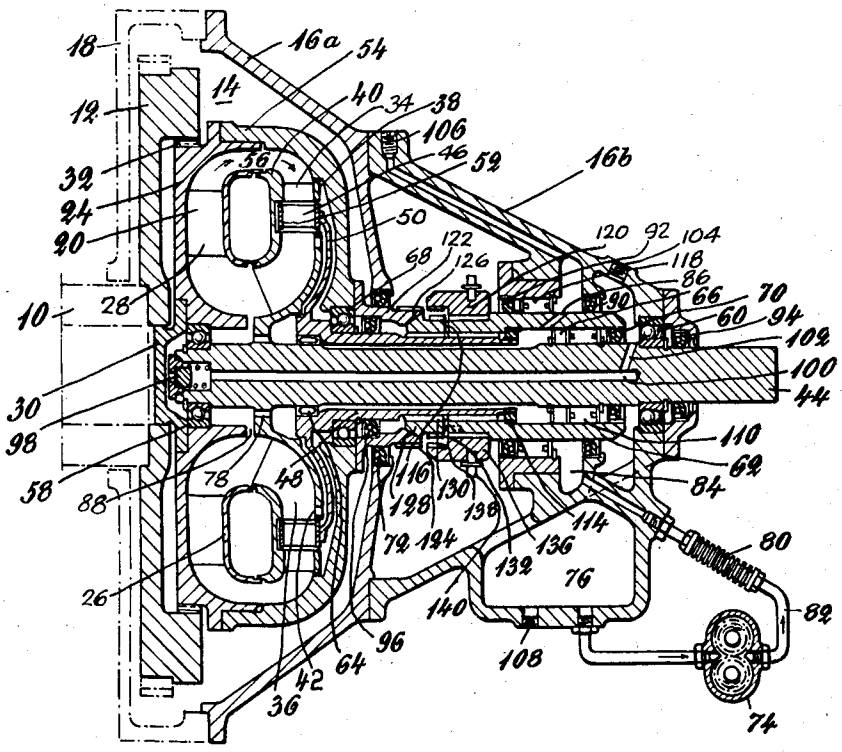
June 2, 1959 K. G. ÅHLÉN 2,888,842

HYDRAULIC TORQUE CONVERTERS

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Fig.1.



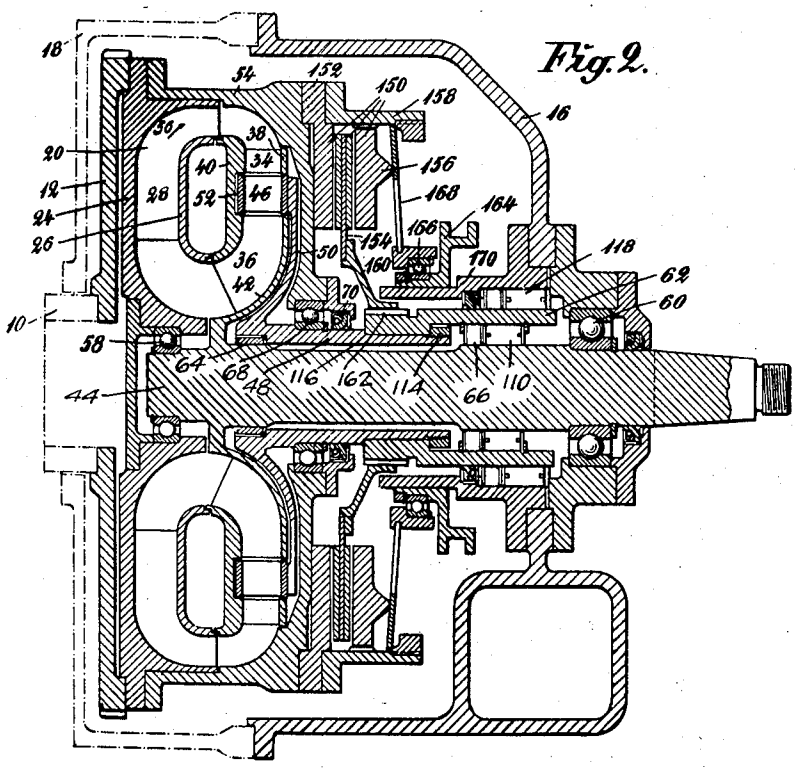
INVENTOR
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HYDRAULIC TORQUE CONVERTERS

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Fig. 2.



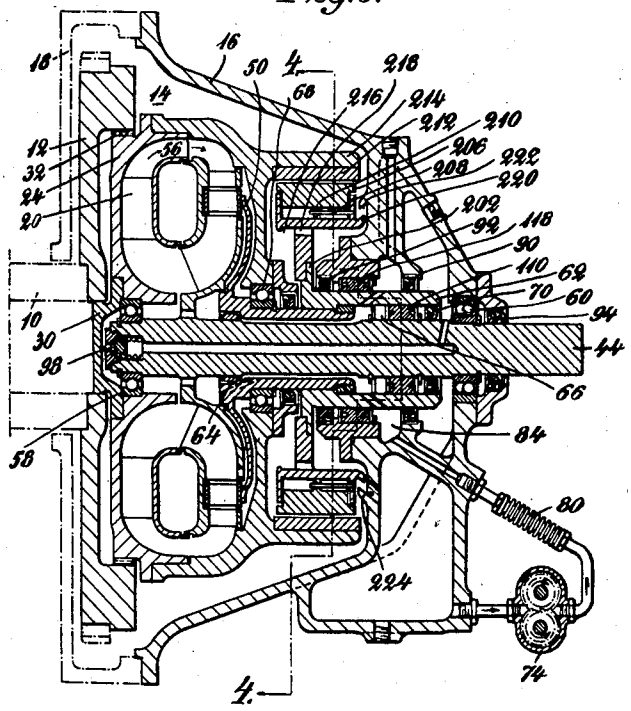
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Fig. 3.



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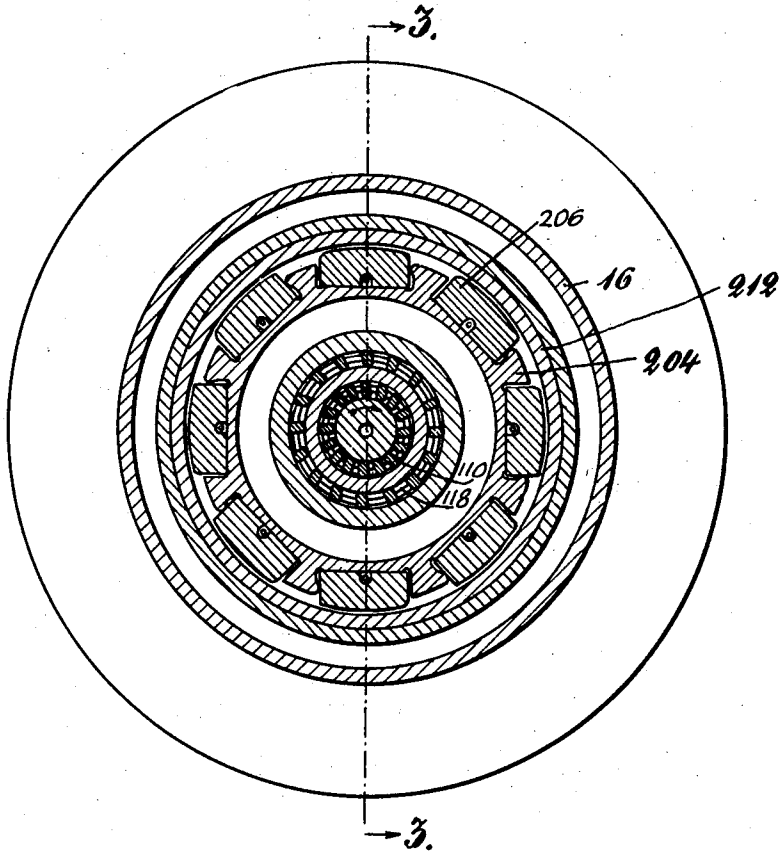
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Fig. 4.



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Fig. 6.

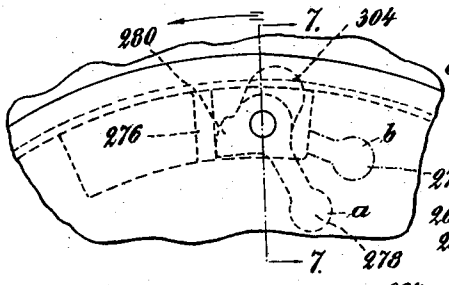
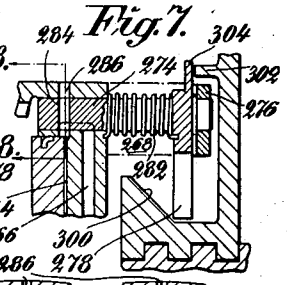


Fig. 7.



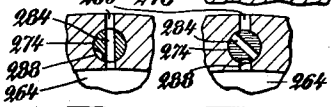


Fig. 5.

Fig. 8. Fig. 9.

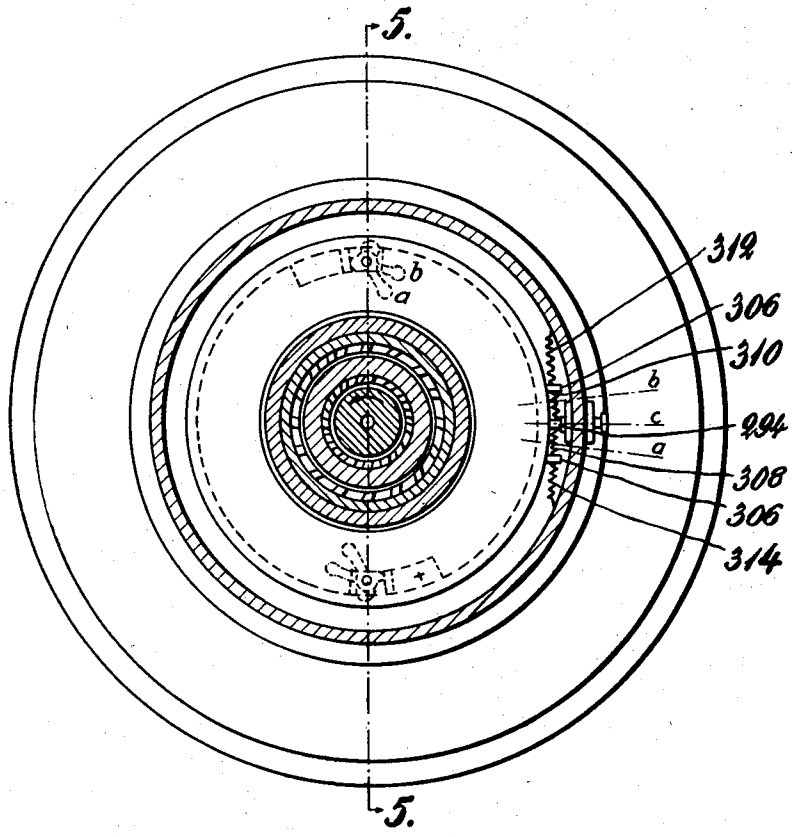
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HYDRAULIC TORQUE CONVERTERS

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Fig. 10.



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Fig. 13.

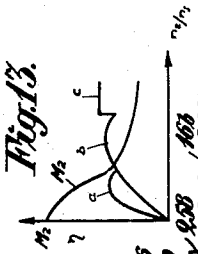
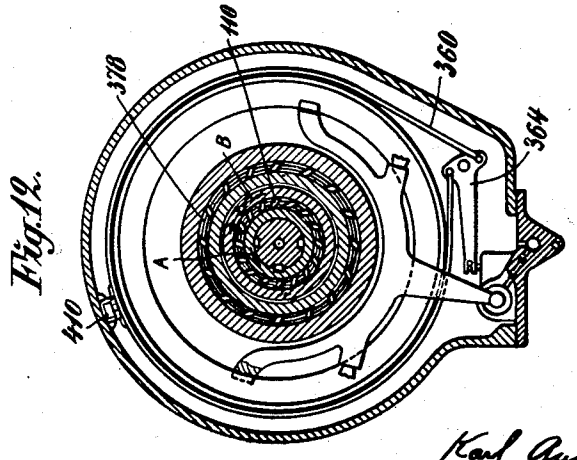


Fig. 11.

Fig. 12.



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Fig. 16.

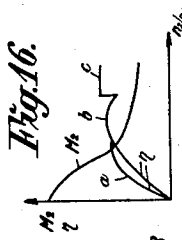
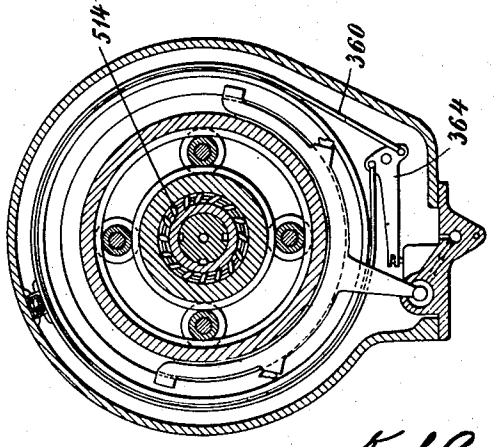


Fig. 14.

Fig. 15.



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United States Patent Office 2,888,842 Patented June 2, 1959

2,888,842

HYDRAULIC TORQUE CONVERTERS

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Original application May 27, 1948, Serial No. 29,446, now Patent No. 2,719,616, dated October 4, 1955. Divided and this application October 4, 1955, Serial No. 538,355

5 Claims. (Cl. 74—730)

The present invention relates to hydraulic power transmissions or so-called hydraulic torque converters consisting essentially of a pump member, a turbine member and a reaction member, the reaction member generally being arranged as a stationary guide blade ring between the rotating blade rings of the turbine member. The torque converter is employed to multiply the engine torque for starting and for acceleration periods as long as the pump or primary speed n_1 can be kept high enough in relation to the turbine or secondary speed n_2. With increasing secondary speed n_2 relative to the primary speed n_1, that is increasing speed ratio n_2/n_1, the secondary torque M_2 drops continuously and at the moment when the ratio n_2/n_1 is equal to the actual efficiency of the torque converter the primary torque M_1 and the secondary torque M_2 are equal and there exists no longer any torque multiplication. With further increasing value of n_2/n_1 the secondary torque M_2 will be less than M_1 which means that the converter above this mentioned specific point has lost its function as a torque multiplier and has only a negative effect on the power transmission. For this reason the hydraulic torque converters have been provided with some kind of device for establishing direct drive between the input and output shaft as soon as the speed ratio n_2/n_1 has reached the above-mentioned characteristic value, approximately equaling the value of the converter efficiency. Certain hydraulic converters have been designed to operate as hydraulic couplings at higher speed ratios, and in other systems the type of design is such that the converter will be completely disconnected and direct drive immediately established between the input and output shafts. Both these systems have been used earlier.

The present invention relates primarily to the type of device in which hydraulic drive is combined with means for providing a direct drive to be used in alternation with the hydraulic drive. More particularly the invention contemplates a transmission device in which a primary casing member, a secondary member and a reaction member are rotatably mounted within a stationary housing member.

A general object of the invention is the provision of novel forms of construction whereby the direct drive connection between the primary and secondary member may be established through a connection including a part of the reaction member as a power transmitting element and in which simple and effective means are provided for controlling the operation of the device either as a hydraulic torque converter or as a direct drive power transmitter either manually or automatically under predetermined conditions of operation.

Other and more detailed objects relating to features of control and other more detailed features will appear as the ensuing description proceeds.

For a device according to these general characteristics it is unimportant whether the reaction member is mounted for rotation in one direction or in both directions. If the reaction member is designed for rotation

in both directions the resulting advantage will be, that, during its rotation in the opposite direction against the primary and secondary members, a steeper increasing efficiency curve—that is higher torque multiplication—is obtained if the torque from the reaction member is transmitted via a reverse gearing to the output shaft. When, however, with the above-mentioned design, upon counter-rotation of the reaction member, the top of the efficiency curve has been reached, the efficiency curve will, as known, fall rapidly. For this reason it is to be preferred to prevent counter-rotation of the reaction member as soon as the single rotating system is more efficient and for this purpose the arrangement according to the invention must be completed with a brake or blocking device, preventing rotation of the reaction member in a direction opposite to that of the primary and secondary members. If this blocking device is made free to engage or disengage, the converter with disengaged blocking device can operate with the reaction member in counter-rotation at the beginning of the starting period and by engaging the locking device change to a system with stationary reaction member. When the suitable speed ratio n_2/n_1 for direct drive is reached, the locking device is disengaged again and instead of it the reaction member is connected to the primary and secondary members of the converter in order to form a connecting and power transmitting member between said members. If a simpler design is desired, the system is provided immediately with a locking device, preventing the rotation of the reaction member in the direction opposite that of the primary and secondary members. The simplest device for preventing such rotation of the reaction member is a free wheel, a self-locking brake or an automatic claw-clutch.

In order to connect the reaction member to the primary member a claw-clutch, possibly synchronized, a friction clutch—for instance of friction disc or centrifugal weight type or the like—can be used, whereas the connecting means between reaction member and secondary member can be carried out more simply and can, for instance, consist of a free wheel or a self-locking brake, preventing the reaction member from rotating faster than the secondary member.

In the following the invention will be described more in detail, referring to the accompanying drawings, which as examples show some preferred arrangements.

Fig. 1 shows a longitudinal section through a hydraulic torque converter with rotating pump casing and synchronized claw-clutch between the reaction member (guide vane shaft) and primary member (pump casing) as well as a free-wheel clutch between the reaction member and secondary member (turbine shaft) and, finally, a free wheel preventing rotation of the guide vane shaft in a direction opposite to that of the pump and turbine shafts.

Fig. 2 shows an arrangement corresponding to that of Fig. 1 but with a manually operated friction disc clutch instead of synchronized claw-clutch between guide and the pump member.

Fig. 3 shows an arrangement corresponding to that of Fig. 1 but with the synchronized claw-clutch replaced by a centrifugal weight clutch.

Fig. 4 is a section through the centrifugal weight clutch along the lines IV—IV in Fig. 3.

Fig. 5 shows an arrangement with pressure fluid actuated friction disc clutch between the guide member shaft and the pump casing.

Figs. 6–9 show details of the control devices to the disc clutch according to Fig. 5, where Figs. 6 and 7 are details showing a valve governed by a centrifugal force actuated weight for regulation of the pressure fluid sup-

ply to and from the friction clutch, Figs. 8 and 9 are cross-sections through the valve rod showing the channels in the same for the two clutch positions, namely Fig. 8, giving direct passage to the outlet from the working chambers and Fig. 9 showing the inlet passage of pressure fluid to the working chamber of the friction disc clutch.

Fig. 10 is a section along line X—X in Fig. 5.

Fig. 11 shows an arrangement corresponding to Fig. 5, where the reaction member (guide member) is arranged for rotating in both directions and provided with a reverse gearing for torque transmission from the reaction member to the output shaft.

Fig. 12 is a section along the line XII—XII in Fig. 11.

Fig. 13 is a diagram for an arrangement according to Fig. 11, showing the efficiency, tractive effort and secondary torque curves within the different converter drives as functions of the speed ratio n_2/n_1.

Fig. 14 shows an arrangement of the same type as in Fig. 11 but with a different type of reverse gearing.

Fig. 15 is a section along the line XV—XV in Fig. 14; and

Fig. 16 shows a corresponding diagram as that of Fig. 13 for an arrangement according to Fig. 14.

In the drawings **10** indicates the crank shaft and **12** the fly-wheel of an internal combustion engine cooperating with the hydraulic converter **14**. The housing **16** of the converter (Figs. 2 and 3) is in certain arrangements composed of two parts **16***a* and **16***b*, rigidly connected to each other and to the engine casing **18**, together with the crank shaft **10** supporting the rotatable parts, bearings and seals of the converter. The primary or pump member **20** of the converter is of the centrifugal type, consisting of the pump discs **24** and **26** and the pump blades **28**, supported and driven by the fly-wheel **12** through the hub part **30** and the toothed rims at **32**. The secondary or the turbine member consists of two blade rings **34** and **36**, the blades of which are connected as well between the annular discs **38** and **40** as between the disc **40** and the disc **42** extending from the turbine shaft **44**. The reaction or guide member consists of a blade ring **46**, inserted between the two turbine rings **34** and **36** and supported by the disc **50** extending from the guide member shaft **48**. The guide blades are fixed between discs **50** and **52**. The blade system of the converter is, consequently, of the two-stage type with rotating outer pump casing according to a design known per se. The torus shaped circuit for the working fluid is outwardly limited by the pump disc **24** and its extension in the shape of a shell **54**, both forming the rotating casing, and furthermore by the discs **38** and **42** and the outer part of the disc **50**. The inner wall of said circuit is formed by the impeller disc **26** and the discs **40** and **52**. The direction of flow of the fluid is indicated by the arrow **56**.

The turbine shaft **44** is journaled on the two bearings **58** and **60**, which are supported by the pump disc **24** and the stationary casing **16** respectively.

The guide member shaft **48** and its extension **62**, as hollow shafts concentrically surrounding the turbine shaft **44**, are centered in relation to said shaft **44** by means of the bearings **64** and **66** in the same way that the third bearing **68** supports and centers the guide member shaft against the shell **54**, forming part of the rotating pump casing.

In order to prevent unnecessary leakage between the turbine shaft **44** and the guide shaft extension **62** a seal **70** is provided, and for the same reason a seal **72** is provided between the guide shaft **48** and the rotating pump casing.

As the fluid of the hydraulic system preferably operates under a certain basic pressure an auxiliary pump **74**—as indicated by the drawing—sucks in a suitable quantity of fluid from the tank **76** and forces it through the cooler **80** in the piping **82** up to a distributing chamber **84** and further through the bores **86** in the guide shaft extension **62**, through the space between the turbine and guide shafts, through the bores **78** in the disc **42** to the hydraulic circuit via the narrow space **88** between the discs **24** and **42**. The seals **90** and **92**, being placed on each side of the distributing chamber **84**, thereby prevent unnecessary leakage flow back to the tank **76**. The tank in its turn is sealed against the rotating parts of the converter at **94** and **96**.

Dangerously high pressure-rises in the hydraulic system are prevented by a spring-loaded valve **98**, normally closed, but returning part of the fluid from the circuit to the tank **76** via the bores **100** and **102** in the turbine shaft at excess pressures.

Filling and vent openings are indicated at **104** and **106** respectively as well as a bottom hole **108** for emptying the tank **76**.

In order to obtain direct drive with a torque converter as described above without using a special intermediate shaft between the crank shaft and the driven shaft connected to the turbine shaft, a free-wheel **110** has been inserted—as shown by the drawings—between the turbine shaft **44** and the guide shaft extension **62**, the latter part being rigidly connected to said guide shaft **48** by means of key and groove at **112** and a nut **114**, pressing the guide shaft extension **62** against a stop ring or step **116** on the shaft **48**. This free-wheel **110** is so designed that it prevents the guide or reaction member from rotating faster than the turbine shaft **44** during the conditions when the reaction and the secondary parts have same direction of rotation. A second free-wheel **118** between the guide shaft extension **62** and the stationary casing **16***b* prevents the reaction member during converter drive from rotating in the opposite direction as compared with the secondary member.

The mechanical connection of the primary or pump member and the reaction or guide member is in the arrangement shown in Fig. 1 effected by means of a synchronized claw-clutch **120** known per se, which consists of the following parts and acts as follows.

A sleeve **122** is secured to the hub part of the pump casting part **54** concentrically in relation to the guide shaft **48**, the outer circumference of the sleeve being provided with a rim of straight teeth or splines **124** and, further, with a conical surface **126**, against which surface a second sleeve **128** is supported on a surface of corresponding conical shape. This sleeve **128** is guided in relation to the guide shaft **48** by splines **130**, which are made with comparatively small clearance in radial direction but with large clearance in peripheral direction, so that the sleeve **128** will turn from one end position to the other in relation to the hollow shaft **48**, when the relative speed between pump and guide shaft **48** changes direction. The sleeve **128** also has an outer rim of splines **132** and is pressed against the conical surface **126** by means of springs, acting between the stop or step **116** and the end surface of said sleeve **128**. A third sleeve **136**, sliding axially on splines along the guide shaft extension **62**, has a second inner rim of splines **138**, fitting in the rim of splines **124** on the sleeve **122**, when the sleeve **136** has been moved to the left in Fig. 1 by means of, for instance, a lever acting upon the two pins **140**. It is also characteristic for the design that the rim of splines **132** of the sleeve **128** is locking the sleeve **136** against axial movement as long as said first sleeve **128** by the friction forces between the conical surfaces at **126** is peripherally moved to that side, determined by driving the converter so that the pump casing has a higher rotational speed than the guide member, but permits an axial movement of said sleeve **136**, when the guide member obtains a higher rotational speed than the pump casing and the sleeve is turning over to the peripherally opposite end position.

The shifting from hydraulic to direct drive may preferably occur at the moment when the guide no longer is

subjected to any reaction torque from the fluid of the converter circuit and, consequently, the primary or pump member torque and the secondary or output torque of the turbine are equal. This condition occurs at the operation stage when the speed ratio n_2/n_1 and the efficiency of the converter are equal. To engage the direct drive clutch in a design according to Fig. 1 it is, however, necessary to give the guide member a speed corresponding to that of the impeller or even slightly higher. As the free-wheel **110** at the same time is so designed as to prevent the guide member shaft from running faster than the turbine shaft, the adjustment to the correct relation between the impeller and guide member speeds may only be effected by decreasing the torque of the engine. A momentary throttling of the fuel supply to the engine and by this decreasing the speed of the primary member can temporarily make the guide member shaft overrun the pump casing speed, whereby the sleeve **126** turns and unlocks the sleeve **136**, so that direct drive can be established by moving over said sleeve **136** to the left, thereby engaging the rims **124** and **138**. Consequently, direct drive is achieved, owing to the fact that, as mentioned above, the free-wheel **110** prevents the turbine from running slower than the guide member.

The return from direct drive to converter drive is achieved by moving the sleeve back into the position shown in Fig. 1.

Fig. 2 shows a slightly modified design of the arrangement according to Fig. 1.

The difference between the two designs is chiefly that the synchronized clutch **120** is exchanged for a friction clutch **150** of conventional type. This friction clutch consists of a disc **152**, rigidly connected to the pump casing, against which a second friction disc **154** can be pressed by means of a disc **156**, supported by—but axially movable on—a ring **148**, rigidly connected to the pump casing **54**. The friction disc **154** in its turn is carried by a hub **160**, which by means of key and groove or splines **162** is axially movable on the extension **62** of the guide member shaft. In order to have the possibility of engaging and disengaging the friction clutch a spring disc **168** is mounted between a projection of part **158** of the rotating pump casing, a ridge on the disc **156** and an outer ring on the axially movable ball bearing **166**. The inner ring of the bearing **166** is secured to a sleeve **164**, provided with connection means for a forked lever (not shown by the figure) and slidably, but not turnably, journaled on a hub **170**, fixed to the casing **16** and concentric in relation to the turbine shaft. Direct drive can for this arrangement be established independently of the torque on the reaction member and without previously decreasing the motor speed, in contrast to the design according to Fig. 1. The return to converter drive is effected in the same way as mentioned for the previous design by disengaging the friction clutch.

Figs. 3 and 4 illustrate a design where the connection between the pump casing and the guide member is established by means of a so-called centrifugal weight clutch, which is engaged as soon as the reaction member begins to rotate. The change from converter drive to direct drive and vice versa for this arrangement is established automatically at a suitable point of operation of the hydraulic system.

Owing to the fact that other parts of the converter have not been essentially altered compared to the description in connection with Figs. 1 and 2, the following detailed description of this arrangement according to Figs. 3 and 4 is limited to the clutch itself between the pump casing and guide member shaft as well as the operation of said clutch when engaged and disengaged.

The extension **62** of the guide member shaft is formed at one end of a flange **202**, supporting a ring **204** with U-shaped outer rim where a number of brake shoes **206** is mounted. In the illustrated design (Fig. 4) eight such shoes are equally spaced around the circumference. Each brake shoe is pressed against the bottom of the rim by an elastic pin **208**, which at one end is fixed to a flange of the ring **204**, whereas the other free end passes through a hole in the thin flange **210** on the side of the shoe facing said ring. The force that presses the shoes against the ring **204** can be comparatively small and only insure that the shoes are out of touch with the brake band **212**, fixed to a ring member **214** projecting from the pump casing, when the guide member is stationary.

Any leakage fluid from the seals is caught by the groove **216** at the inside of ring **204** and conducted through bores **218** toward the rim edge **220**, formed in a suitable manner according to the figure, so that the leakage fluid flows down or is thrown outwardly against the groove **222**, where the fluid is collected and drained through the bore **224** into the lower part of the groove. This arrangement is used in order to keep the braking surfaces dry and, consequently, guarantee a maximum of braking effect.

The clutch operates in the following manner: at low speed ratios n_2/n_1 between the primary and secondary members the guide member is actuated by a torque from the fluid in the circuit, tending to turn said guide member in a direction opposite to that of the primary member, a rotation, however, that is prevented by the free-wheel **118**, locking the guide member to the casing **16** as long as the torque has the mentioned direction. With increasing speed ratio n_2/n_1 the reaction torque decreases progressively first to zero and then changes direction of action, so that the guide member, no longer locked against rotation by the free-wheel **118**, starts rotating in the same direction as the impeller and turbine. The centrifugal force that acts on the brake shoes when the guide member rotates exceeds the spring force of the pins **208**, and as soon as the shoes **206** come in contact with the brake band **212** the rotation of the guide member is further accelerated by the directly transmitted primary torque. This action rapidly increases the surface pressure between the shoes and the brake band and, consequently, rapidly increases the friction torque of the clutch so that the guide member is locked to the pump casing. At the same time the free-wheel **110** causes turbine and pump or primary member to rotate at the same speed.

The return from direct to converter drive with this clutch arrangement occurs as soon as the torque speed ratio M_1/n_1 of the engine exceeds a certain limit value, where the engine torque is higher than the maximum torque transferable via the centrifugal weight clutch at the actual speed n_1. In such case the clutch begins to slip, further diminishing the friction torque by decreasing surface pressure between the shoes and the friction band **212**, as well as by decreased specific friction coefficient when changing from stationary to sliding friction, at the same time that the fluid in the circuit begins to circulate, acting on the guide member with a torque having a direction opposite to that of the rotation of the impeller.

In the arrangement according to this invention illustrated by Fig. 5 and associated Figs. 6–10 the connecting member between the primary and the reaction member consists of a fluid-actuated friction clutch **250**, but other parts of the conventer are substantially identical to previously described arrangements, especially with regard to the position of the free-wheels between the turbine and the guide member shafts as well as between the guide member shaft and the casing. The inlet and outlet openings for the operating fluid to and from the clutch are automatically controlled by a valve system, receiving its operating impulses from the guide member. In any case, however, this shifting can also be done manually by means of a separate control system.

The extension **62** of the guide member shaft has in Fig. 5 a bell-shaped member **252** carrying in its turn three friction discs **254**, a stationary side wall **256** and a second axially movable side wall (or plunger) **258**, the

latter wall being provided with sealing members **260** and **262** respectively at the inner and outer circumference against the bell **252**, which in this way creates an operating chamber **264** for the pressure fluid delivered by the gear pump **74**. By means of a radial bore **266** in said bell **252** the operating chamber **264** can be made to communicate with or be cut off from the channel system for pressure fluid from the pump **74** by shifting a valve **268**, mounted in the outer part of said bore **266**. In cutting off the operating chamber **264** from communicating with the pressure fluid the valve **268** simultaneously opens an outlet **284** for emptying said operating chamber. The primary number of the friction clutch consists of three discs **270**, supported by a hub **272** that is fixed to the impeller casing.

The design of valve **268** on a large scale is shown by Figs. 6–9. The valve sleeve **274** is rotatably mounted in an axial bore in the bell **252** as well as in a bore through a supporting bracket **276** fixed to said bell. From the sleeve there also extends a lever with weight **278** and projection **280**; the latter rests when the clutch is disengaged—against a suitable surface of the bracket **276** owing to the spring force of the valve spring **282**. This valve position—indicated by *a* in Fig. 6—makes, as shown by Figs. 7 and 8, the outlet **286** communicate with the operating chamber **264** by the sleeve channel **284** when at the same time communication with the bore **266** is interrupted. Consequently, the operating chamber will be emptied in this position, that is the clutch will be disengaged and the reaction member disconnected from the impeller casing, which means that the torque transmission is effected via the hydraulic converter system.

As soon as the operating conditions of the converter are changed so that the reaction torque on the guide member rotates said member in the same direction as the impeller, the centrifugal weight **278** is turned over to position *b* by the centrifugal force and the valve position will be that illustrated by Fig. 9. The outlet **286** is now closed and pressure fluid enters the operating chamber from the bore **266** via the groove **288** in the valve sleeve **274** and presses the two clutch halves against each other, rigidly connecting the pump and reaction members, whereupon the free-wheel **110** between the turbine and reaction member will accomplish the direct drive. The pressure fluid to the operating chamber must not necessarily be supplied under especially high pressure, due to the fact that the influence of the centrifugal force will in any case provide sufficiently high fluid pressure on the piston **258**.

The change from direct drive to converter drive is achieved automatically in a similar manner as described for the centrifugal weight clutch according to Figs. 3 and 4. With increasing value M_1/n_1 the engine torque exceeds the friction torque of the clutch at a certain speed n_1, so that the two clutch halves begin to slip. Owing to this the fluid pressure in the operating chamber of the clutch decreases rapidly when at the same time the fluid in the converter circuit starts circulating, thereby providing a reaction torque on the guide member and slowing down same into stationary position against the free-wheel **118**, whereupon spring **282** can bring the sleeve from position *b* back to position *a*, in order that the operating chamber **264** of the clutch may be emptied.

Fig. 5 shows also an arrangement by the aid of which the shifting to and from direct drive can be achieved manually. The ring **290**, fixed to the casing, is provided on the outer race with threads, engaging an outer ring **292** sliding in cooperating threads when turned by the lever **294**, which with one end acts in a groove on the circumference of the ring **292** and with its other end is connected to the shaft **296**, rotatably mounted in the casing and in its turn controlled by the aid of the lever **298**. Obviously, when turning the lever **294** in one direction or the other the outer ring **292** will be moved to the right or the left. If moved to the right the beveled surface **300** of said ring will raise the weight **278** from position *a* to position *b*, which, as has been described earlier, results in a supply of pressure fluid to the clutch **250** and shifting from converter to direct drive. If the ring is moved to the left, the annular projection **302** of said ring will brake against the valve sleeve part **304**, creating a turning movement of said valve sleeve and returning it to position *a*, so that the drive will be done via the hydraulic converter system.

In order to help the lever **294** and the ring **292** into neutral position between the operating positions described above, the ring **292** is provided with two projections **306** as retainers for the springs **308**, **310**, **312** and **314**, extending between the casing and the retainers as well as between the retainers and the lever **294**.

Figs. 11 and 12 show a modified design of the converter as illustrated by Fig. 5, a design permitting the turbine side to rotate as a double or counter-rotation system as well as single rotation system. The shifting from one type of operation to the other is done by means of servomotors, and to transmit torque from the reaction member operating as a double rotation stage to the turbine shaft a reverse gear has been adopted.

The friction clutch between the pump and the reaction member is of the same design as described in connection with Fig. 5, but the fluid pressure regulating sleeve is in this case controlled by a servomotor **352**, the piston of which acts on a forked lever **354** that in its turn axially moves the ring **356** and the sleeve **350** fixed to said ring.

The free-wheel **118** according to Figs. 1–5 is substituted by a brake **358** between the reaction member and the stationary casing (as shown in Fig. 11), said brake consisting of a brake band **360** and a lever system **364** actuated by the servomotor **362**. The brake drum is formed on the bell **252**, surrounding the friction clutch. The brake has to lock during those operating conditions when the guide vanes must operate as stationary reaction member but is free during double rotation and direct drive.

The reverse gear between the extension **62** of the guide member shaft and the turbine shaft **44** consists of a gear **366** formed as one single piece with the extension **62**, a gear **368** secured to the turbine shaft and planet gears **370**, supported by pins projecting from a ring **376** rotatably supported by the bearings **372** and **374**. A free-wheel **378** between said rings and the casing prevents the ring from rotating in a direction opposite that of the pump but leaves it free to rotate in the same direction. In other words, the reverse gear is inactive under all conditions except when torque applied in counter-rotation direction from the blading **46** is transmitted through the gear.

The pressure fluid to the friction clutch and to the two servomotors is provided by a gear pump **380**, driven by the primary member of the converter via the three gears **382**, **384** and **386**. The pump sucks in fluid through the opening **388** and delivers it to the bore **390**, branching into two bores **392** and **394**, the first of which communicates with the channel system of the regulation sleeve **396** and the second of which is connected to the converter circuit in substantially the same way as has been described for the pump **74** in Fig. 1.

At a valve position as indicated by *a* in Fig. 11 the communication between the servomotors and the pressure pipe **392** is interrupted and, instead, said motors are in open communication with the bottom tank of the converter via the channels **398** and **402**, **400** and **404** respectively, so that the spring-loaded servomotor pistons can take the positions that correspond to emptied cylinders. The servomotor **352** and the sleeve **350** are thereby mechanically so connected via the ring **356** and the lever **354** that said sleeve establishes open communication between the operating chamber **264** and the outlet opening **286** at the same moment when the pressure pipe is closed,

a position corresponding to disengaged friction clutch. The servomotor **366** whose piston rod acts on the lever system **364** connected to the brake band **360** also releases the guide member in this position, so that said guide member is mechanically connected to the turbine shaft only by the reverse gearing but, for the rest, can rotate freely under the influence of the fluid flow in the converter circuit. At low speed ratios n_2/n_1, for instance at starting, the arrangement as to Fig. 11 is especially advantageous due to the fact that the reaction member rotates in a direction opposite to that of the blade rings **34** and **36** so that the high torque multiplication of the double rotation turbine can be utilized.

Fig. 13 is a diagram showing the relation between the efficiency η and the output torque M_2 as function of the speed ratio n_2/n_1 in a converter according to Figs. 11 and 12. The efficiency curve consists of three branches *a*, *b* and *c* corresponding to the efficiency curves for double rotation, single rotation and direct drive respectively.

In Fig. 12 the arrow A indicates the direction of rotation of the turbine shaft and the arrow B the corresponding direction of the guide member during double rotation drive. As appears from Fig. 12 the design of the free-wheel **110** permits such drive. The torque acting upon the reaction member is carried over to the turbine shaft by means of the before-mentioned reverse gears **366**, **368** and **370**. The ring **376** is subjected to torque trying to turn the ring in a direction opposite to that of the impeller but this movement is prevented by the free-wheel **378**. Due to smaller diameter of the gear **366** connected to the counter-rotating blading **46** as compared with the ring gear **368** connected to the driven shaft, the speed of rotation of the counter-rotating blading is higher than that of the forwardly rotating turbine blading **34**, **36**.

If the regulation sleeve is moved to position *b* in Fig. 11 the conditions for the servomotor **352** will be unchanged but the outlet of the other servomotor **362** is closed and the working chamber is connected to the pressure pipe **392**, whereupon the inducted fluid moves the servomotor piston to its other end position, immediately resulting in a locking of the guide member to the casing by means of the brake **358**. After that, the secondary member of the converter operates as a single rotation turbine having an efficiency curve the peak of which is moved to higher n_2/n_1 values as compared with the characteristics of the double rotation system.

During this operation at point *b* (single rotation) the ring **376** with the planet gears **370** rotates between the gears **366** and **368** in the same direction as the turbine shaft without transmitting any torque, released by the free-wheel **378**.

By moving the regulation sleeve **396** over to position *c* the two servomotors reverse their action compared with position *b*. The servomotor **362** is cut off from the pressure pipe **392** and the outlet for the operating fluid opens again to the bottom tank, resulting in a release of the brake band **360** acting on the reaction member. Simultaneously, the outlet from the operating chamber of the servomotor **352** is closed and communication opened to the pressure pipe **392**, resulting in a movement of the ring **356** that carries the sleeves **350**, so that, after closing the outlet **286**, the operating chamber **264** of the friction clutch is made to communicate with the radial bore **266** with the result that pressure fluid starts acting upon the piston **258** and engages the clutch compelling the reaction member and, by means of the free-wheel **110**, also the turbine member, to have the same speed as the pump. Direct drive is thus established.

The reverse gearing **376** now rotates in the same direction and with the same speed of rotation as the remaining rotating parts of the converter.

The arrangement shown in **410** only has the purpose of lifting the brake band from the brake drum as soon as the brake is released in order to prevent useless wear of the brake members.

Figs. 14–15 show a modification of the reverse gearing as illustrated in Figs. 11–13.

The planet gears **500** are mounted on shafts **502**, directly connected to the converter casing. The planet gears mesh with a toothed rim on the bell **252** as well as a gear **512** journaled at **508** and **510** on the sleeve **506**.

In the same manner as for the previously described arrangement the position *a* corresponds to a condition of operation where the reaction member rotates in a direction opposite to that of the pump, that is double rotation drive, at low speed ratios n_2/n_1. A free-wheel **514** during this operation prevents the sun gear **512** from rotating relative to the sleeve **506** and the turbine shaft **44**.

In this instance, in contrast with the case of the construction shown in Fig. 11, the blading **46**, connected to the ring gear **504**, rotates at lower speed than does the forwardly rotating turbine blading **34**, **36** which is connected to the smaller sun gear **512**. The reversing gearing can in this case as well as in the design according to Figs. 11–13 be composed of conical gears in order to facilitate a changed gear ratio, if desired.

In position *b* of the sleeve **596** the band brake locks the reaction member to the casing. The sleeve **506**, released by the free-wheel **514**, now can rotate freely in relation to the stationary gear **512**.

In position *c* of the sleeve **596** the band brake releases the reaction member at the same moment when the friction clutch is engaged and connects the pump and reaction members. The free-wheel **110** between the reaction and turbine members also forces the latter to take primary speed when at the same time the free-wheel **514** is free-wheeling.

With the double rotation type of converter, of which the embodiments shown in Figs. 11 and 13 are examples, higher torque multiplication at stall is obtained than with a single rotation converter, other things being equal. The reason for this may be explained as follows. In all cases the secondary or output torque must equal the sum of the primary or input torque and the reaction torque transmitted to the stationary casing or other stationary abutment. This may be expressed for the cases of the two different types of converters by means of the following equations:

Single rotation—$M_2=M_1+R_1(R_1=R)$
Double rotation—$M_2=M_1+R_1(R_1=R+R\times a)$

in which:

M_2 is secondary torque;
M_1 is primary torque;
R_1 is reaction torque transmitted to the casing;
R is torque hydraulically applied to the reaction blading;

a is gear ratio of $\frac{\text{reaction member}}{\text{driven member}}$

In the case of the single rotation converter all of the hydraulically applied reaction torque R is transmitted to the stationary casing and therefore R_1 equals R. In the case of the double rotation converter, however, in which the reaction and driven members are geared together, the reaction torque transmitted to the casing through whichever member of the gearing may be anchored is not equal to the reaction torque applied to the blading but is equal to the value of that torque plus the value of that torque multiplied by the ratio of the gearing between the reaction member and the driven member. Thus, in all cases, the value of M_2 at stall, other things being equal, will be greater in the case of the double rotation converter than in the case of the single rotation converter, by the amount of the factor $(R\times a)$.

If we now consider the embodiment of the double

rotation converter shown in Fig. 11, it will be apparent from the drawing that the diameter of the sun gear **366** to which the reaction member is connected is approximately half the diameter of the ring gear **368** connected to the driven member. Consequently, the value of the gear ratio *a* is approximately 2 and represents torque multiplication of the hydraulic torque imposed on the blades **46**, which as has previously been noted, rotate at a higher speed than do the turbine blades **34** and **36** with this gear arrangement.

On the other hand, in the case of the arrangement shown in Fig. 14 the reverse is true, the diameter of the ring gear **504** to which the reaction blades are connected being approximately double the diameter of the sun gear **512** which is connected through the free-wheel **110** to the turbine member. In this case the gear ratio *a* is 0.5, so that the value of the torque applied to the driven member from the reaction blading is less in this case than in the construction shown in Fig. 11. However, even in this case the value of the total torque on the secondary member is greater than would be the case with a single rotation converter.

The reason for utilizing different gear arrangements resulting in different stall torque characteristics in converters which are otherwise substantially the same is best illustrated by comparison of Figs. 13 and 16 showing secondary torque and efficiency characteristics in terms of the speed ratio n_2/n_1 between the primary and secondary members. As shown in both of these figures the curve *a* represents the efficiency of the converter in double rotation operation, curve *b* the converter efficiency in single rotation operation and *c* the efficiency in direct drive. As previously noted the secondary torque M_2 at stall obtainable with the design of Fig. 11 is higher than the comparable torque obtainable with the design of Fig. 14, but owing to the higher rate of speed of rotation of the reaction blading in double rotation operation, which is characteristic of the design of Fig. 11, the efficiency of the converter not only rises very rapidly with increase of n_2/n_1 from stall but also falls comparatively rapidly at a relatively low value of n_2/n_1. On the other hand, with the arrangement shown in Fig. 14, while the secondary torque at stall is lower, the lower speed of rotation of the reaction blades in double rotation operation results in the production of an efficiency curve which while not rising so rapidly from stall as the curve of Fig. 13, maintains relatively high efficiency until a higher value of n_2/n_1 is obtained. Consequently, double rotation operation may be efficiently maintained to higher vehicle speed with the arrangement shown in Fig. 14 than with the arrangement shown in Fig. 11, and while the maximum torque obtainable at stall with the former arrangement is lower than with the latter, the advantage of the relatively higher torque multiplication obtainable with double rotation operation may be used over a wider vehicle speed with the former than with the latter.

The value to be chosen of the gear ratio between the reaction and turbine members in a double rotation converter will be dictated largely by the operating characteristics desired for a given vehicle, and also the power and torque characteristics of the engine which furnishes the motive power. In this connection, however, it is to be noted that with a proper gear ratio between the reaction member and the turbine member of a double rotation converter a sufficiently high stall torque multiplication characteristic may readily be obtained in the converter to equal or exceed the adhesion capacity of the vehicle wheels when the motive power for the converter is supplied by an engine of the size and power suitable for giving desired operation characteristics to the vehicle throughout its normal speed range.

This application is a division of my U.S. application Serial No. 29,446, filed May 27, 1948, now U.S. Patent No. 2,719,616, granted October 4, 1955.

What I claim is:

1. A hydrodynamic torque converter providing a hydraulic drive utilizing a primary member providing pump blading, a secondary member providing turbine blading and a reaction member providing reaction blading, gearing interconnecting the reaction and secondary members for rotation of the reaction member in a direction counter to that of the secondary member to increase the torque transmitted to the secondary member, fluid pressure actuated means for selectively holding the reaction member against rotation or releasing it for rotation in either direction, means comprising a fluid pressure actuated clutch for selectively establishing an alternative drive between said primary member and said secondary member and a fluid pressure activated control system having a control member for selectively controlling said fluid pressure actuated means and said fluid pressure actuated clutch.

2. A converter as set forth in claim 1, in which said gearing includes a part anchored to a rotationally stationary abutment and gears associated with said part for causing the reaction member to rotate in counter direction at a speed greater than that of the secondary member in forward direction.

3. A converter as set forth in claim 1, in which said gearing includes a part anchored to a rotationally stationary abutment and gears associated with said part for causing the reaction member to rotate in counter direction at a speed less than that of the secondary member in forward direction.

4. A hydraulic transmission comprising a hydrodynamic torque converter having a primary driving member, a secondary driven member and a reaction member, means including a releasable clutch for providing a mechanical driving connection from the driving member to the driven member, releasable brake means for holding said reaction member rotationally stationary or permitting it to rotate in either direction, gearing operatively connecting said reaction and driven members for transmitting torque in forward direction to the driven member from the reaction member when the latter is rotating in counter-direction, an automatically releasable one way clutch operatively associated with said gearing, said one way clutch being arranged to engage when the reaction member rotates in counter-direction and thereby cause reaction torque to be transmitted to a rotationally stationary element and to disengage when said reaction member rotates forwardly, and means for selectively engaging and disengaging said clutch and said brake means.

5. A transmission as set forth in claim 4, including a first fluid pressure actuated servomotor for controlling engagement and disengagement of said clutch, a second fluid pressure actuated servomotor for controlling engagement and disengagement of said brake means, and a common control member for controlling the operation of said servomotors, said control member being selectively movable to three operative positions in the first of which positions both said clutch and said brake means are released, in the second of which positions the brake means is engaged and the clutch is disengaged and in the third of which positions the brake means is disengaged and the clutch is engaged.

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