(12) UK Patent Application (19) GB (11) 2 033 983 A

(21) Application No **7933982**
(22) Date of filing **1 Oct 1979**
(30) Priority data
(31) **53/120980**
(32) **29 Sep 1978**
(33) **Japan (JP)**
(43) Application published
29 May 1980
(51) **INT CL³**
F16H 5/64
(52) Domestic classification
F2D A005 A009 A022 A106 A107 A133 C085 C090 C094 C095 C097 C098 C099 C100 C116 C117 C118 C121 C122 C123 C126 C127 C129
(56) Documents cited
GB 1489019
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GB 1315409
GB 1215669
(58) Field of search
F2D
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(54) Shock Reducing Apparatus for an Automatic Transmission

(57) A transmission includes a plurality of friction units (104, 105, 106, 107) to effect shifting between gear ratios, one of which (104) includes driving and driven elements which may operate with a speed differential at the time of shifting between gear ratios. A first or line pressure (16) is generated which is variable in a first pattern corresponding approximately to the pattern of torque output of the vehicle engine. This pressure is supplied to a friction unit (106) other than the one friction unit to operate such other friction unit. A valve (15) is instrumental in generating a second pressure which is variable in a second pattern corresponding approximately to a combination of the pattern of the engine torque output and the pattern of differential speed between the driving and driven elements of said one friction unit (104) at the time of engagement of the elements. The second pressure is directed to said one friction unit (104) to operate such one friction unit.

FIG.2

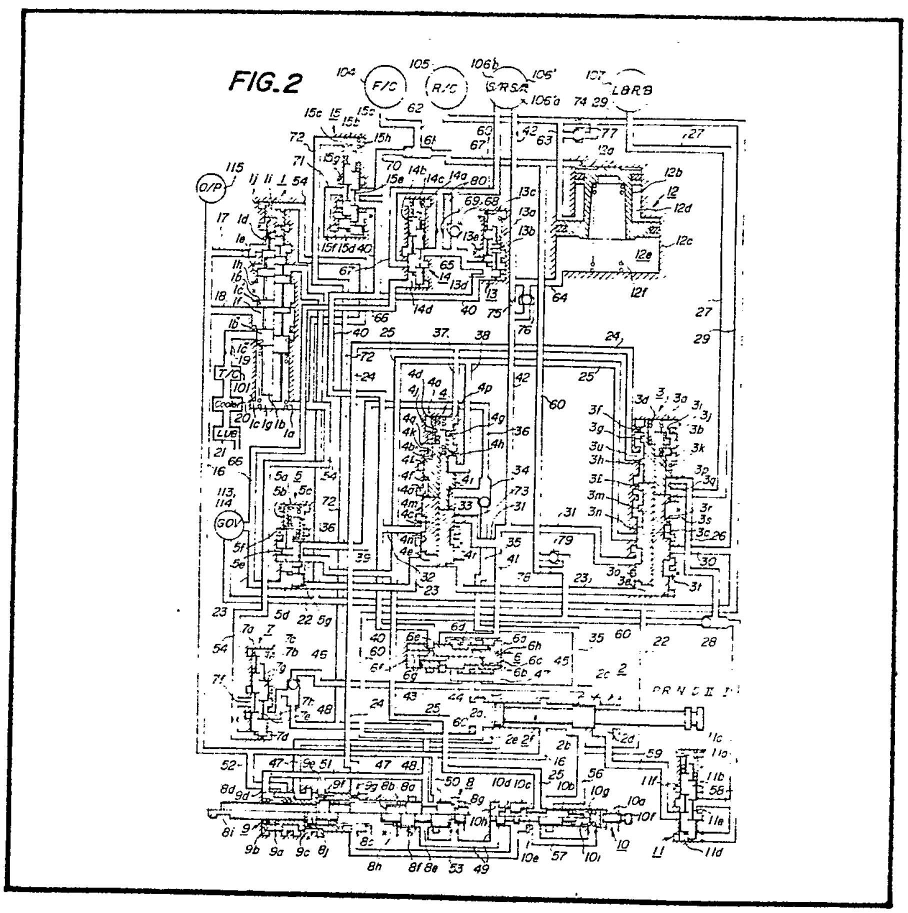

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FIG. 1

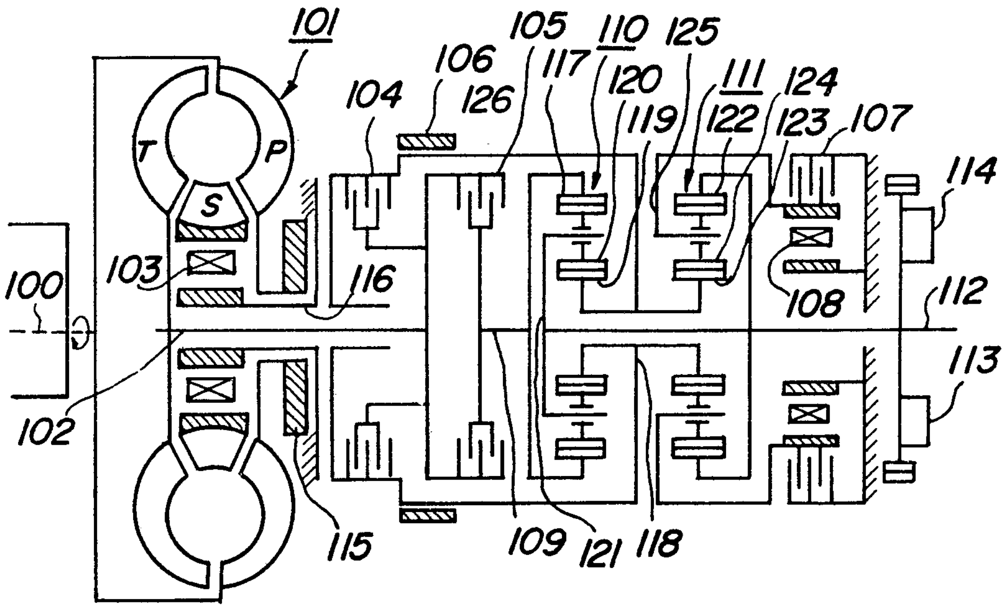


FIG. 2

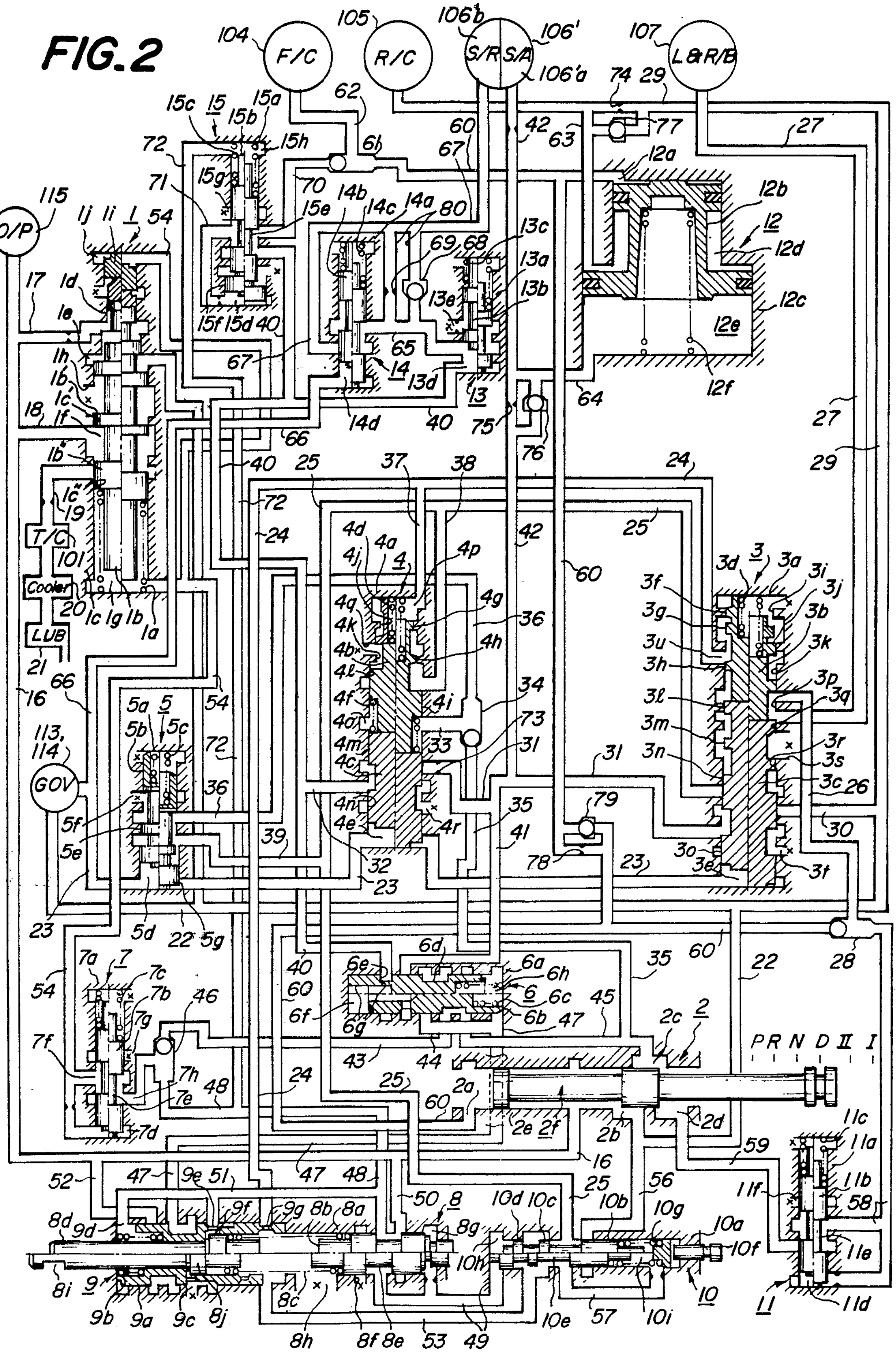
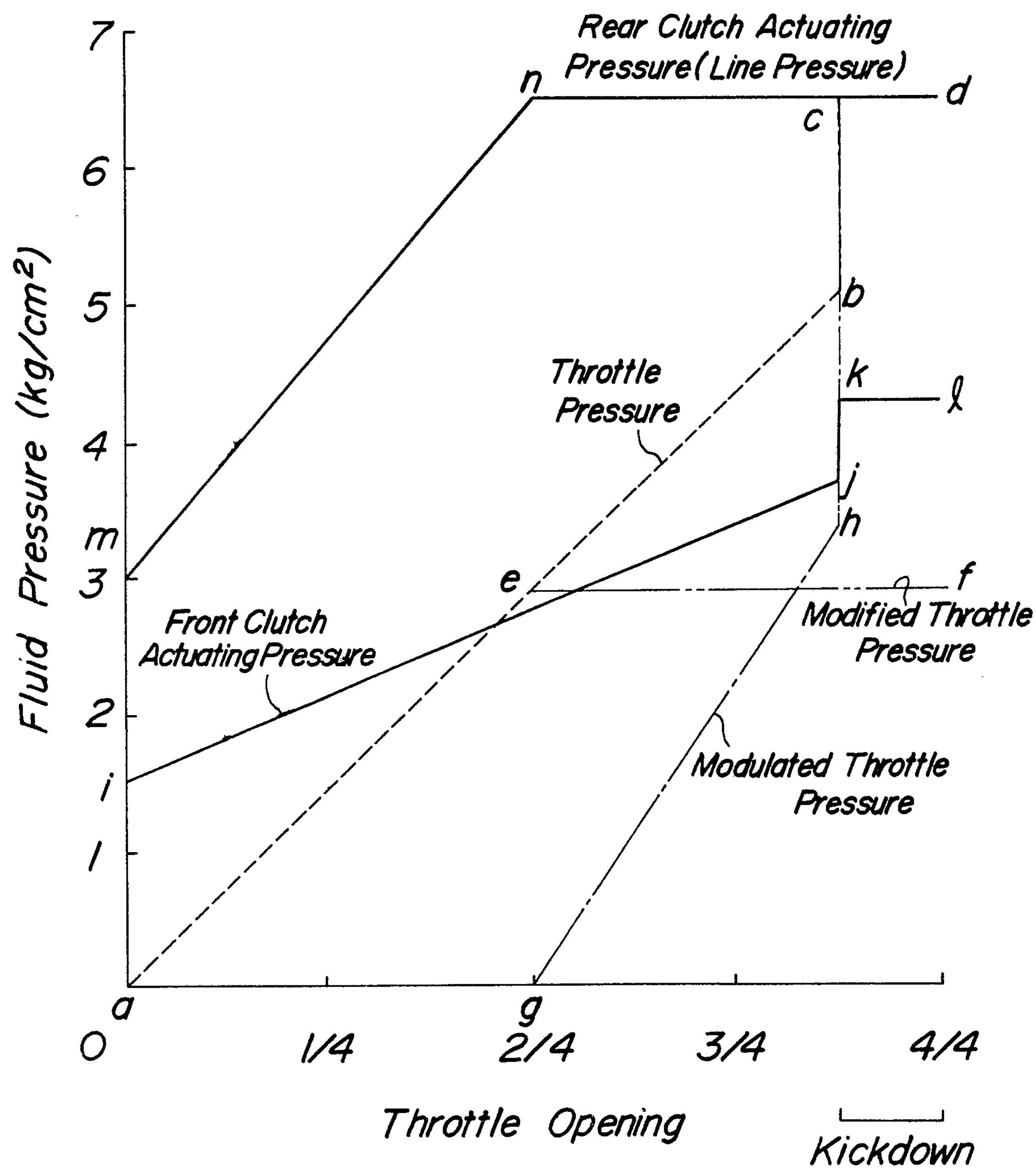


FIG. 3


7
6
5
4
m
3
2
i
1
Fluid Pressure (kg/cm²)
Rear Clutch Actuating Pressure (Line Pressure)
n
d
c
b
k
l
j
h
e
f
Throttle Pressure
Modified Throttle Pressure
Front Clutch Actuating Pressure
Modulated Throttle Pressure
a
g
0
1/4
2/4
3/4
4/4
Throttle Opening
Kickdown

SPECIFICATION

Shock Reducing Apparatus for an Automatic Transmission

The present invention relates to an apparatus for hydraulic control of an automatic transmission.

The transmission mechanism of an automatic transmission for an automotive vehicle has incorporated therein a plurality of friction units which are to be selectively made operative and inoperative to make shifts between various forward and reverse drive gear ratios and portions, as is well known in the art. In an automatic transmission having three forward drive and one reverse drive gear ratios, such frictional units typically include front clutch, a rear clutch, a second brake, and a low-and-reverse brake. Of these transmission clutches and brakes, the rear clutch in particular is maintained coupled throughout the conditions in which the gear ratios in the automatic and manual forward drive ranges are to be selected and established in the transmission mechanism. If all the other frictional units than the rear clutch are held inoperative, the transmission mechanism is conditioned to produce the first gear ratio in the automatic forward drive range in cooperation with a one-way clutch also incorporated into the transmission mechanism. If the second brake is put into operation additionally to the rear clutch, then the second gear ratio in the automatic or manual forward drive range is selected depending upon the position to which the manually operated transmission gear shift lever has been moved. If the second brake is thereafter released and, in turn, the front clutch is actuated to couple, a shift is made from the second gear ratio to the third gear ratio in the automatic forward drive range. If, on the other hand, the front clutch and the low-and-reverse brake are in operation concurrently with the rear clutch held inoperative, then the reverse drive gear ratio is selected in the transmission mechanism.

The clutches and brakes which effect the shifting between the various gear ratio conditions of the transmission mechanism are hydraulically operated by means of a hydraulic control system which is basically operated by manipulating the above-mentioned transmission gear shift lever which is usually installed on the floor, the steering column or otherwise of an automotive vehicle. More specifically, each of the clutches and brakes provided in the transmission mechanism is operated by a control fluid pressure, or line pressure, produced by a pressure regulator valve incorporated into the control system. In the hydraulic control system of a known automatic power transmission, the line presssure thus produced by the pressure regulator valve is developed on the basis of another control pressure, or throttle pressure, which represents the opening degree of the throttle valve provided in the carburetor of an internal combustion engine with which the power transmission is to operate in an automotive vehicle. The clutches and brakes for controlling the shifts between gear ratios are, for this reason, operative with forces which vary with the opening degree of the carburetor throttle valve.

The line pressure supplied from the pressure regulator is distributed to the transmission clutches and brakes selectively by and through a suitable number of gear shift valves which are responsive to both the throttle pressure and a third control fluid pressure, or governor pressure, which represents the road speed of the vehicle in operation. The transitive points, or shift points, at which shifts are to be automatically made between the gear ratios available are thus determined by the relationship between the vehicle speed and the opening degree of the carburetor throttle valve.

For good driveability under low load condition of the engine with small throttle opening degrees, it is desirable that an upshift from the lower gear ratio to the higher gear ratio takes place at a vehicle speed falling in a narrow range or at a predetermined vehicle speed.

As mentioned in the preceding, according to the conventional practice, a line pressure is produced based on a throttle pressure so that the line pressure varies approximately with the characteristic curve of the engine torque. If a hydraulic control system including a rear clutch servo and a front clutch servo is operated with this line pressure, there will not take place a noticeable shock nor slippage in the case of the rear clutch since this clutch is kept energized or engaged during forward running, but there will occur a noticeable shock upon energization of the front clutch. The reason for the occurrence of the noticeable shift shock in the case of the front clutch is that the magnitude vs. throttle opening degree characteristics of this line pressure does not agree with that of a desired servo actuating pressure which is considered to be suitable for actuating the front clutch in a shockless manner over the whole throttle opening degrees. Explaining it further, if, in design, a magnitude of the line pressure is tailored to a suitable magnitude for actuation of the front clutch upon its engagement during upshifting under high speed high load engine operating condition, a magnitude of the line pressure for the clutch actuation becomes excessively large under a low speed, low load engine operating condition, thus causing a noticeable magnitude of shift shock under this low speed, low load engine operating condition. If a magnitude of the line pressure is tailored to a suitable magnitude for the clutch operation of the front clutch under a low speed, low load engine operating condition, a magnitude of the line pressure for the clutch actuation becomes excessively small under a high speed, high load engine operating condition, thus causing a slippage of the front clutch under this high speed, high load engine operating condition.

According to another conventional practice, a line pressure is produced based on a throttle pressure so that the line pressure varies in proportion to the throttle opening degree without precisely approximating the characteristic curve of the engine torque. If this line pressure is used, a noticeable magnitude of upshift shock under a low speed, low load engine operating condition is unavoidable since the line pressure is set to have a magnitude large enough, not zero, even at zero throttle opening degree for the purpose of producing a sufficiently high pressure for the lubrication and for the production of a governor pressure.

Considering a shift shock taking place upon engagement of the front clutch during upshifting from the second gear ratio to the third gear ratio, the magnitude of the shift shock is considered to be variable with the kinetic energy differential between the clutch elements immediately before the clutch engagement. The kinetic energy differential is in part a function of the engine torque and vehicle speed.

The present invention is based on the recognition that the shift shock, during upshifting, will be reduced by actuating a frictional unit to be actuated for this upshifting with a servo actuating pressure which will have a magnitude, during the upshifting at least, indicative of not only the engine torque but also the vehicle speed at the time of the upshifting.

An object of the present invention is to provide a hydraulic control system for an automatic transmission, wherein an upshift between two gear ratios during automatic forward drive range takes place without a noticeable magnitude of shock.

According to one feature of a hydraulic contol system of the present invention, a friction unit which is to be engaged for effecting an upshift is operable or actuatable with a fluid pressure which is variable in pattern corresponding to a combination of the pattern of the output torque of the vehicle engine and the pattern of the differential speed of driving and driven elements of said friction unit at the time of upshifting.

It may be safely said that a differential speed between driving and driven elements of such a friction unit is variable in a pattern corresponding to the pattern of the vehicle speed at the time of upshifting.

A hydraulic control system according to the present invention is adapted for use with an automatic transmission for an automotive vehicle having an engine. The transmission is adapted for shifting between different gear ratios according to a shift pattern and has a plurality of units for effecting shifting, at least one of which includes driving and driven elements which are frictionally engageable under conditions which at certain points along the shift pattern involves a speed differential between the driving and driven elements. The control system of the present invention comprises: a fluid source, means communicating with the fluid source for generating a first fluid pressure which is variable in a first pattern corresponding approximately to the pattern of torque output of the vehicle engine, the first fluid pressure generating means being adapted to supply the first fluid pressure to a friction unit other than the one frictional unit, means for changing the first pattern of variation of the fluid pressure to a second pattern corresponding approximately to a combination of the pattern of engine torque output and the pattern of differential speed between the driving and driven elements of the one frictional unit at the time of engagement of the elements, the changing means generating a second fluid pressure. The first pressure generating means is coupled for communication with the changing means. The changing means is adapted to supply the second fluid pressure to the one frictional unit to operate such one frictional unit. The second fluid pressure reaching the one frictional unit is sufficient to meet torque reaction requirements and differential speed compensation requirements of the one frictional unit at all shift points and is sufficiently low to avoid substantial shocks in effecting shifting by the one frictional unit.

Another object of the present invention is to provide an automatic transmission for an automotive vehicle having an engine with a throttle which opens in degrees. The transmission is shiftable between a plurality of gear ratios. The transmission according to the present invention comprises: a friction unit with driving and driven elements which may operate with a speed differential at the time of shifting between gear ratios, said friction unit including a fluid actuator, said friction unit and its actuating means cooperating to contribute to a shifting between gear ratios, means for generating an actuating fluid pressure which is approximately linearly proportional to the degree of throttle opening, a conduit extending from said pressure generating means to said fluid actuator of said friction unit to selectively apply said linearly proportional fluid pressure thereto, whereby said linearly proportional fluid pressure reaching said friction unit is sufficient to meet torque reaction requirements and differential speed compensation requirements of the friction unit at all shift points and is sufficiently low to avoid substantial shock in effecting shifting by the one friction unit.

Another object of the present invention is to provide a hydraulic control system for an automatic transmission for an automotive vehicle having an engine with a throttle which opens in degrees. The transmission is adapted for shifting between different gear ratios and having a plurality of friction units with fluid pressure operated servos to effect shifting between gear ratios. The control system according to the present invention comprises: a fluid source, means communicating with said fluid source for generating a line pressure which varies approximately linearly with the degree of engine throttle opening from low to medium degrees of opening and which is approximately constant from medium to high degrees of throttle opening, said line pressure generating means being adapted to supply line pressure to one of the friction units, means, communicating with said fluid source, for generating a non-servo actuating throttle pressure which is proportional to the degree of throttle opening over most of the throttle opening range, means, connected for communication with both said line pressure and non-serve actuating throttle pressure generating means, for generating a reduced servo actuating pressure which is proportional to the degree of throttle opening over most of the throttle opening range by reducing said line pressure under the influence of said non-servo actuating throttle pressure, said reduced servo pressure actuating means being adapted to supply said reduced servo actuating pressure to the servo of the one friction unit, whereby said reduced servo actuating pressure is sufficient to actuate the one friction unit so as to provide an effective grip for shifting but at the same time is not so high as to create substantial shocks during shifting under the influence of the one friction unit.

Another object of the present invention is to provide a hydraulic control system for an automatic transmission for an automotive vehicle having an engine with a throttle which opens in degrees. The automotive vehicle has an operating range extending from a state in which conditions such as the carburetor throttle opening, vehicle speed, torque and the engine speed are at, or near, their minimums to states in which such conditions are at, or near, their maximums. The transmission is adapted for shifting between different gear ratios and has a plurality of friction units each with at least one fluid pressure operated actuator to effect shifting between gear ratios. The control system according to the present invention comprises: a source of fluid, means connected with said fluid source for generating one fluid pressure which is continuously variable over the majority of the vehicle operating range, means connected with said fluid source for generating another fluid pressure which is always greater than the one pressure and which varies in a generally discontinuous manner over the same operating range over which the one pressure under the action of the one pressure to provide a reduced servo-actuating pressure which is continuously variable over approximately the same range over which the one pressure is continuously variable, said one pressure generating means and said other pressure generating means both being coupled for communication with said changing means, said changing means also being coupled for communication with an actuator of one of the friction units to selectively supply said reduced servo-actuating fluid pressure to the one friction unit to effect actuation thereof, whereby said reduced servo actuating pressure is sufficiently high to actuate the one friction unit so as to provide an effective grip for shifting but at the same time is not so high as to create substantial shocks during shifting of the transmission under the influence of the friction unit.

More detailed features and advantages of the present invention will be made apparent from the following description taken in conjunction with the accompanying drawings, in which:

Fig. 1 is a schematic view showing the general construction of a transmission mechanism for which a hyraulic control system according to the present invention may be used in an automatic transmission;

Fig. 2 is a schematic view showing the arrangement of a hydraulic control system embodying the present invention and compatible with the transmission mechanism illustrated in Fig. 1; and

Fig. 3 is a graph indicating, by way of example, the characteristics of various control pressures developed in the hydraulic control system illustrated in Fig. 2 in terms of the opening degree of the carburetor throttle valve provided in an internal combustion engine.

Fig. 1 shows the construction of the power train provided in an automatic transmission of the three-forward-speed and one-reverse-speed design comprising a crankshaft 100 driven by an engine, a torque converter 101, an input shaft 102, a front clutch 104, a rear clutch 105, a second brake 106, a low-and-reverse brake 107, a one-way clutch 108, an intermediate shaft 109, a first planetary gear assembly 110, a second planetary gear assembly 111, and output shaft 112, a first governor valve 113, a second governor valve 114 and an oil pump 115. The torque converter 101 comprises a pump impeller P, a tubine runner T and a stator or reaction member S, of which the pump impeller P is driven by the crankshaft 100 so that the torque converter working oil contained therein is caused to swirl and imparts torque to the turbine runner T which is secured to the input shaft 102. The torque is further transmitted through the input shaft 102 to the change-speed gearing arrangement. The stator S is mounted about a sleeve 116 with the one-way clutch 103 interposed therebetween. The one-way clutch 103 is contructed and arranged in such a manner as to permit a rotation of the stator S in the same direction as the direction of rotation of the crankshaft 100, vis., the direction indicated by the arrow (abbreviated hereinafter as the forward rotation) and to prevent the opposite rotation of the stator (abbreviated hereinafter as the opposite rotation). The first planetary gear assembly 110 comprises an internally toothed gear 117 rotatable with the intermediate shaft 109, a sun gear 119 rotatable with a hollow transmission shaft 118, two or more planet pinions 120, each meshing with the internally toothed gear 117 and the sun gear 119 so that it rotates and moves along an orbit, and a front planet carrier 121 rotatable with the output shaft 112 and having the planet pinions 120 thereon; while the second planetary gear assembly 111 comprises an internally toothed gear 122 rotatable with the output shaft 112, a sun gear 123 rotatable with the hollow transmission shaft 118, two or more planet pinions 124, each meshing with the internally toothed gear 122 and the sun gear 123 so that it rotates and moves along an orbit, and a rear planet carrier 125 having the planet pinions 124. The front clutch 104 is operative to establish a connection between the transmission input shaft 102 to be driven by the turbine runner T and the hollow transmission shaft 118, rotatable in unison with the two sun gears 119 and 123 through a drum 126, while the rear clutch 105 is operative to connect the input shaft 102 and the internally toothed gear 117 of the first planetary gear assembly 110 through the intermediate shaft 109. The second brake 106 is operative to tighten a band around the drum 126 secured to the hollow transmission shaft 118 so as to fix the two sun gears 119 and 123, while the low-and-reverse brake 107 is operative to fix the rear planet carrier 125 of the second planetary gear assembly 111. On the other hand, the one-way clutch 108 is so constructed and arranged as to permit the forward rotation of the rear planet carrier 125, but prevent the opposite rotation of same. The first governor valve 113 and second governor valve 114 are fixed to the output shaft 112 and are operative to produce a governor pressure corresponding to vehicle speed.

Description will be hereinafter made of the power transmission paths which are established when the selector lever is in the D (forward automatic drive) position.

Under this condition, the rear clutch 105 serving as the forward input clutch is engaged. The power from the engine having passed through the torque converter 101 is transmitted, through the input shaft 102 and rear clutch 105, to the internally toothed gear 117 of the first planetary gear assembly 110. The rotation of the internally toothed gear 117 causes the planet gear 120 to rotate in the forward direction. Since the sun gear 119 tends to rotate in the opposite direction to urge the sun gear 123 of the second planetary gear assembly 111 which is rotatable with the sun gear 119 to rotate in the opposite direction, the planet gear 124 of the second planetary gear assembly 111 tends to rotate in the forward direction. The one-way clutch 108 is operative to prevent the rear planet carrier 125 from tending to rotate in the opposite direction, so that the sun gear 123 serves as a reaction brake in the forward direction. As a consequence, the internally toothed gear 122 of the second planetary gear assembly 111 rotates in the forward direction. It therefore follows that the output shaft 112 which is rotatable with the internally toothed gear 122 also rotates in the forward direction, thereby producing the first forward drive gear ratio. When, under this condition, the second brake 106 is applied after the vehicle speed has increased, the power which has passed through the input shaft 102 and the rear clutch 105 as in the first gear condition is transmitted to the internally toothed gear 117. The second brake 106 is operative to fix the drum 126 to prevent rotation of the sun gear 119, thus serving as a reaction brake in the forward direction. Accordingly, the planet pinions 120 rotate and move along an orbit around the sun gear 119 which is held stationary with the result that the front planet carrier 121 and the transmission output shaft 112 integral with the former rotate in the forward direction although with a reduction ratio higher than the speed which would be achieved under the first gear condition, thereby producing the second forward drive gear ratio. When the second brake 106 is released and the front clutch 104 is engaged after the vehicle speed has increased further, the power delivered to the input shaft 102 splits into a part transmitted through the rear clutch 105 to the internally toothed gear 117 and into the remainder which is transmitted through the front clutch 104 to the sun gear 119. Therefore, the internally toothed gear 117 and the sun gear 119 are interlocked with each other to rotate together with the front planet carrier 121 and the output shaft 112 at a common rotational speed in the forward direction, thereby producing the third forward drive gear ratio. Under this condition, the front clutch 104 and the rear clutch 105 may be referred to as an input clutch and there is no reaction brake so that the planetary gear assemblies do not lend themselves to a multiplication of torque.

The power transmission path to be established when the selector lever is in the R (reverse drive gear) position will be hereinafter described.

When this position is selected, both the front clutch 104 and the low-and-reverse brake 107 are made operative. The power from the engine having passed through the torque converter 101 is transmitted from the input shaft 102 through the front clutch 104 and the drum 126 to the sun gears 119 and 123. Since, under this condition, the rear planet carrier 125 is fixed by the low and reverse brake 107, the rotation of the sun gears 119 and 123 in the forward direction causes the internally toothed gear 122 to rotate at a reduced speed in the reverse direction with the result that the output shaft 112 which is rotatable with the internally toothed gear 122 rotates in the reverse direction, thereby producing the reverse drive gear ratio.

Fig. 2 is a hydraulic circuit diagram showing a gear shift shock reducing apparatus according to the present invention as incorporated in the gear shift control circuit of the above described automatic transmission, which control circuit comprises a regulator valve 1, a manual valve 2, a 1—2 shift valve 3, an 2—3 shift valve 4, a 3—2 downshift valve 5, a line pressure booster valve 6, a pressure modifier valve 7, a throttle valve 8, a throttle failsafe valve 9, a throttle modulator valve 10, a manual first gear range pressure reducing valve 11, an accumulator 12, a 2—3 timing valve 13, a 3—2 timing valve 14 and a front clutch pressure reducing valve 15, all these devices being connected as shown in the illustrated circuit network to the torque converter 101, the rear clutch 105, a band servo 106' for operating the above described second brake 106 (see Fig. 1), the low-and-reverse brake 107, the governor valves 113 and 114 and the oil pump 115. The gear shift shock reducing apparatus according to the present invention comprises the pressure modifier valve 7, throttle modulator valve 10 and front clutch pressure reducing valve 15 as the major component elements of the apparatus.

The oil pump 115 is driven by the engine through the crankshaft 100 and the pump impeller P of the torque converter 101 and is operative to draw in from an oil reservoir, not shown, the oil cleared of harmful dust by means of an oil strainer (not shown) and fed to a line pressure circuit 16 when the engine is in operation. The regulator 1 which is adapted to regulate the pressure of the oil to a predetermined level comprises a valve spool 1*b*, which is urged by means of a spring 1*a* to move toward a raised position indicated by the left half of the spool in the drawing, slidably mounted within a housing 1*c* and also comprises four chambers 1*d*, 1*e*, 1*f*, and 1*g*. To each of the chambers 1*d* and 1*f* is fed an oil pressure from the line pressure circuit 16 by way of oil passageways 17 and 18. To the chamber 1*e* is fed a line pressure from the port 2*b* of the manual wave 2 through an oil passageway 22 when the manual valve 2 in in any one of the D range, II range and I range positions. Indicated at 1*i* is a plug and achamber 1*j* formed above the plug and a chamber 1*g* formed below the valve spool 1*b* which are in communication with an oil passageway 54. The valve spool 1*b* has a land 1*b'* having a diameter slightly smaller than the diameter of the corresponding rib 1*c'* of the housing 1*c* so as to form therebetween a small clearance which serves as a variable-area orifice. The oil in the chamber 1*f* is constantly discharged through this clearance and a drain port 1*h* at a rate which is determined by an overlap between the land 1*b'* and the rib 1*c'* so that a high line pressure proportional to the amount of overlap is developed in the line pressure circuit 16. The valve spool 1*b* further has a land 1*b''* which is slightly smaller in diameter than the bore 1*c''* in the housing 1*c* so as to form a small clearance therebetween so that the oil in the chamber 1*f* is supplied through this clearance and an oil passageway 19 to the torque converter 101, the oil cooler 20 and the various lubricating parts 21 in the transmission mechanism.

The line pressure developed in the line pressure circuit 16 is directed to the manual valve 2, which serves as a fluid-flow direction change-over valve adapted to provide communication from the line pressure circuit 16 selectively to any of the ports 2*a*, 2*b*, 2*c* and 2*d* when the selector lever (not shown) is manipulated for gear selection, the valve comprising a valve spool 2*f* which is slidably mounted within a housing 2*e*. The valve spool 2*f* is movable between a neutral position (N), an automatic forward drive gear position (D), a manual second gear position (II), a manual first gear position (I), a reverse drive gear position (R) and a parking position (P) and is arranged to make the line pressure circuit 16 communicate with the ports indicated by the sign "o" in the following table when the selector lever is operated to move the valve spool 2*f* to these positions. The ports which are not in communication with the line pressure circuit 16 are all made open to the openings on both sides of the housing 2*e* and thus serve as drain ports.

Ranges \ Ports	2a	2b	2c	2d
P				
R	o			
N				
D		o		
II		o	o	
I		o	o	o

The first governor valve 113 and the second governor valve 114 are operative to develop a governor pressure corresponding to vehicle speed under forward drive conditions of a vehicle. When the manual valve 2 is in any of the forward drive gear positions D, II and I, the line pressure is first fed to the second governor valve 114 through the port 2*b* communicating with the line pressure circuit 16 and further by way of the circuit 22 as will be understood from the above table and, when vehicle is running, the line pressure in the second governor valve 114 is rendered into a governor pressure varying with the vehicle speed, the governor pressure being extended to the first governor valve 113. When the vehicle speed increases beyond a predetermined value, the first governor valve 113 allows the governor pressure into a governor pressure circuit 23. The governor pressure is thereafter distributed through the circuit 23 to the 1—2 shift valve 3, 2—3 shift valve 4 and 3—2 downshift valve 5 and controls the motions of these valves in the manners to be described later.

The 1—2 shift valve 3 comprises a housing 3*a* and two valve spools 3*b* and 3*c* which are arranged axially in line with each other and which are slidably mounted within the housing. That end face of the valve spool 3*b* which is furthest from the valve spool 3*c* is acted upon by a spring 3*d* and that end face of the valve spool 3*c* which is furthest from the valve spool 3*b* is located in a chamber 3*e*. The valve spool 3*b* is formed with lands 3*f*, 3*g* and 3*h* which are larger in diameter in this sequence, while the housing 3*a* is formed with ribs 3*i*, 3*j* and 3*k* which correspond to these lands, respectively. The valve spool 3*c* is further formed with lands 3*e* and 3*m* and lands 3*n* and 3*o* larger in diameter than the former two, while the housing 3*a* is formed with two ribs 3*p* and 3*q* associated with the land 3*e* and a rib 3*r* associated with the land 3*m*. The 1—2 shift valve 3 is in communication with the governor pressure circuit 23, a kickdown pressure circuit 24 and a gear shift control pressure circuit 25 as shown in the drawing and further with an oil passageway 27 which communicates with an oil passageway 26 or a drain port 3*s* depending upon the axial position of the land 3*l*. The governor pressure circuit 23 communicates with the groove between the lands 3*g* and 3*h* when the valve spool 3*b* is in the position indicated by the right half thereof and with the groove between the lands 3*g* and 3*h* and the groove between the lands 3*f* and 3*g* when the valve spool 3*b* is in the position indicated by the left half thereof. On the other hand, the gear shift control pressure circuit 25 is in communication with the groove between the lands 3*m* and 3*n* when the valve spool 3*c* is in the position indicated by the right half of the spool and closed by the land 3*n* when the valve spool 3*c* is in the position indicated by the left of the spool. The oil passageway 26 is in communication with an output port of a shuttle valve 28, and the oil passageway 27 is in communication with the low-and-reverse brake 107. The 1—2 shift valve 3 is further in communication with an oil passageway 30 leading from an oil passageway 29 branched to the rear clutch 105 from the oil passageway 22 leading from the port 2*b* of the manual valve 2 and to the governor valves 113 and 114.

Between the 1—2 shift valve 3 and the 2—3 shift valve 4 is provided an oil passageway 31 which communicates with, or isolated from the oil passageway 30 depending upon the axial position of the land 3*n*. The oil passageway 31 communicates with a drain port 3*t* when the valve spool 3*c* is in the position indicated by the right half of the spool. In the oil passageway 29 is provided an orifice 74 and a check valve 77 which are arranged in parallel with each other.

The 2—3 shift valve 4 comprises a housing 4*a* having two valve spools 4*b* and 4*c* axially arranged in line with each other and slidably mounted within the housing. That end face of the valve spool 4*b* which is furthest form the valve spool 4*c* is acted upon by a spring 4*d*, while that end face of the valve spool 4*c* which is furthest from the valve spool 4*b* is located in a chamber 4*e* with a spring 4*f* provided between the valve spools 4*b* and 4*c*. The valve spool 4*b* is formed with lands 4*g*, 4*h* and 4*i* which are larger in diameter in this sequence, while the housing 4*a* is formed with ribs 4*j*, 4*k* and 4*l* which are respectively associated with these lands. On the other hand, the valve spool 4*c* is formed with two lands 4*m* and 4*n*, and connected with the 2—3 shift valve 4 is an oil passageway 32 which communicates with, or isolated from, an oil passageway 31 provided with an orifice 73 depending upon the axial position of the land 4*m*. When the valve spool 4*c* is in the position indicated by the right half thereof, the oil passageway 32 is in communication with a drain port 4*r* and at the same time the chamber 4*o* formed between the valve spools 4*b* and 4*c* is in communication with the output port of a shuttle valve 34 through an oil passageway 33. The shuttle valve has one input port communicating with the port 2*c* of the manual valve 2 through an oil passageway 35 and the other input port communicating with an oil passageway 36. The chamber 4*e* is in communication with the governor pressure circuit 23 and a chamber 4*p* having the spring 4*d* accommodated therein is in communication with the kickdown pressure circuit 24 through an oil passageway 37. The kickdown pressure circuit 24 is such that a kickdown pressure is applied to the upper pressure acting face of the land 4*g* when the valve spool 4*b* is in the position indicated by the right half thereof and to the upper and lower pressure acting faces of the land 4*g* when the valve spool 4*b* is in the position indicated by the left half thereof. The 2—3 shift valve 4 is further in communication with the gear shift control pressure circuit 25 through an oil passageway 38 so that a gear shift control pressure is developed between the land 4*h* and the land 4*i* when the valve spool 4*b* is in the position indicated by the right half thereof. There is further provided a drain 4*q* which is open to the groove between the land 4*h* and the land 4*i* when the valve spool 4*b* is in the position indicated by the left half thereof.

The 3—2 downshift valve 5 comprises a housing 5*a* having a valve spool 5*b* slidably mounted therein. The valve spool 5*b* has one end face acted upon by a spring 5*c* and the other end face located in a chamber 4*d*. The 3—2 downshift valve 5 is in communication with the above mentioned oil passageway 36 in such a manner as to communicate with either an oil passageway 39 leading from the gear shift control pressure circuit 25, or a drain port 5*f*, depending upon the axial position of a land 5*e*, a chamber 5*d* being in communication with the governor pressure circuit 23.

The line pressure booster valve 6 comprises a housing 6*a* having a valve spool 6*b* slidably mounted therein, the valve spool 6*b* being urged to move leftwardly in the drawing by means of a spring 6*c*. The valve spool 6*b* is formed with grooves 6*d* and 6*e* and an oil passageway 6*g* for providing communication between the groove 6*e* and a chamber 6*f*. The line pressure booster valve 6 is in communication with an oil passageway 40 open to the groove 6*e* when the valve spool 6*b* is moved to the left and an oil passageway 41 open to the groove 6*e* when the valve spool is moved to the right. The oil passageway 40 is joined to the oil passageway 32 and is thus in communication with the 2—3 timing valve 13 and the front clutch pressure reducing valve 15, while the oil passageway 41 is in communication with the oil passageway 31 and through the latter with the servo apply chamber 106'*a* of the band servo 106'. The line pressure booster valve 6 is further in communication with an oil passageway

43 which is constantly open to the groove 6*d* and with oil passageways 44 and 45 which are to be selectively brought into communication with the oil passageway 43 through the groove 6*d* depending upon the axial position of the valve spool 6*b*. The oil passageway 43 is in communication with one input port of a shuttle valve 46 and the oil passageway 44 is in communication with the throttle failsafe valve 9 through an oil passageway 47 leading from a chamber 6*h* having the spring 6*c* accommodated therein, while the oil passageway 45 is in communication with the port 2*c* of the manual valve 2.

The transmission throttle valve 8 comprises a housing 8*a* having a valve spool 8*b* slidably mounted therein and a plunger 8*d* provided in line with the valve spool across a spring 8*c*. The plunger 8*d* is connected to the accelerator pedal by means of, for example, a mechanical linkage and is adapted to be moved rightwardly in the drawing from an idling position indicated by the lower half of the plunger and thereby adding to the force of the spring 8*c* when the accelerator pedal is depressed. The valve spool 8*b* is formed with a groove 8*e*. A throttle pressure circuit 48 and an oil passageway 49 are provided in communication with the throttle valve 8 in such a manner as to be constantly open to this groove. The throttle valve 8 is further provided with a drain port 8*f* which communicates with the throttle pressure circuit 48 through the groove 8*e* depending upon the aixal position of the valve spool 8*b*, while an oil passageway 50 leads from the line pressure circuit 16, the throttle valve and the oil passageway 49 is in communication with a chamber 8*g*. Increasing the force of the spring 8 by moving the plunger 8*d* rightwardly as the accelerator pedal is depressed causes a throttle pressure to be created within the chamber 8*g* and fed to the throttle pressure circuit 48, the throttle pressure being determined so as to balance with the spring force of this spring by relieving the line pressure fed to this chamber from the oil passageway 50 by drainage through the drain port 8*f*. Thus, the throttle valve 8 delivers, by modifying the line pressure, a throttle pressure which corresponds to the force of the spring 8*c* (viz., the distance of stroke of the accelerator pedal depressed) and which is proportional to the degree of throttle opening. When the accelerator pedal is depressed to a kickdown position, the plunger 8*d* compresses the spring 8*c* fully and is brought into abutting engagement with the valve spool 8*b* and forces the valve spool 8*b* to move to a limit position closing the drain port 8*f* and thereby providing communication between the throttle pressure circuit 48 and the oil passageway 50. Under these conditions, the throttle pressure is equal in value to the line pressure as indicated by plot c—d in Fig. 3.

The throttle pressure circuit 48 is in communication with the other input port of the shuttle valve 46 and further leads to the throttle failsafe valve 9 through an oil passageway 51. The throttle failsafe valve 9 comprises a sleeve 9*a* which is slidably mounted within a housing 8*a* in such a manner as to be capable of guiding the plunger 8*d*, the leftward movement of the sleeve being resiliently limited by means of a spring 9*b*. The oil passageway 47 providing communication between the line pressure booster valve 6 and the throttle failsafe valve 9 is normally open to the drain port 9*c* of the throttle failsafe valve 9. On the other hand, the oil passageway 51 is in communication with a chamber 9*c* having the spring 9*b* accommodated therein and further through a port 9*d* with a chamber 9*f* into which an enlarged portion 8*j* of the plunger 8*d* projects. An oil passageway 52 leads from the line pressure circuit 16 to the throttle failsafe valve 9 wherein this oil passageway is normally closed, but under an unusual condition when the sleeve 9*a* takes up the lower half position in the drawing, the oil passageway communicates with the oil passageway 47, the operation of this valve under this condition being described in detail later. During the movement of the plunger 8*d* in such a direction as being depressed, the throttle pressure developed in the throttle pressure circuit 48 is directed through the oil passageway 51 and the port 9*e* to the chamber 9*f* and acts on the enlarged portion 8*j* of the plunger 8*d* and imparts to the plunger 8*d* a force effective to move the plunger inwardly against the force of the spring 8*c*, so the accelerator pedal is prevented from being excessively loaded by the spring 8*c* when the pedal is depressed. When, furthermore, the plunger 8*d* is moved into a kickdown position, the kickdown pressure circuit 24 which has been in communication with the drain port 8*h* through the port 9*g* is isolated from the drain port 8*h* and is permitted to communicate with the oil passageway 51 through the port 9*e*, chamber 9*f* and port 9*g*. Under this condition, the valve spool 8*b* is moved rightwardly in the drawing as previously described and, as a consequence, the line pressure in the oil passageway 50 is passed into the throttle pressure circuit 48 without being drained off, thereby developing in the circuit 24 a kickdown pressure which is equal to the line pressure. The kickdown pressure thus delivered is supplied also to the throttle modulator valve 10 by way of an oil passageway 53. If damage should take place in the mechanical linkage interconnecting the accelerator pedal and the plunger 8*d*, and the plunger 8*d* should disengage from the accelerator pedal, the plunger 8*d* would be moved by a return spring, not shown, to the idling position which is indicated by the lower half of the plunger, so the plunger 8*d* would force the sleeve 9*a* to move leftwardly into the position indicated by the lower half thereof. There being no force imparted to the valve spool 8*b* by the spring 8*c* under these conditions, the valve spool 8*b* assumes a position allowing the drain port 8*f* to be slightly open and substantially fully closing the oil passageway 50. Furthermore, the oil passageway 51 is on the one hand in communication with the drain port 9*c* through the port 9*e* and the chamber 9*f* for maintaining the throttle pressure at zero level and on the other hand permits the oil passageway 47 to communicate with the oil passageway 52 for directing the line pressure to the oil passageway 47. The line pressure thus developed in the oil passageway 47 is passed through the line pressure booster valve 6, the oil passageway 43 and the shuttle valve 46 to the pressure modifier valve 7 and is modified into a pressure which is equal to the force of the spring 7*c* in a condition in which the valve spool 7*b* is held in the position indicated by the left half thereof, the modified pressure being further directed at a peak value to the chambers 1*g* and 1*j* of the pressure regulator valve 1 through an oil passageway 54, thereby causing the line pressure to rise to a peak value thereof. As a consequence, the frictional elements are actuated by the line pressure of the peak value and enable the vehicle to run to a repair shop without assistance and without causing overheating damage due to a slip in the frictional elements.

The pressure modifier valve 7 comprises a housing 7*a* having a valve spool 7*b* slidably mounted within the housing and having one end face acted upon by a spring 7*c* and the other end face located in a chamber 7*d*. The valve spool 7*b* is formed with a groove 7*e*, while the housing 7*a* is formed with an output port 7*f* constantly open to the groove, a drain port 7*g* and an input port 7*h*. The ports 7*g* and 7*h* are arranged in such a manner that one of the ports is on the point of being opened when the other of the ports is on the point of being closed during movement of the valve spool 7*b*. The port 7*f* is in communication on the one hand with the chamber 7*d* and on the other hand with the chamber 1*g* and the chamber 1*j* into which projects the plug 1*i* disposed in series with the valve spool 1*b* in the pressure regulator valve 1, while the port 7*h* is in communication with the output port of the shuttle valve 46.

With the pressure modifier valve 7 thus constructed, the spring 7*c* holds the valve spool 7*b* in a position which is lower than the position indicated by the left half in the drawing when the oil pressure developed in the port 7*h* is less than the predetermined force of the spring 7*c* (the force of the spring 7*c* as achieved when the valve spool 7*b* is held in the position indicated by the left half thereof), thereby closing the drain port 7*g* and providing communication between the port 7*f* and the port 7*h* so that the oil pressure developed in the port 7*h* is directed to the port 7*f* and further through the oil passageway 54 to the regulator valve 1. Throughout these conditions, the oil pressure is directed also into the chamber 7*d* and causes the valve spool 7*b* to move from the position indicated by the right half thereof to the position indicated by the left half thereof against the force of the spring 7*c* as the oil pressure increases. If, however, the oil pressure to be delivered from the port 7*f* tends to further increase, the valve spool 7*b* is moved upwardly beyond the position indicated by the left half of the spool and permits the port 7*f* to communicate with the drain port 7*b* with the result that the oil pressure to be delivered into the oil passageway 54 cannot be augmented beyond a certain value which is dictated by the force of the spring 7*c* in a condition in which the valve spool 7*b* is held in the position indicated by the left half thereof. In this manner, the oil pressure delivered from the pressure modifier valve 7 into the oil passageway 54 when the throttle pressure in the circuit 48 is being directed to the port 7*h* by way of the shuttle valve 46 becomes the modified throttle pressure which varies in such a manner as to remain constant for a throttle valve opening degree larger than, for instance, two fourths (2/4) of the full opening degree, as demonstrated by plot a—e—f in Fig. 3.

The throttle modulator valve 10 comprises a housing 10*a* having slidably mounted therein a valve spool 10*e* which is formed with three lands 10*b*, 10*c* and 10*d* and which has one end face acted upon by a spring 10*g* having a spring force adjustable by means of an adjuster 10*f* and the other end face located in a chamber 10*h*. The circuit 25 is arranged in the housing 10*a* in such a manner as to be at all times open to the groove between the lands 10*b* and 10*c*, while an oil passageway 53 and an oil passageway 56 leading from the port 2*b* of the manual valve 2 arranged in the housing 10*a* in such a manner that one of these oil passageways starts to open when the other thereof completely closes during movement of the valve spool 10*e*. The housing 10*a* is further connected with an oil passageway 57 aligned with the oil passageway 25, the oil passageway 57 being in communication with a chamber 10*i* having a spring 10*g* accommodated therein. Furthermore, a chamber 10*h* is provided which is in communication with the throttle valve 8 through an oil passageway 49.

With the throttle modulator valve 10 thus constructed, the valve spool 10*e* is held in the position indicated by the lower half thereof by the force of the spring 10*g* when the throttle pressure directed through the oil passageway 49 into the chamber 10*h* is at zero level. Under these conditions, the oil passageway 56 leading from the manual valve 2 is isolated from the circuit 25 and the oil passageway 57 by means of the valve spool 10*b* and the gear shift control pressure circuit 25 and the oil passageway 57 are held in communication with the drain port 8*h* through the oil passageway 53 and the port 9*g* of the throttle failsafe valve 9 so that there is no oil pressure developed in the circuit 25 and the oil passageway 57. As the throttle pressure rises, the valve spool 10*e* is moved beyond the position indicated by the upper half thereof against the force of the spring 10*g* to permit the line pressure which is directed from the port 2*b* of the manual valve 2 to the oil passageway 56. The line pressure is further applied through the oil passageway 57 up to the chamber 10*i*, and cooperates with the force of the spring 10*g*, to move the valve spool 10*e* back toward the balanced position indicated by the upper half thereof. The throttle modulator valve 10 is thus capable of regulating the line pressure from the oil passageway 56 by the throttle pressure directed into the chamber 10*h* and thereby delivering a modulated throttle pressure which appears in response to throttle opening degree larger than, for instance, two fourths (2/4) of the full opening degree and which thereafter varies substantially in direct proportion to the degree of throttle valve opening as indicated by plot g—h in Fig. 3. Under kickdown conditions in which the plunger 8*d* of the throttle valve 8 is moved inwardly, the port 9*g* is isolated from the drain port 8*h* as previously discussed and, as a consequence, the throttle pressure (indicated by the plot b—c—d in Fig. 3) corresponding to the line pressure is fed from the port 9*g* to the throttle modulator valve 10 by way of the oil passageway 53 so that an oil pressure corresponding to the line pressure is developed in the gear shift control pressure circuit 25 and the oil passageway 57 and is directed into the chamber 10*i* for moving the valve spool 10*e* to the leftward limit position thereof with the result, under kickdown conditions, an oil pressure corresponding to the line pressure as indicated by plot h—c—d in Fig. 3 is constantly developed in the circuit 25.

The manual first gear range pressure reducing valve 11 comprise a housing 11*a* having slidably mounted therein a valve spool 11*b* which has one end face acted upon by a spring 11*c* and the other end face located in a chamber 11*d*. The valve spool 11*b* is formed with a groove 11*e* and the housing 11*a* is formed with an oil passageway 58 which is constantly open to the groove, the oil passageway 58 being in communication with one input port of the shuttle valve 28 and with the chamber 11*d*. The housing 11*a* is further formed with a drain port 11*f* and is in communication with an oil passageway 59 leading from the port 2*d* of the manual valve 2, the drain port 11*f* and oil passageway 59 being arranged in such a manner that one of them starts to open when the other completely closes during movement of the valve spool 11*b*.

Thus, the manual first gear range pressure reducing valve 11 is operative to partially discharge the line pressure directed from the manual valve 2 to the oil passageway 59 and reduce the line pressure to a constant oil pressure determined by the force of the spring 11*c* in a condition indicated by the left half thereof when the I range is selected, the pressure thus obtained being delivered to the oil passageway 58 so that the low-and-reverse brake 107 which is to be also operative under the reverse drive gear condition is precluded from producing an excessively large torque transmission capacity.

The port 2*a* of the manual valve 2 is in communication with the other input port of the shuttle valve 28 through an oil passageway 60 and further with one input port of a shuttle valve 61, the output port of the shuttle valve being in communication with the front clutch 104 through an oil passageway 62. The oil passageway 60 has provided therein a parallel combination of an orifice 78 and a check valve 79 and is branched upstream of these elements for communication with a chamber 12*a* of the accumulator 12. The accumulator comprises a stepped piston 12*b* and a stepped cylinder 12*c* having the piston slidably mounted therein defining two chambers 12*d* and 12*e* in addition to the above mentioned chamber 12*a* in the cylinder, the piston 12*b* being urged upwardly in the drawing by means of a spring 12*f*. The chamber 12*d* is in communication with the oil passageway 29 through an oil passageway 63, while the chamber 12*e* is in communication with the oil passageway 42 through an oil passageway 64. The oil passageway 42 is arranged with a parallel combination of an orifice 75 and a check valve 76 which are located upstream of the accumulator 12.

The 2—3 timing valve 13 comprises a housing 13*a* having slidably mounted therein a valve spool 13*b* which has one end face acted upon by a spring 13*c* and the other end face located in a chamber 13*d*, the chamber 13*d* being in communication with the oil passageway 40. The valve spool 13*b* is urged by means of the spring 13*c* to move toward a lowered position allowing an oil passageway 65 to be open to a drain port 13*e* as indicated by the right half of the spool and is movable into a raised position allowing the oil passageway 65 to be in communication with the oil passageway 40 through the chamber 13*d* as indicated by the left half of the spool in the drawing.

The 3—2 timing valve 14 comprises a housing 14*a* having slidably mounted therein a valve spool 14*b* which has one end face acted upon by a spring 13*c* and the other end face located in a chamber 14*d*. The valve spool 14*b* is responsive to the governor pressure directed from the governor pressure circuit 23 to the chamber 14*d* by way of an oil passageway 66 and is movable into a lowered position allowing the oil passageway 65 to be open to an oil passageway 67 leading to the servo release chamber 106'*b* of the band servo 106' as indicated by the right half of the valve spool 14*b* and a raised position isolating the oil passageway 65 from the oil passageway 67 as indicated by the left half of the spool. Between the oil passageways 65 and 67 is provided a parallel combination 80 of a check valve 68 and an orifice 69 which bypasses the 3—2 timing valve 14.

The front clutch pressure reducing valve 15 comprises a housing 15*a* having slidably mounted therein a valve spool 15*b* which has one end face acted upon by a spring 15*c* and the other end face located in a chamber 15*d*. The valve spool 15*b* is formed with a groove 15*e* forming lands on both sides thereof and further with a land 15*f* which is larger in diameter than these lands. On the other hand, the housing 15*a* is formed with an oil passageway 70 which is constantly open to the groove 15*e* and which is in communication with the other input port of the shuttle valve 61. The housing 15*a* is further formed with a drain port 15*g* and is in communication with the oil passageway 40, the drain port and the oil passageway being arranged so that one of them is permitted to open when the other of them is completely closed during movement of the valve spool 15*b*. The housing 15*a* is still further formed with an oil passageway 71 which is open in alignment with the oil passageway 70 and which is in communication with the chamber 15*d*. A chamber 15*h* having the spring 15*c* mounted therein is in communication with the throttle pressure circuit 48 through an oil passageway 72.

With the front clutch pressure reducing valve 15 thus arranged, the valve spool 15*b* is urged by the spring 15*c* to move toward a lowered position isolating the oil passageway 70 from the drain port 15*g* as indicated by the right half of the valve spool. When, therefore, the line pressure is directed through the oil passageways 31 and 32 under the control of the 2—3 shift valve 4 as will be described later, the line pressure is directed through the oil passageway 70 and the shuttle valve 61 to the front clutch 104. Since, however, the line pressure thus conducted is restricted in the orifice 73 in the oil passageway 31, the pressure developed in the front clutch increases initially at a limited rate and thereafter at a gradually increasing rate. Such a pressure is also fed through the oil passageway 71 to the chamber 15*d* and causes the valve spool 15*b* to move upwardly in the drawing. On the other hand, there is developed in the chamber 15*h* a throttle pressure which is directed from the throttle pressure circuit 48 by way of the oil passageway 72 and which is proportioanl to the throttle valve opening degree (see the plot a—b—c—d in Fig. 3). In cooperation with the force of the spring 15*c*, the throttle pressure causes the valve spool 15*d* to move downwardly in the drawing until the valve spool 15*b* rests in a position in which the downward force thus applied thereto is equalized with the force urging the pool upwardly. When the pressure being supplied to the front clutch 104 reaches a predetermined value after the pressure has appeared, the valve spool 15*b* is moved upwardly into the position having the oil passageway 70 isolated from the oil passageway 40 and brought into communication with the drain port 15*g* as indicated by the left half of the spool and is balanced in the particular position. For this reason, the oil pressure supplied to the front clutch cannot be increased beyond the above-mentioned predetermined value. Because, however, of the fact that the throttle pressure is directed into the chamber 15*h* and lends itself to the control of the pressure to be supplied to the front clutch, the pressure supplied to the front clutch increases as the throttle valve opening degree increases, as will be seen from the plot i—j shown in Fig. 3. Under kickdown conditions, on the other hand, the throttle pressure rises to the level of the line pressure (see the plot b—c—d in Fig. 3) as previously noted with the result that the oil pressure supplied to the front clutch also varies with the throttle valve opening degree as indicated by the plot j—k—l in Fig. 3.

The operation of the gear shift control circuit provided with the apparatus thus constructed and arranged in accordance with the present invention as has been hereinbefore described will be explained in the following.

In the first place, the pressure regulator valve 1 is supplied with the pump pressure directed to the chamber 1*d* from the oil pump 115, the modified throttle pressure directed to the chamber 1*j* from the pressure modifier valve 7 and the line pressure directed to the chamber 1*e* from the port 2*b* of the manual valve 2 only when the D range, II range of I range position is selected, having the valve spool 1*b* urged downwardly in the drawing. Into the chamber 1*g* is directed the modified throttle pressure which, in cooperation with the force of the spring 1*a*, urges the valve spool 1*b* upwardly in the drawing. The valve spool 1*b* is held in a position in which the forces thus exerted thereon are balanced in the opposite directions, thereby developing in the circuit 16 a line pressure which is determined by such a position of the valve spool. The line pressure thus developed is constantly directed through the circuit 16 into the corresponding port of the manual valve 2. When the P range, R range or N range position is selected, the chamber 1*e* of the pressure regulaıor valve 1 is drained off through the port 2*b* of the manual valve 2 with the result that the line pressure to be developed in the circuit 16 under such conditions is made higher than that achieved under any of the D range, II range and I range conditions because of the fact that there is no force effective to urge the valve spool 1*b* to move downwardly in the absence of the line pressure in the chamber 1*e* of the pressure regulator valve 1.

When the driver of the vehicle moves the manual valve 2 from the N range position to the D range position, the line pressure circuit 16 communicates with the port 2*b* so that the line pressure in the port 2*b* is directed on the one hand through the oil passageway 56 to the throttle modulator valve 10 and on the other hand through the oil pasageways 22 and 29 to the rear clutch 105. The line pressure directed through the oil passageway 56 to the throttle modulator valve 10 is modulated into the previously mentioned modulated throttle pressure by means of the particular valve and is delivered from the oil passageway 25. The line pressure being passed through the oil passageway 29 is restricted by the orifice 74 on its way to the rear clutch 105 and is fed to the rear clutch 105 initially at a limited rate and thereafter at a gradually increasing rate. The pressure thus fed to the rear clutch is also directed through the oil passageway 63 to the chamber 12*d* of the accumulator 12 and causes the stepped piston 12*b* to move downwardly toward the large-diameter side against the force of the spring 12*f*. By virtue of this, the oil pressure supplied to the rear clutch is augmented slowly so that the rear clutch 105 is enabled to couple softly without producing shocks encountered with the selection of the D range position from the N range position. The rear clutch being thus coupled, the automatic transmission enables the vehicle to start with the first gear ratio.

The line pressure passed through the port 2*b* of the manual valve 2 to the oil passageway 22 is also directed to the governor valves 113 and 114, which deliver to the circuit 23 a governor pressure which corresponds to vehicle speed, as previously described. The port 2*b* of the manual valve 2 being constantly open to the line pressure circuit 16 and thus allowing the line pressure to extend to the oil passageway 22 throughout the forward drive range conditions (D), (II) and (I), the governor pressure is delivered into the governor pressure circuit 23 when the manual valve 2 is in any of these positions.

When the vehicle speed reaches a certain value after the vehicle has started, the governor pressure varying with the vehicle speed and directed to the chamber 3*e* of the 1—2 shift valve 3 overcomes the downward force which the spring 3*d* exerts on the valve spools 3*b* and 3*c* in the positions indicated by the right halves of the spools and the downward force which the modulated throttle pressure directed from the circuit 25 acts on the differential pressure acting area between the lands 3*m* and 3*n*, causing the valve spools 3*b* and 3*c* to move upwardly from the positions indicated by the right halves thereof. When the land 3*m* is moved beyond the rib 3*r* during such movement of the valve spools, the chamber formed between the lands 3*m* and 3*n* is brought into communication with the drain port 3*s* and at the same time the modulated throttle pressure directed from the circuit 25 acts on the differential pressure acting area between the loads 3*m* and 3*n*, eliminating the downward force which has been exerted on the valve spool 3*c* and thereby causing the valve spools 3*b* and 3*c* to move instantaneously into the positions indicated by the left halves of the spools. As a consequence, the oil passageway 30 branched from the oil passageway 29 is in communication with the oil passageway 31 so that the line pressure which has been directed into the oil passageway 29 as previously described is passed through the oil passageway 30 and the 1—2 shift valve 3 to the oil passageway 31. The line pressure is thereafter fed to the servo apply chamber 106′*a* of the band servo 106′ by way of the oil passageway 42 and is restricted by the orifice 75 on its way toward the servo apply chamber with the result that the servo apply pressure increases initially at a restricted rate and thereafter at a gradually increasing rate. The servo apply pressure is directed by way of the oil passageway 64 to the chamber 12*e* of the accumulator 12 so that the stepped piston 12*b* which has been moved into the lowered position thereof as previously described is moved back by the servo apply pressure which acts on the piston in cooperation with the force of the spring 12*f*. This causes the servo apply pressure to rise slowly and enables the band servo 106′ to actuate the second brake 106 slowly. The second brake being thus actuated, a shift is made from the first gear ratio to the second gear ratio in the automatic power transmission having the rear clutch 105 maintained in the coupled condition. The gear shift shocks to be produced during the shifting condition are alleviated by the above described motion of the accumulator 12.

As the vehicle speed further increases under conditions in which the vehicle is operating with the second gear ratio, the governor pressure corresponding the vehicle speed and directed through the circuit 23 to the chamber 4*e* of the 2—3 shift valve 4 overcomes the downward force which the spring 4*d* exerts on the valve spools 4*b* and 4*c* and the downward force which the modulated throttle pressure directed from the circuits 25 and 38 acts on the differential pressure acting area between the land 4*h* and the land 4*i*, thereby causing the valve spools 4*b* and 4*c* to move upwardly from the positions indicated by the right halves of the spools. When the land 4*h* is moved beyond the rib 4*k* during such movement of the valve spools, the chamber formed between the lands 4*h* and 4*i* communicates with the drain port 4*q* for eliminating the downward force which has been exerted by the modulated throttle pressure, thereby causing the valve spools 4*b* and 4*c* to instantaneously move upwardly into the positions indicated by the left halves of the spools. As a consequence, communication is provided between the oil passageway 31 and the oil passageway 32 with the result that the line pressure which has been directed into the oil passageway 31 as previously described is passed through the 2—3 shift valve 4 and the oil passageway 31 to the oil passageway 40. The line pressure thus passed to the oil passageway 40 is extended through the passageways 6*e* and 6*g* of the line pressure booster valve 6 to the chamber 6*f* of the valve for causing the valve spool 6*b* to move rightwardly in the drawing from the position indicated by the upper half to the position indicated by the lower half of the valve spool and is further directed to the chamber 13*d* of the 2—3 timing valve 13 and the corresponding port of the front clutch pressure reducing valve 15. Since, in this instance, the line pressure to be directed to the oil passageway 40 is restricted by the orifice 73 provided in the oil passageway 31, the pressure passed to the former passageway rises initially at a restricted rate and thereafter at a gradually increasing rate. It therefore follows that the oil pressure directed from the oil passageway 40 to the chamber 13*d* is initially unable to move the valve spool 13*b* against the force of the spring 13*c* and permits the valve spool 13*b* to stay in the position indicated by the right half thereof. Likewise, the pressure which has been directed from the oil passageway 40 to the chamber 15*d* of the front clutch pressure reducing valve 15 by way of the oil passageway 71 is initially unable to overcome the downward force exerted in the valve spool 15*b* by the spring 15*c* and the downward force exerted on the valve spool by the throttle pressure directed from the throttle circuit 48 to the oil passageway 72 and is not capable of moving the valve spool upwardly from the position indicated by the right half thereof. For these reasons, the pressure directed from the oil passageway 40 to the chamber 13*d* is prevented from being extended beyond the chamber with the result that the pressure fed from the oil passageway 40 to the front clutch pressure reducing valve 15 is passed without modification through the oil passageway 70 and the shuttle valve 61 and is fed through the oil passageway 62 to the front clutch 104. As the pressure in the oil passageway 40 rises thereafter and causes the valve spool 15*b* to move upwardly into the position indicated by the left half thereof, the front clutch 104 is supplied not with the line pressure *per se* but with an oil pressure which is produced by reducing the line pressure and modifying the pressure in accordance with the throttle pressure in the oil passageway 72 by the previously described pressure modifying function of the front clutch reducing valve 15 and which is thus substantially proportional to the output energy of the engine. At a point in time when the oil pressure in the oil passageway 40 is increased to a value capable of overcoming the force of the spring 13*c*, the oil pressure causes the valve spool 13*b* to move upwardly from the position indicated by the right half to the position indicated by the left half of the valve spool so that the chamber 13*d* communicates with the oil passageway 65 and as a consequence the pressure which has been directed to the chamber 13*d* by way of the oil passageway 40 is passed to the oil passageway 65. The pressure is thereafter passed through the oil passageway 65 and further through the oil passageway 80 provided with the check valve 68 and is fed by way of the oil passageway 67 to the servo release chamber 106'*b*. When the line pressure is thus supplied to the servo release chamber 106'*b*, the piston of the band servo 106 is moved back toward the servo apply chamber 106'*a* since the pressure acting area of the servo apply chamber 106'*a* is less than that of the servo release chamber 106'*b*. After the pressure supplied to the front clutch 104 is increased beyond a certain value by virtue of the above described operations and the front clutch is enabled to commence the coupling action thereof, the supply of the pressure to the servo release chamber 106'*b* is effected (vis., the second brake 106 is released by the band servo 106') so that the coupling motion of the front clutch is slightly overlapped in time with the actuation of the second brake. Thus, the engine is prevented from racing as would otherwise occur if the front clutch and the second brake were allowed to be inoperative concurrently, while a shift is made from the second gear ratio to the third gear ratio in the automatic transmission mechanism by the actuation of the front clutch 104 and with the rear clutch 105 maintained in the coupled condition as previously noted.

As the vehicle speed rises to a certain value while the vehicle is running with the third gear ratio, the governor pressure directed from the circuit 23 to the chamber 5*d* of the 3—2 downshift valve 5 causes the spool 5*b* of the valve to move upwardly from the position indicated by the left half thereof against the force of the spring 5*c*. If the accelerator pedal is depressed and as a consequence the throttle valve opening degree is increased under such conditions, the modulated throttle pressure corresponding to such a valve opening degree and directed from the gear shift control pressure circuit 25 to the 3—2 downshift valve 5 by way of the oil passageway 39 acts on the differential pressure acting area between the land 5*e* and the land 5*g* and, in cooperation with the spring 5*c*, causes the valve spool 5*b* to move downwardly into the position indicated by the right half of the spool. Communication is now provided between the oil passageways 36 and 39 so that the modulated throttle pressure is fed through the oil passageways 39 and 36 and the shuttle valve 34 to the chamber 4*o* of the 2—3 shift valve 4 and cause the valve spool 4*c* to move downwardly from the position indicated by the left half to the position indicated by the right half thereof against the force resulting from the governor pressure developed in the chamber 4*e*. This causes the oil passageways 31 and 32 to be isolated from each other so that the supply of the line pressure to the oil passageway 32 is interrupted and at the same time the oil passageway 32 communicates with the drain port 4*r* whereby the oil pressure which has been fed to the front clutch 104 and the servo release chamber 106'*b* under the third gear ratio condition is discharged in the manner to be described in the following. As the oil pressure in the chamber 15*d* of the front clutch pressure reducing valve 15 is eliminated and as a consequence the valve spool 15*b* is moved by the force of the spring 15*c* to the position providing communication between the oil passageways 40 and 70 as indicated by the right half of the valve spool, the front clutch pressure is discharged at a comparatively high rate through the oil passageway 62, the shuttle valve 61, the oil passageways 70, 40 and 32 and the drain port 4*r*. Since, on the other hand, there is no oil pressure developed in the chamber 13*d* of the 2—3 timing valve 13, the valve spool 13*b* is moved by the force of the spring 13*c* into the position allowing the oil passageway 65 to be open to the drain port 13*e* as indicated by the right half of the valve spool, the servo release pressure is discharged at a comparatively low rate through the oil passageway 67, the oil passageway 80, the orifice 69, the oil passageway 65 and the drain port 13*e*. When the vehicle speed is reduced to a certain degree under these conditions, the governor pressure corresponding to such a vehicle speed and directed from the circuit 23 to the chamber 14*d* of the 3—2 timing valve 14 through the oil passageway 66 cannot move the valve spool 14*b* upwardly toward the position indicated by the left half thereof against the force of the spring 14*c* so that the valve spool 14*b* is moved downwardly into the position indicated by the right half of the spool, thereby providing communication between the oil passageways 65 and 67. In this instance, the servo release pressure is discharged through the oil passageway 67, the 3—2 timing valve 14, the oil passageway 65 and the drain port 13*e* at a rate which is comparatively higher than the rate to be achieved when the vehicle speed is at a relatively high value as previously described. In this manner, the servo release pressure is discharged, when compared with the discharge rate of the front clutch pressure, at a relatively low rate determined by the flow rate of oil through the orifice 69 under high vehicle speed conditons and at a relatively high rate when the vehicle speed is at a relatively low value. For these reasons, the actuation of the band servo 106′ (and accordingly of the second brake 106 as well) as compared with the timing at which the front clutch 104 is uncoupled is retarded at high vehicle speeds. thereby making it possible to achieve a prolonged neutral interval during which the engine is enabled to increase its output speed to a value to match the vehicle speed while a downshift is being made from the third gear ratio to the second gear ratio without producing gear shift shocks. At low vehicle speed, the amount of retardation in the actuation of the second brake 106 as compared with the timing at which the front clutch 104 is to be uncoupled is reduced so that the amount of retardation provides the period of time which is required for the engine to increase its revolution speed to a level to match the vehicle speed, thereby making it possible to reduce the gear shift shocks to be produced during the above described downshift.

It may be mentioned that, when the vehicle speed and accordingly the governor pressure developed in the chamber 5*d* decreases, the 3—2 downshift valve 5 produces functions similar to those which the valve produces in response to an increase in the throttle valve opening degree as above described, enabling the automatic transmission mechanism to make a downshift from the third gear ratio to the second gear ratio.

As the vehicle speed further decreases, the governor pressure in the chamber 3*e* of the 1—2 shift valve 3 becomes no longer capable of overcoming the force of the spring 3*d* so that the spring causes the valve spools 3*b* and 3*c* to move downwardly from the positions indicated by the left halves to the positions indicated by the right halves in the drawing, thereby isolating the oil passageways 30 and 31 from each other and permitting the oil passageway 31 to communicate with the drain port 3*t*. As a consequence, the line pressure which has been fed to the servo apply chamber 106*a*′ is passed through the check valve 76 in the oil passageway 42 and is discharged through the oil passageway 31 and the drain port 3*t*, thereby causing the band servo 106′ to release the second brake 106. In the friction elements, only the rear clutch 105 is now held in the operative condition so that a downshift is effected from the second gear ratio to the first gear ratio in the automatic transmission mechanism.

When the manual valve 2 is thereafter moved back to the N range position, the port 2*b* is drained off so that the line pressure which has been passed to the rear clutch 105 is discharged through the oil passageway 29, the check valve 77 and the oil passageway 22 and further by way of the port 2*b* of the manual valve 2 and renders all the frictional elements of the automatic transmission mechanism inoperative, producing a neutral condition in which the transmission of power is interrupted.

When the accelerator pedal is fully depressed to produce a kickdown condition while the vehicle is operating with the above described third gear ratio, the plunger 8*d* of the throttle valve 8 is moved into the rightward limit position thereof as previously described so that a kickdown pressure (line pressure) is developed in the circuit 24. The kickdown pressure thus developed is directed on the one hand to the port 3*u* of the 1—2 shift valve 3 and on the other hand to the chamber 4*p* of the 2—3 shift valve 4 through the oil passageway 37. The kickdown pressure directed into the chamber 4*p* acts on the upper and lower pressure acting areas of the land 4*g* and the upper pressure acting area of the land 4*h* of the valve spool 4*b* held in the position indicated by the left half of the spool and, in cooperation with the spring 4*d*, causes the valve spools 4*b* and 4*c* to move downwardly into the positions indicated by the right halves thereof. As a consequence, the 2—3 shift valve 4 effects a downshift from the third gear ratio to the second gear ratio in the automatic transmission mechanism in a manner similar to that which has been described. If the vehicle speed is further reduced, the kickdown pressure which is fed from the circuit 24 to the port 3*u* of the 1—2 shift valve 3 acts on the upper pressure acting area of the land 3*h*, the upper and lower pressure acting areas of the land 3*g* and the lower pressure acting area of the land 3*f* and, in cooperation with the spring 3*d*, causes the valve spools 3*b* and 3*c* to move downwardly from the positions indicated by the left halves to the positions indicated by the right halves of the spools. As a consequence, the 1—2 shift valve 3 effects a downshift from the second gear ratio to the first gear ratio in a manner similar to that which has been described.

When a kickdown condition is produced while the vehicle is operating with the first gear ratio, the line pressure appearing in the circuit 25 as above described acts on the differential pressure acting area between the lands 3*m* and 3*n* of the valve spool 3*c* held in the position indicated by the right half of the spool in the 1—2 shift valve 3 and further on the differential pressure acting area between the lands 4*h* and 4*i* of the valve spool 4*b* held in the position indicated by the right half of the spool in the 2—3 shift valve 4, thereby urging each of these valve spools to move downwardly in the drawing. Furthermore, the kickdown pressure developed in the circuit 24 acts on the differential pressure acting area between the lands 3*g* and 3*h* of the valve spool 3*b* in the position indicated by the right half of the spool in the 1—2 shift valve 3 and further on the land 4*g* of the valve spool 4*b* in the position indicated by the right half of the spool in the 2—3 shift valve 4, thereby urging each of these spools downwardly, Furthermore, the shift valves 3 and 4 are subjected to the downward forces which are exerted by the springs 3*d* and 4*d*, respectively. The above described downward forces thus exerted on the spools of the shift valves 3 and 4 are opposed by the forces resulting from the governor pressure developed in the chambers 3*e* and 4*e* so that, when the vehicle speed becomes such that the governor pressure resulting therefrom overcomes the downward force exerted on the valve spool of the 1—2 shift valve, then the 1—2 shift valve 3 effects an upshift from the first gear ratio to the second gear ratio on the manner previously described and, when the vehicle speed becomes such that the governor pressure resulting therefrom overcomes the downward force exerted on the valve spool of the 2—3 shift valve 4, then the 2—3 shift valve 4 effects an upshift from the second gear ratio to the third gear ratio in the manner previously described. Since, however, the downward forces exerted on the valve spools of shift valves 3 and 4 are larger than the downward forces which are exerted thereon under ordinary throttle valve opening degree conditions, an upshift cannot be effected and accordingly acceleration can be achieved by a large driving power with a low gear ratio before the vehicle speed is increased beyond levels corresponding to the ordinary throttle valve opening degrees.

The operation to be achieved when the II range position is selected under conditions in which the vehicle is operating with the manual valve 2 held in the D range position will now be desribed. During conditions in which the third gear ratio in the D range is in operation, the line pressure directed to the oil passageway 40 is passed through the groove 6*e* and the oil passageway 6*g* to the chamber 6*f* and causes the valve spool 6*b* to move from the position indicated by the upper half to the position indicated by the lower half thereof against the force of the spring 6*c*, whereupon the valve spool is maintained in the latter position by the line pressure which is directed from the oil passageway 41 to the chamber 6*f* by way of the groove 6*e* and the oil passageway 6*g*, When the manual valve 2 is moved to the II range position thereof under these conditions, the line pressure circuit 16 is open to the ports 2*b* and 2*c* so that the line pressure is directed through the port 2*b* to the same places as those previously described and through the port 2*c* to the chamber 4*o* of the 2—3 shift valve 4 by way of the oil passageway 35, shuttle valve 34 and oil passageway 33 for acting on the land 4*m* and thereby causing the valve spool 4*c* to move downwardly from the position indicated by the left half to the position indicated by the right half of the spool in the drawing, the line pressure passed through the latter port being further directed to the oil passageway 45. The 2—3 shift valve 4 is therefore rendered into the same conditions as those attained under the previously described kickdown conditions with the result that the oil pressure which has been directed to the front clutch 104 and the servo release chamber 106′*b* is discharged to effect a downshift from the third gear ratio to the second gear ratio in the automatic transmission mechanism. Since, in this instance, the line pressure fed into the chamber 4*o* maintains the valve spool 4*c* in the lowered position thereof, an upshift to the third gear ratio could not result from an increase in the vehicle speed. The line pressure directed into the oil passageway 45 is passed through the oil passageway 43 and the shuttle valve 46 to the port 7*h* of the pressure modifier valve 7 with the line pressure booster valve 6 held under the above described conditons. The throttle modifier valve 7 is thus operative to deliver in the oil passageway 54 a modified throttle pressure of a peak value (indicated by the plot e—f in Fig. 3) irrespective of the variation in the degree of throttle valve opening by reason of the previously described pressure modifying functions of the valve, the modified throttle pressure being fed to the pressure regulator valve 1. As a consequence, the pressure regulator valve 1 enabled, by reason of the previously described operational functions thereof, to develop in the line pressure circuit 16 a line pressure having its peak value (indicated by the plot n—d in Fig. 3) irrespective of the variation in the degree of throttle valve opening. For this reason, a sufficiently high line pressure can be produced and accordingly the rear clutch 105 and the band servo 106′ are actuated powerfully at low to medium throttle valve opening degrees, thereby assuring sufficient engine braking effect under the II range condition.

When the vehicle speed is reduced to a certain value while the vehicle is running under the II range condition, the valve spool 3*b* of the 1—2 shift valve 3 is moved downwardly from the position indicated by the left half to the position indicated by the right half of the spool under the influence of the spring 3*d* with the result that a downshift is effected from the second gear ratio to the first gear ratio in the automatic transmission mechanism in manners similar to those previously described. Under these conditions, the oil pressure which has been developed in the oil passageway 31 is eliminated and, as a consequence the valve spool 6*b* of the line pressure booster valve 6 is released from a force holding the valve spool in the righthand position indicated by the lower half of the spool and is therefore moved back into the position indicated by the upper half of the spool by the force of the spring 6*c*. As a result, the line pressure in the oil passageway 45 is prevented from being extended beyond the line pressure booster valve 6 and the oil passageway 43 communicates through the oil passageways 44 and 47 with the drain port 9*c* of the throttle failsafe valve 9. To the port 7*h* of the pressure modifier valve 7 is thus supplied the line pressure from the circuit 48 by the switching action of the shuttle valve 46, thereby enabling the pressure modifier valve 7 to feed the modified throttle pressure to the pressure regulator valve 1 by way of the oil passageway 54 as previously described so that the pressure regulator valve 1 is made operative to develop the above mentioned line pressure in the circuit 16.

As the vehicle speed increases thereafter and as a consequence the governor pressure developed in the chamber 3*e* of the 1—2 shift valve 3 moves the shift valve into the upshifting conditions thereof, an upshift is made from the first gear ratio to the second gear ratio in the automatic transmission mechanism in the manner previously described. When the line pressure developed in the oil passageway 31 is directed to the line pressure booster valve 6 by way of the oil passageway 41 under these conditions, the valve spool 6*b* of the valve is maintained in the position indicated by the upper half of the spool and is not caused to move rightwardly. Accordingly, upon shifting into the second gear ratio only when the II range has been selected under the condition wherein the vehicle is running with the third gear ratio or only when the I range has been selected under the condition wherein the vehicle is running with the third gear ratio the latter case being discussed later, the line pressure is maintained at a constant, relatively high value throughout the entire range of the throttle valve opening degree as previously described and enables the second brake to grip the clutch drum with a sufficient force so as to provide an assured engine braking effect under II range conditions. Once the first gear ratio is achieved, however, the line pressure cannot be boosted and accordingly the gear shift shocks are not amplified even when repeated downshifts are made from the second gear ratio to the first gear ratio. When a shift is to be made from the second gear ratio in the D range to the II range or the I range, the second brake is held in a condition gripping the clutch drum so that the clutch capacity thereof may be smaller under engine braking conditions than when the II range is selected under third gear ratio conditions and, for this reason, the line pressure need not be augmented by means of the line pressure booster valve.

When the manual valve 2 is then moved into the I range position thereof, the line pressure circuit 16 is permitted to communicate with not only the ports 2*b* and 2*c* but the port 2*d*. The line pressure passed through the ports 2*b* and 2*c* is directed to the same places as those previously described and the line pressure passed through the port 2*d* is supplied to the manual first range pressure reducing valve 11. There being initially no oil pressure in the chamber 11*d* of the pressure reducing valve 11, the valve spool 11*b* is maintained in the lowered position indicated by the right half of the spool by the force of the spring 11*c*. When, however, the line pressure directed from the oil passageway 59 is passed to the chamber 11*d* and causes the valve spool 11*b* to move upwardly so that the line pressure is partially discharged through the drain port 11*f*, the valve spool assumes a balanced position indicated by the left half of the spool, thereby reducing the line pressure to a value equalized with the force of the spring 11 acting on the valve spool in the particular position. The line pressure directed to the oil passageway 59 is reduced to a constant value and the reduced pressure thus obtained is passed through the oil passageway 58, the shuttle valve 28 and the oil passageway 26 and acts on the land 3*l* of the spool of the 1—2 shift valve 3, exerting a downward force on the valve spool 3*c*. At vehicle speeds at which the downward force is smaller than the upward force resulting from the governor pressure developed in the chamber 3*e*, the valve spools 3*b* and 3*c* are held in the positions indicated by the respective left halves of the spools and maintain the automatic transmission mechanism in the second gear ratio conditions, thereby preventing the engine from overrunning which would otherwise be caused when, for example, the I range is selected while the vehicle is running at a high speed. In this instance, the line pressure is augmented by means of the line pressure booster valve 6 only when the I range is selected to produce the second gear ratio under third gear ratio conditions as previously described in connection with the operation under II range conditions. As the vehicle speed is reduced and as a consequence the upward force resulting from the governor pressure developed in the chamber 3*e* decreases, the valve spool 3*c* is moved downwardly into the position indicated by the right half of the spool by the downward force resulting from the previously mentioned constant, reduced pressure acting on the land 3*l* of the valve spool 3*c*, while the valve spool 3*b* is maintained in the position indicated by the left half thereof and spaced from the valve spool 3*c* by the above mentioned constant reduced oil pressure with the spring 3*d* compressed. Under this condition, the oil passageway 27 which has been open to the drain port 3*s* communicates with the oil passageway 26 and allows the constant reduced oil pressure in the oil passageway 26 to be directed through the oil passageway 27 to the low-and-reverse brake 107 so that the automatic power transmission mechanism is enabled to drive the vehicle under I range conditions while producing an engine braking effect by means of the low-and-reverse brake thus actuated and the rear clutch 105 which is maintained coupled. The manual first range pressure reducing valve 11 is adapted to reduce the line pressure from the oil passageway 59 to a constant value dictated by the force of the spring 11*c* and deliver the reduced oil pressure to the oil passageway 58 so that the shift point for the 1—2 shift valve 3 can be selected to occur at a desired constant vehicle speed under manual first gear ratio conditions for thereby preventing, without any delay, the engine from overrunning throughout the range of throttle valve opening.

When the manual valve 2 is moved from the N range position to the R range position, the line pressure circuit 16 is in communication with the port 2*a* alone. From the port 2*a*, the line pressure is passed through the oil passageway 60 and is directed on one hand through the shuttle valve 28 and the oil passageway 26 to the 1—2 shift valve 3 and further through the oil passageway 27 to the low-and-reverse brake 107 with the valve spools 3*b* and 3*c* held in the positions indicated by the right halves thereof in the absence, in the chamber 3*e*, of a governor pressure which is to be developed only under forward drive gear conditions, and on the other hand through the orifice 78, shuttle valve 61 and oil passageway 62 to the front clutch 104. The line pressure to be passed to the front clutch 104 is restricted by the orifice 78 on its way to the front clutch so that the oil pressure to be developed in the front clutch increases initially at a low rate and thereafter at a gradually increasing rate. The oil pressure to be thus supplied to the front clutch is also directed through a branch passage from the oil passageway 60 to the chamber 12*a* of the accumulator and causes the stepped piston 12*b* to move downwardly against the force of the spring 12*f*. As a consequence, the oil pressure supplied to the front clutch increases slowly and accordingly the front clutch 104 is caused to couple slowly without producing shocks which would otherwise be caused when the manual valve 2 is moved from the N range position to the R range position. The automatic transmission mechanism is thus made operative to drive the vehicle rearwardly with the front clutch 104 coupled and the low and reverse brake 107 actuated.

When the manual valve 2 is moved back into the N range position, the port 2*a* is made open to the drain port so that the line pressure in the front clutch 104 is discharged quickly through the oil passageway 62, the shuttle valve 61, the oil passageway 60, the check valve 79 and the port 2*a* of the manual valve 2 while the line pressure in the low-and-reverse brake 107 is discharged quickly by way of the oil passageway 27, the shuttle valve 28, the oil passageway 60 and the port 2*a* of the manual valve 2, thereby rendering the automatic transmission mechanism into the neutral condition thereof.

As is apparent from the foregoing description, the automatic transmission provided with the gear shift shock reducing apparatus according to the present invention is constructed and arranged in such a manner that the line pressure control signal for the regulator valve 1 is produced not by directly using the throttle pressure proportional to the throttle valve opening degree but by using the modified throttle pressure supplied from the pressure modifier valve 7, viz., an oil pressure proportional to the degree of throttle valve opening within a low to medium range and substantially constant irrespective of the variation in the degree of the throttle valve opening larger than the particular range as indicated by the plot a—e—f in Fig. 3 for enabling the regulator valve 1 to produce a line pressure proportional to the torque output of the engine as indicated by the plot m—n—d in Fig. 3 and that the gear shift control signal is produced not by directly using the throttle pressure but by using the modulated throttle pressure supplied from the throttle modulator valve 10, viz., an oil pressure which is proportional to the degree of throttle valve opening larger than a certain medium value as indicated by the plot g—h in Fig. 3. Further, the above mentioned line pressure is reduced to be proportional to the energy output of the engine by means of the front clutch pressure reducing valve 15 responsive to the throttle pressure which is substantially proportional to the energy output of the engine and the oil pressure thus reduced is supplied to the front clutch 104 arranged to select the third gear ratio. In this manner, according to the present invention, the torque transmission capacity of the front clutch 104 can be set at a value accurately corresponding to the energy output of the engine and the gear shift shocks to be produced during upshifting from the second gear ratio to the third gear ratio can be reduced reliably in spite of the fact that an excessive capacity is usually required for the front clutch to be used also as the frictional element for the selection of the reverse drive gear condition.

Claims

1. A hydraulic control system for an automatic transmission for an automotive vehicle having an engine, the transmission being adapted for shifting between different gear ratios according to a shift pattern, the transmission having a plurality of frictional units for effecting shifting, at least one of which includes driving and driven elements which are frictionally engageable under conditions which at least at certain points along the shift pattern involve a speed differential between the driving and driven elements, the control system comprising:

a fluid source,

means communicating with said fluid source for generating a first fluid pressure which is variable in a first pattern corresponding approximately to the pattern of torque output of the vehicle engine, said first fluid pressure generating means being adapted to supply said first fluid pressure to a frictional unit other than the one frictional unit to operate such other frictional unit,

means for changing said first pattern of variation of said fluid pressure to a second pattern corresponding approximately to a combination of the pattern of engine torque output and the pattern of differential speed between the driving and driven elements of said one frictional unit at the time of engagement of the elements, said changing means generating a second fluid pressure, said first pressure generating means being coupled for communication with said changing means, said changing means being adapted to supply said second fluid pressure to said one frictional unit to operate such one frictional unit,

whereby said second fluid pressure reaching said one frictional unit is sufficient to meet torque reaction requirements and differential speed compensation requirements of the one frictional unit at all shift points and is sufficiently low to avoid substantial shocks in effecting shifting by the one frictional unit.

2. A hydraulic control system for an automatic transmission as claimed in claim 1, wherein the automatic transmission includes:

a pair of planetary gear sets which are operatively coupled together and which provide a plurality of transmission drive conditions including a plurality of forward drive gear ratios, including a lowest drive ratio, and including a reverse drive ratio,

the other frictional unit being a forward drive frictional unit which cooperates with said planetary gear sets to provide a forward drive condition of the transmission in all forward gear ratios,

the one frictional unit being a front clutch, the automatic transmission further including a low-and-reverse friction unit, each frictional unit having a fluid actuator, said front clutch cooperating with said planetary gear sets to establish, when actuated and when the low-and-reverse friction unit is deactuated, a forward drive gear ratio other than the lowest foreward drive gear ratio and establishing, when actuated in combination with the low-and-reverse friction unit, a reverse drive condition, the hydraulic control system further including:

a gear position selector valve communicating with said fluid source and with each of said friction units for selectively supplying pressurized fluid to the actuators of said friction units to contribute to selection of the drive condition of the transmission, said gear position selector valve having a valve member movable between at least a forward drive position and a reverse position,

a pair of passageway branches establishing the communication between the gear position selector valve and the front clutch, one of said branches being connected with said gear position selector valve so as to establish communication with the front clutch when the gear position selector valve member has been moved to the reverse position, the other branch establishing communication between the gear selector valve and the front clutch when the gear position selector valve member has been moved to the forward drive position and other conditions are met for actuating the front clutch,

said changing means being interposed in said other branch between the front clutch and said gear position selector valve,

whereby said reduced fluid pressure is fed to said front clutch only when said gear position selector valve member is in the forward drive position and other conditions are met for actuating the high-and-reverse friction unit.

3. A hydraulic control system as claimed in claim 1, wherein said changing means includes a pressure reducing valve.

4. An automatic transmission for an automotive vehicle having an engine with a throttle which opens in degrees, the transmission being shiftable between a plurality of gear ratios, the transmission comprising:

a friction unit with driving and driven elements which may operate with a speed differential at the time of shifting between gear ratios,

said friction unit including a fluid actuator, said friction unit and its actuating means cooperating to contribute to a shifting between gear ratios,

means for generating an actuating fluid pressure which is approximately linearly proportional to the degree of throttle opening,

a conduit extending from said pressure generating means to said fluid actuator of said friction unit to selectively apply said linearly proportional fluid pressure thereto,

whereby said linearly proportional fluid pressure reaching said friction unit is sufficient to meet torque reaction requirements and differential speed compensation requirements of the friction unit at all shift points and is sufficiently low to avoid substantial shock in effecting shifting by the one friction unit.

5. An automatic transmission as claimed in claim 4, wherein said friction unit is a front clutch and wherein the automatic transmission includes:

a pair of planetary gear sets which are operatively coupled together and which provide a plurality of transmission drive conditions including a plurality of forward drive gear ratios, including a lowest drive ratio, and including a reverse drive ratio,

a forward drive friction unit having a fluid actuator which cooperates with said planetary gear sets to provide a forward drive condition of the transmission in all forward gear ratios,

a low-and-reverse friction unit, having a fluid actuator, said front clutch cooperating with said planetary gear sets to establish, when actuated and when the low-and-reverse friction unit is deactuated, a forward drive gear ratio other than the lowest forward drive gear ratio and to establish, when actuated in combination with the low-and-reverse friction unit, a reverse drive condition;

a gear position selector valve communicating with each of said friction units for selectively supplying pressurized fluid to the actuators of said friction units to contribute to selection of the drive condition of the transmission, said gear position selector valve having a valve member movable between at least a forward drive position and a reverse position,

a pair of passageway branches establishing the communication between the gear position selector valve and the front clutch, one of said branches being connected with said gear position selector valve so as to establish communication with the front clutch when the gear position selector valve member has been moved to the reverse position, the other branch establishing communication between the gear selector valve and the front clutch when the gear position selector valve member has been moved to the forward drive position and other conditions are met for actuating the front clutch,

said means for generating a linearly proportional actuating pressure being interposed in said other branch between the front clutch and said gear position selector valve,

whereby said reduced fluid pressure is fed to said front clutch only when said gear position selector valve member is in the forward drive position and other conditions are met for actuating the front clutch.

6. A hydraulic control system as claimed in claim 4, wherein said means for generating a linearly proportional actuating pressure includes a pressure reducing valve.

7. A hydraulic control system as claimed in claim 4, including means for effecting a kickdown condition in the hydraulic control system of the automatic transmission, said kickdown effecting means providing a downshift tendency at very high degrees of throttle opening,

said kickdown effecting means being operatively coupled with said means for generating a linearly proportional actuating pressure to render said means for generating a linearly proportional actuating pressure ineffective during said kickdown condition.

8. A hydraulic control system as claimed in claim 7, wherein the very high degree of throttle opening for actuating said kickdown effecting means is approximately 7/8 of the full throttle opening.

9. A hydraulic control system substantially as described with reference to, and as illustrated in, the accompanying drawings.

Printed for Her Majesty's Stationery Office by the Courier Press, Leamington Spa, 1980. Published by the Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained.