

PATENT SPECIFICATION

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(54) HYDROMECHANICAL TRANSMISSION

(71) We, SUNDSTRAND CORPORATION, a corporation organised and existing under the laws of the State of Delaware, United States of America, of 4751 Harrison Avenue, Rockford, Illinois 61101, United States of America, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The invention relates to hydromechanical transmissions for providing a constant speed output from a variable speed input. Transmissions of this type are useful in aircraft, the aircraft engine driving the transmission at a variable input speed. The input power is transmitted both hydraulically and mechanically through the transmission, the outputs of the hydraulic and mechanical sections of the transmission being summed to drive an output shaft at a constant speed. A generator is connected to the output shaft and thus driven at a constant speed so as to produce a constant frequency source of electrical power for the various accessories of the aircraft.

The invention provides a hydromechanical transmission comprising an input shaft, an axial piston hydraulic pump assembly associated with the input shaft to be driven thereby, a movable swashplate operatively associated with the pump assembly for selectively varying the displacement thereof, an axial piston hydraulic motor assembly coaxial with the pump assembly, a floating valve plate disposed between and engaging the pump and motor assemblies, a sun gear carried by one of the pump and motor assemblies, a ring gear carried by the other of the pump and motor assemblies, a rotatable output carrier coaxial with the pump assembly and at least one planet gear carried by the rotatable output carrier and meshed with both the sun and ring gears.

The invention is illustrated by the drawings of which:

Figure 1 is a fragmentary, cross-sectional view of a hydromechanical transmission according to the invention;

Figure 2 is a diagrammatic illustration of parts of an epicyclic train of gears employed in the hydromechanical transmission of Figure 1;

Figure 3 is a graph in which efficiency is plotted against input speed for the hydromechanical transmission of Figure 1 and a prior art hydromechanical transmission;

Figure 4 is a speed chart for the hydromechanical transmission of Figure 1; and

Figure 5 is a speed chart for a prior art hydromechanical transmission.

With reference to Figure 1, a hydromechanical transmission 9, particularly suited for use in providing a constant speed drive for a low power generator such as one having a power rating generally in the range of from 20 to 60 kilowatts, includes an input shaft 10 adapted to be connected at one end to a prime mover (not shown) such as the engine of an aircraft. The other end 12 of the input shaft 10 is splined and received in a complimentary splined bore 14 in a cylinder block 16 of an axial piston hydraulic pump assembly 18. The pump assembly 18 supplies a driving fluid to a smaller displacement motor assembly 50 which, in turn, through a suitable epicyclic gear train drives a gear 108 to transmit rotational power to a generator at a constant speed. Preferably, the pump assembly 18 and the motor assembly 50 are constructed in the so-called "half-head" manner such as that disclosed in United States Patent Specification No. 3,396,670.

The cylinder block 16 is provided with a plurality of chambers or bores 20, extending parallel to the input shaft 10 and in a concentric annular array around the shaft. Each of the bores 20 receives a piston 22 for reciprocation therein and a rod 24 interconnects each piston 22 and an annular plate 26 which is in slidable engagement with a conventional, pivotably mounted swashplate 28. In addition, the cylinder block 16 includes a plurality of circumferentially spaced bores 30, each receiving a spring 32 which bears against a hemispherical member 34 which, in turn, is re-

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The pump assembly 18 is received in a hemispherical recess 36 in a retainer 37 secured to the plate 26. The springs 32 bias the cylinder block 16 and the hemispherical member 34 away from each other. The relationship of the parts is such that the swashplate 28 may be pivoted about an axis extending through a point 38 to vary the displacement of the pump assembly 18 in a conventional manner. The rods 24 are maintained generally perpendicular to the plate 26 by guidance in grooves 39 in the retainer but have some freedom of movement in the grooves as taught in the aforementioned United States Patent Specification No. 3,396,670.

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The transmission 9 further includes a housing 40 and the cylinder block 16 is journaled for rotation in the housing 40 by bearings 42. Mounted within the housing coaxially with the pump assembly 18 is the motor 50 which includes a rotatable cylinder block 52 provided with an annular array of axial chambers 54 each receiving a piston 56. Each of the pistons 56 is connected by a rod 58 to an annular plate 60, which slidably engages a fixed swashplate 62 secured to the housing 40 by any suitable means. The cylinder block 52 also includes an annular array of axial bores 64 for receipt of springs 66 which bear against an axially fixed hemispherical element 67 and serve to bias the rotatable cylinder block 52 toward the right as seen in Figure 1. The hemispherical element 67 seats in a retainer 68 corresponding to the retainer 37 in the pump assembly 18.

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Between the pump assembly 18 and the motor assembly 50, a valve plate 70 is provided with a pair of arcuate ports 72 and 73 arranged in conventional fashion to establish fluid communication between the pump and the motor. Because of the difference between the fluid displacements of the pump assembly 18 and the motor assembly 50, the axial forces generated by operation of the pump assembly 18 are greater than those generated by operation of the motor assembly 50. In order to achieve a balance of these axial forces on opposite sides of the port plate 70, the axial forces generated by the pump in excess of the axial forces generated by the motor are taken up by utilizing bearings 42 which are of the axial thrust, pre-loaded type. Herein, these bearings 42 act between the periphery of the cylinder block 16 and the housing 40 and provide active support in both radial and axial directions. The bearings, however, are only one way of supporting the unbalanced load caused by operation of the larger pump assembly 18 and smaller motor 50.

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The valve plate 70 located between the pump and motor cylinder blocks 16 and 52 is mounted so as to "float" in both radial and axial directions. In the present instance,

the plate 70 includes an elongated stem 71 which extends through a bore 74 in the motor cylinder block 52 and is received within a bore 76 in the housing 40. The interior of the stem 71 is provided with a conduit 78 which is disposed in fluid communication with a conduit 80 in the housing 40 for receipt of pressurized oil from supply (not shown) to replace fluid lost during operation of the transmission. Preferably, the stem 71 is sealed against the bore 76 by an O-ring 82 and is keyed in a desired circumferential position by a key 81. Leakage fluid within the housing is collected and returned to the supply.

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Engaging the opposite faces of the floating valve plate 70 are bearing lips 86 which are integrally formed with the motor cylinder block 52 and the pump cylinder block 16 and engage the plate 70 around the opposite ends of the ports 72 and 73. It will be recognized that the only securement of the valve plate 70 is that provided by the relatively loose fit of the stem 71 within the bore 76. As a consequence, clearances between the stem 71, the motor cylinder block 52 and the housing 40 enable the valve plate 70 to move or "float" in both the axial and radial directions. This movement, while slight, advantageously helps avoid metal-to-metal contact of the valve plate 70 with the cylinder blocks 16 and 52 such as might otherwise occur when an unbalance of forces in the transmission 9 tends to cock the cylinder blocks relative to their normal rotational axis during operation of the transmission. By floating, the plate 70 is free to move with the cylinder blocks 16 and 52 in both axial and transverse directions so that the hydrostatic films on either side of the valve plate 70 are kept from being penetrated by the rotating cylinder blocks 16 and 52 thereby avoiding scoring or metal-to-metal wear between the valve plate 70 and the cylinder blocks 16 and 52. Typically, any axial and transverse movement of the cylinder blocks 16 and 52 caused by unbalanced forces is due to tolerance clearances between parts of the unit as well as hydrostatically generated clearances.

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Providing the mechanical power transmitting section of the transmission 9 is an epicyclic train gear which, herein, includes an internal ring gear 46 carried by the peripheral flange of a bell-shaped housing 44 secured to the pump cylinder block 16. The ring gear 46 meshes with one or more, generally at least three, planet gears 92 journaled on circumferentially spaced stub shafts 96 extending from one edge of a tubular carrier 98. The carrier 98 includes an internal bore 100 in which the motor cylinder block 52 is journaled by means of roller bearings 102, providing support

against radially directed movement of the motor cylinder block 52. Within the bore 100, the planet gears 92 mesh with a sun gear 90 which is mounted on the periphery of the motor cylinder block 52. Provided on the periphery of the carrier 98 is a gear 106 which serves as an output from the transmission 9 and meshes with the gear 108 to provide a constant speed drive for the generator.

Because the rotation of the planet gears 92 reverses during operation of the transmission 9, special antifriction bearings 94 support the planet gears 92 rotatably on the carrier 98 so as to avoid losing the hydrodynamic oil film in the connection between the planet gears 92 and the carrier 98 when the gears reverse turning direction. Supporting the carrier 98 for rotation within the housing 40 are suitable roller bearings 104.

With the transmission 9 employed in a constant speed drive, a speed sensor (not shown) is utilized in a conventional fashion to alter the angle of the swashplate 28 in response to speed variation to thereby adjust the displacement of the pump assembly 18 so as to keep the output speed constant. As is well known, the speed of the motor assembly 50 will be varied by the volume of the fluid provided to it at any given period of time. As consequence, its rate of rotation will vary depending upon the volumetric flow rate to it, with the consequence that the interaction between the sun and ring gears 90 and 46 respectively via the planet gears 92 will be varied to thereby maintain desired constant output speed.

In operation of the transmission 9, the two cylinder blocks 16 and 52 always are rotated in the same direction, although at different relative speeds, so that, within the normal speed range of the input shaft 10, the torque transmitted hydraulically always is added to the torque transmitted mechanically. As illustrated diagrammatically in Figure 2, forces F_r and F_s are applied by the ring and sun gears 46 and 90, respectively, to the carrier 98 so that the forces are additive and rotate the carrier in the same direction that the ring and the sun gears 46 and 90 respectively are rotated. At an idle speed for the transmission (e.g. 4500 rpm), the rotation produced hydraulically is at a maximum because the swashplate 28 is set for maximum pump displacement. But, as the speed of the input shaft 10 increases toward its maximum speed and the angle of the swashplate 28 is decreased, the hydraulic percentage of output rotation also decreases, approaching toward zero at maximum speed of the input shaft 10 so that the far greater percentage of output rotation is supplied through the epicyclic section of the transmission 9. This

output is produced more efficiently simply because the epicyclic section of the transmission 9 operates more efficiently than the hydraulic section. Accordingly, with the aircraft operating within the normal cruise speed range of the aircraft so the input shaft 10 rotates at a speed below a maximum speed (e.g. 9000 rpm), a more efficient output is achieved. In prior systems, operating at cruise speeds, the percentage of hydraulic rotation is a higher percentage of the output and, therefore, is less efficient.

The graph in Figure 3 shows the relationship between the output efficiencies of the above described hydromechanical transmission and a prior hydromechanical transmission in which the mechanical transmitting sections are mounted separately from the hydraulic transmitting sections. Typically, with such a prior system, at some point generally central of the input speed range, the pump displacement is zero so the motor cylinder block speed is zero and all of the rotational output is supplied through the mechanical section of the transmission. This point is referred to as the "straight through point" of the transmission. While it is desirable to run at the straight through point from the standpoint of efficiency, it is extremely difficult to maintain proper control of the transmission due to system instability which is caused primarily by the difference in the static and kinetic coefficients of friction involved when moving the motor cylinder block from zero speed. For example, when operating the prior transmission at the straight through point, if demand is made for slightly increased or decreased output speed, the control adjustment made in the position of the pump swashplate tends to overcompensate in order to overcome the static friction associated with the motor cylinder block. This tends to increase or decrease the output speed beyond that which is needed, thereby causing a second control demand in the reverse direction. The above described hydromechanical transmission according to the invention avoids the foregoing problem because its straight through point is beyond the normal operating speed range of the input shaft 10.

In comparing the speed chart (Figure 4) for the transmission according to the invention and the speed chart (Figure 5) for the prior art transmission, it is seen that the rotational speeds of the planet gears 92 in the transmission according to the invention are below the speed of the output gear 108 whereas the rotational speeds of the planet gears in the prior art transmission may be more than double the speed of the output gear, e.g. as high as 28,000 rpm. The point at which the speeds of the planet gears 92 drop to zero is, of course, the point at

which, by reason of control of the swash-plate 28, the displacement of the pump assembly 18 equals the displacement of the motor assembly 50 so that the two cylinder blocks 16 and 52 rotate at the same speed. The reason the planet gears 92 rotate within this lower speed range is because the ring gear 46, the sun gear 90 and the carrier 98 are rotated in the same direction with the hydraulic rotation always being added to the mechanical rotation. The lower operating speeds of the planet gears 92 of the transmission according to the invention enable them to last longer and to be made of lower strength, less expensive materials.

In the hydromechanical transmission according to the invention, even though the ring gear 46 and the sun gear 90 are rotated in the same directions, the relative speeds may be different. In fact, initially the sun gear 90 rotates much faster than the ring gear causing the planet gears 92 to rotate on their axis in the opposite direction to the direction of rotation for the sun gears. As the speeds of the sun and ring gears 90 and 46 respectively approach each other from their idle speeds, the planet gears slow down eventually to reverse turning direction as the speed of the ring gear becomes greater than the speed of the sun gear (see Figure 4). Thus, within the normal operating range of the present transmission, the planet gears 92 rotate at fairly low speeds.

From the foregoing, it will be appreciated that the invention brings to the art a novel hydromechanical transmission 9 which, by virtue of mounting the epicyclic sections of the transmission directly on the cylinder blocks 16 and 52 and concentrically with the rotational axes of such blocks, enables the transmission to be made more compactly and with fewer interconnecting parts than prior hydromechanical transmissions. As a result, the present transmission is substantially lighter in weight thereby providing a high power density transmission, that is, one which provides high horsepower with respect to weight or size. Moreover, the transmission operates more efficiently within normal cruise speed ranges by taking advantage of the more efficient output of the epicyclic section of the transmission to provide the greater percentage of output power. Still further, the present construction eliminates midrange control problems by eliminating the straight through point from the normal operating speed ranges of the transmission. Additionally, the transmission according to the invention is of sturdier construction, providing a longer useful service life by avoiding high speed operation of various parts of the epicyclic sections of the transmission and by providing a unique mounting arrangement for the valve plate 70 so that the

plate "floats" so as to avoid metal-to-metal contact between the cylinder blocks 16 and 52 should be an imbalance of forces within the transmission tend to cock the cylinder blocks 16 and 52 with respect to the valve plate 70.

WHAT WE CLAIM IS:—

1. A hydromechanical transmission comprising an input shaft, an axial piston hydraulic pump assembly associated with the input shaft to be driven thereby, a movable swashplate operatively associated with the pump assembly for selectively varying the displacement thereof, an axial piston hydraulic motor assembly coaxial with the pump assembly, a floating valve plate disposed between and engaging the pump and motor assemblies, a sun gear carried by one of the pump and motor assemblies, a ring gear carried by the other of the pump and motor assemblies, a rotatable output carrier coaxial with the pump assembly and at least one planet gear carried by the rotatable output carrier and meshed with both the sun and ring gears.

2. A hydromechanical transmission according to claim 1 wherein the pump assembly is secured to the input shaft for rotation therewith.

3. A hydromechanical transmission according to claim 1 or claim 2 wherein the sun gear is carried by the motor assembly.

4. A hydromechanical transmission according to any preceding claim wherein the rotatable output carrier includes an interior bore within which the motor assembly is journaled for rotation.

5. A hydromechanical transmission according to any preceding claim wherein the sun, planet and ring gears are coplanar and lie in a plane located between the pump and motor assemblies.

6. A hydromechanical transmission according to any preceding claim wherein each of the pump and motor assemblies comprises rotatable block having a series of chambers formed therein and a piston reciprocable in each chamber, the pistons being positioned within the chambers through their reciprocation.

7. A hydromechanical transmission according to any preceding claim wherein the floating valve plate includes a stem extending through one of the pump and motor assemblies, a conduit for connection to a source of hydraulic fluid being provided in the stem.

8. A hydromechanical transmission according to any preceding claim wherein the ring gear is defined by a bell-shaped housing secured to that one of the pump and motor assemblies which carries the ring gear and opening toward that one of the pump and motor assemblies which carries the sun gear.

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which, by reason of control of the swash-plate 28, the displacement of the pump assembly 18 equals the displacement of the motor assembly 50 so that the two cylinder blocks 16 and 52 rotate at the same speed. The reason the planet gears 92 rotate within this lower speed range is because the ring gear 46, the sun gear 90 and the carrier 98 are rotated in the same direction with the hydraulic rotation always being added to the mechanical rotation. The lower operating speeds of the planet gears 92 of the transmission according to the invention enable them to last longer and to be made of lower strength, less expensive materials.

In the hydromechanical transmission according to the invention, even though the ring gear 46 and the sun gear 90 are rotated in the same directions, the relative speeds may be different. In fact, initially the sun gear 90 rotates much faster than the ring gear causing the planet gears 92 to rotate on their axis in the opposite direction to the direction of rotation for the sun gears. As the speeds of the sun and ring gears 90 and 46 respectively approach each other from their idle speeds, the planet gears slow down eventually to reverse turning direction as the speed of the ring gear becomes greater than the speed of the sun gear (see Figure 4). Thus, within the normal operating range of the present transmission, the planet gears 92 rotate at fairly low speeds.

From the foregoing, it will be appreciated that the invention brings to the art a novel hydromechanical transmission 9 which, by virtue of mounting the epicyclic sections of the transmission directly on the cylinder blocks 16 and 52 and concentrically with the rotational axes of such blocks, enables the transmission to be made more compactly and with fewer interconnecting parts than prior hydromechanical transmissions. As a result, the present transmission is substantially lighter in weight thereby providing a high power density transmission, that is, one which provides high horsepower with respect to weight or size. Moreover, the transmission operates more efficiently within normal cruise speed ranges by taking advantage of the more efficient output of the epicyclic section of the transmission to provide the greater percentage of output power. Still further, the present construction eliminates midrange control problems by eliminating the straight through point from the normal operating speed ranges of the transmission. Additionally, the transmission according to the invention is of sturdier construction, providing a longer useful service life by avoiding high speed operation of various parts of the epicyclic sections of the transmission and by providing a unique mounting arrangement for the valve plate 70 so that the

plate "floats" so as to avoid metal-to-metal contact between the cylinder blocks 16 and 52 should be an imbalance of forces within the transmission tend to cock the cylinder blocks 16 and 52 with respect to the valve plate 70.

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2. A hydromechanical transmission according to claim 1 wherein the pump assembly is secured to the input shaft for rotation therewith.

3. A hydromechanical transmission according to claim 1 or claim 2 wherein the sun gear is carried by the motor assembly.

4. A hydromechanical transmission according to any preceding claim wherein the rotatable output carrier includes an interior bore within which the motor assembly is journaled for rotation.

5. A hydromechanical transmission according to any preceding claim wherein the sun, planet and ring gears are coplanar and lie in a plane located between the pump and motor assemblies.

6. A hydromechanical transmission according to any preceding claim wherein each of the pump and motor assemblies comprises rotatable block having a series of chambers formed therein and a piston reciprocable in each chamber, the pistons being positioned within the chambers through their reciprocation.

7. A hydromechanical transmission according to any preceding claim wherein the floating valve plate includes a stem extending through one of the pump and motor assemblies, a conduit for connection to a source of hydraulic fluid being provided in the stem.

8. A hydromechanical transmission according to any preceding claim wherein the ring gear is defined by a bell-shaped housing secured to that one of the pump and motor assemblies which carries the ring gear and opening toward that one of the pump and motor assemblies which carries the sun gear.



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5 9. A hydromechanical transmission according to any preceding claim wherein the output carrier comprises a tubular structure internally journalling that one of the motor and pump assemblies which carries the sun gear and includes exterior, peripheral drive means for engagement with an element to be driven.

10. A hydromechanical transmission substantially as described herein with reference to Figures 1 and 2 of the drawings. 10

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