haute pression afin de positionner sélectivement ledit élément de valve (34) pour une exécution commandée de ladite pompe (16), laquelle utilise une telle autre source de fluide à haute pression sur l'un desdits pistons (48), et pour une régulation commandée du fluide de dérivation sur l'autre piston (48) associé à la source perdue de fluide à haute pression à travers les orifices variables d'écoulement en retour (64, 66) dans la section de valve respective (38, 40), ceci afin d'assurer un amortissement actif commandé dudit autre piston (48).