Hydraulic control system for automatic transmission with shockless 4-3 and 4-2 shifting.
Description

The present invention relates to a hydraulic control system according to the preamble part of Claim 1. The preamble part of Claim 1 is based on US-A-3 656 373. This document discloses a hydraulic control system having a 3-4 shift valve which comprises two ports. The first port communicates with a first friction unit and the second port communicates with a first supply passage and is allowed to communicate with the first port when the second shift valve assumes the downshift position.

However, the known control system suffers from the drawback that it needs an additional 4-3, 4-2 downshift valve to create a time lapse of an optimum length between the disengagement of the friction members required for the low speed ratio. This, of course, results in a relatively complicated arrangement.

It is therefore an object of the present invention to provide a hydraulic control system according to the preamble part of Claim 1 which provides an optimum downshifting from an n+2nd speed ratio, where n is a positive integer, to an n+1st speed ratio and an optimum downshifting from an n+2nd speed ratio to an nth speed ratio being capable of avoiding the provision of additional valves.

The solution of this object is achieved by the features of Claim 1.

With a hydraulic control system according to Claim 1, it can be achieved that without the necessity of additional valves, the engagement timing of the first clutch during 4-3 shifting is determined by the orifice while the engagement timing of the aforementioned clutch during 4-2 shifting is determined by a second orifice. Therefore, the timing of the 4-3 shifting and the timing during the 4-2 shifting can be set at an optimum without relying on a complicated valve structure.

Depending Claim 2 contains an advantageous embodiment of the present invention.

Brief description of the drawings

Figure 1 is a schematic view of a power transmission mechanism of a four speed automatic transmission as illustrated with an engine having a throttle and an accelerator;

Figures 2(a), 2(b) and 2(c), when combined, illustrate an embodiment of a hydraulic control system for the automatic transmission according to the present invention; and

Figure 3 is a diagram illustrating the embodiment shown in Figures 2(a), 2(b) and 2(c) in a simplified manner.

Detailed description of the invention

Referring to Figure 1, there is illustrated a power transmission mechanism of a four forward speed and one reverse speed automatic transmission having an overdrive. This power transmission mechanism comprises an input shaft I operatively connected via a torque converter T/C to an engine output shaft E of an engine which has a throttle which opens in degrees, an output shaft O operatively connected to road wheels, only one being shown, via a final drive, not shown. A first planetary gear set G1 and a second planetary gear set G2 are connected between the input and output shafts I and O. A plurality of fluid operated friction units are provided which are made operative and inoperative for producing a plurality of speed ratios between the input shaft I and output shaft O. The fluid operated friction units include a first clutch C1, a second clutch C2, a third clutch C3, a first brake B1, a second brake B2, and a one-way clutch OWC. The first planetary gear set G1 comprises a sun gear S1, an internal gear R1, a carrier PC1 carrying pinion gears P1 meshing simultaneously both the gears S1 and R1. The planetary gear set G2 comprises a sun gear S2, an internal gear R2 and a carrier PC2 carrying pinion gears P2 meshing simultaneously both gears S2 and R2. The carrier PC1 is connectable via the clutch C1 with the input shaft I, and the sun gear S1 is connectable via the clutch C2 with the input shaft I. The carrier PC1 is connectable via the clutch C3 with the internal gear R2. The sun gear S2 is constantly connected with the input shaft I. The internal gear R1 and carrier PC2 are constantly connected with the output shaft O. The brake B1 is arranged to anchor the carrier PC1. The brake B2 is arranged to anchor the sun gear S1. The one-way clutch OWC is so constructed that it allows forward rotation (i.e., the same rotation as that of the engine output shaft E), but prevents reverse rotation (i.e., the rotation opposite to the forward rotation). Thus, it acts as a brake only during reverse rotation.

In the above mentioned power transmission mechanism, the rotational state of each of the rotary elements (S1, S2, R, R2, PC1, and PC2) of the planetary gear sets G1 and G2 can be varied by actuating selected one or combination of the clutches C1, C2 and C3, brake B1, (one-way clutch OWC) and brake B2, thus varying the revolution speed of the output shaft O relative to that of the input shaft I. The four forward speed ratios and one reverse speed ratio are produced if the clutches C1, C2 and C3 and brakes B1 and B2 are engaged in the manner as shown in the following Table.
<table>
<thead>
<tr>
<th></th>
<th>C1</th>
<th>C2</th>
<th>C3</th>
<th>B1 (DWC)</th>
<th>B2</th>
<th>Gear ratio</th>
<th>$a_1=0.45$</th>
<th>$a_2=0.45$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st speed</td>
<td>o</td>
<td>o</td>
<td>o</td>
<td>1+a2/a2</td>
<td>3.22</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2nd speed</td>
<td>o</td>
<td>o</td>
<td>o</td>
<td>a1+a2/a2(1+a1)</td>
<td>1.38</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3rd speed</td>
<td>o</td>
<td>o</td>
<td>o</td>
<td>1/1+a1</td>
<td>1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4th speed</td>
<td>o</td>
<td>o</td>
<td>o</td>
<td>1/a1</td>
<td>0.69</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Reverse</td>
<td>o</td>
<td>o</td>
<td>o</td>
<td>1/a1</td>
<td>-2.22</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In the above Table, a sign "o" denotes actuation state of the clutch or brake, $a_1$ and $a_2$ respectively denote ratios of number of teeth of the internal gears R1 and R2 to number of teeth of the corresponding sun gears S1 and S2. A gear ratio is a ratio of the revolution number of the input shaft I to that of the output shaft O. What is denoted by the label (OWC) below the brake B1 means that the first speed ratio is produced owing to the action of the one-way clutch OWC even if the brake B1 is not applied. However, in this first speed ratio, it is not possible for the output shaft O to drive the engine (that is, no engine braking is effected).

Referring to Figures 2(a), 2(b) and 2(c), a hydraulic control system for the above power transmission mechanism is described.

This hydraulic control system comprises a regulator valve 2, a manual valve 4 a throttle valve 6, a throttle fail safe valve 8, a throttle modulator valve 10, a pressure modifier valve 12, a cut back valve 14, a line pressure booster valve 16, a governor valve 18, a 1—2 shift valve 20, a 2—3 shift valve 22, a 3—4 shift valve 24, a 2—3 timing valve 26, a 2—3 timing valve 28, a 3—4 timing valve 30, a 3—2 timing valve 32, a first manual range pressure reducing valve 34, a torque converter pressure reducing valve 36, a 1—2 accumulator 40, a 3—4 accumulator 40, and an overdrive inhibitor solenoid 42. These valves are interconnected as shown in Figures 2(a), 2(b) and 2(c), and connected with an oil pump O/P, the torque converter T/C, clutches C1, C2, C3 and brakes B1, B2 as shown. The brake B2 has a servo apply chamber S/A, i.e., an oil pressure chamber designed to apply the brake when pressurized, and a servo release chamber S/R, i.e., an oil pressure chamber designated to release the brake when pressurized. Since the servo release chamber S/R has a larger pressure acting area than a pressure acting area of the servo apply chamber S/A, the brake B2 is released when the pressure is supplied to the servo release chamber S/R irrespective of the supply of oil pressure to the servo apply chamber S/A. The overdrive inhibitor solenoid 42 is electrically connected with an overdrive inhibitor switch SW.

The hydraulic control system of the present application is different from the prior proposed hydraulic control system in that a port 124h of the 3—4 shift valve 24 is connected with a port 122k of the 2—3 shift valve 22 via an oil conduit 452 provided with an orifice 660 and also connected with an oil conduit 412 leading from the manual valve 4 via a branch conduit 454 provided with another orifice 662, and a port 122l of the 2—3 shift valve 22 is connected with the oil conduit 412 as opposed to the prior proposed hydraulic control system wherein the port 124h is connected with the oil conduit 412 alone.

For convenience of understanding the feature of the present invention, the following description proceeds along with Figure 3 wherein the 2—3 shift valve 22 and 3—4 shift valve 24 are illustrated.

As shown in Figure 3, the port 122l of the 2—3 shift valve 22 (a first shift valve) is connected with the oil conduit 412 which is supplied with a line pressure when the manual valve 4 (not shown in Figure 3) assumes D or II or I position thereof. The port 122k of the 2—3 shift valve 22 is connected via the oil conduit 452 with the port 124h (a second port) of the 3—4 shift valve 24 (a second shift valve). The oil conduit 452 is provided with the orifice 660 (a second orifice). The oil conduit 454 is provided with the orifice 662 (a first orifice). The flow sectional area of the orifice 662 is smaller than that of the orifice 660. The port 122k of the 2—3 shift valve 22 is allowed to communicate with the port 122l thereof when the 2—3 shift valve 22 assumes the upshift position thereof (as indicated by the left half thereof as viewed in Figure 3), while they are prevented from communicating with each other when a plug 223 of the 2—3 shift valve 22 assumes the down position thereof (as indicated by the right half thereof as viewed in Figure 3). The port 122f of the 2—3
shift valve 22 is allowed to communicate with a port 122g thereof when the 2—3 shift valve 22 assumes the upshift position thereof. This port 122g is connected via an oil conduit 434 with the clutch C2. The port 124h of the 3—4 shift valve 24 is allowed to communicate with a port 124g thereof (a first port) when the 3—4 shift valve 24 assumes the downshift position thereof. This port 124g is connected via the oil conduit 442 with the clutch C3.

Next, the operation is described.

The operation during 4—3 shifting is described. During operation with the 4th speed ratio, the 2—3 shift valve 22 and the 3—4 shift valve 24 assume the upshift positions thereof, respectively, as indicated by the left halves thereof as viewed in Figure 3. Although not shown in Figure 3, the 1—2 shift valve 20 also assumes the upshift position thereof and thus the line pressure is supplied to the oil conduit 432. The oil pressure in the oil conduit 432 acts in the servo apply chamber S/A and via the ports 122f and 122g of the 2—3 shift valve 22 to the clutch C2. The clutch C3 is connected via the ports 124g and 124h with the drain port. In this state, when the 3—4 shift valve 24 shifts to the downshift position thereof as indicated by the right half thereof as viewed in Figure 3, 4—3 shifting initiates. In this case, the 2—3 shift valve 22 stays in the upshift position thereof as indicated by the left half thereof as viewed in Figure 3. After the 3—4 shift valve 24 has shifted to the downshift position thereof, the port 124g is allowed to communicate with the port 124h. This causes the oil conduit 442 to communicate with the oil conduit 452 and the oil conduit 452 is supplied with the line pressure via the oil conduit 412, ports 1221 and 122k of the 2—3 shift valve 22. The oil conduit 452 is connected via the oil conduit 454 to the oil conduit 412, thus being supplied with the line pressure also via this path. Therefore, the clutch C3 is supplied with the line pressure, establishing the 3rd speed ratio. The oil pressure supplied to the clutch C3 as mentioned above is supplied via the orifice 660 in the oil conduit 452 and the orifice 662 in the oil conduit 454. However, since the flow cross sectional area of the orifice 660 is larger than the cross sectional area of the orifice 662, the engagement of the clutch C3 is controlled in response to the flow of oil passing through the orifice 660 actually. That is, the engagement timing of the clutch C3 is determined by the orifice 660. Because the flow sectional area of the orifice 660 is set large, the clutch C3 is engaged quickly during 4—3 shifting.

Next, the operation of the 4—2 shifting is described. To effect 4—2 shifting, the 3—4 shift valve 24 and the 2—3 shift valve 22 shift simultaneously from the upshift positions thereof as indicated by the left halves as viewed in Figure 3 to the downshift positions as indicated by the right halves thereof as viewed in Figure 3. This causes the port 124g of the 3—4 shift valve 24 to communicate with the port 124h thereof. What is done here is the same as in 4—3 shifting. However, since the 2—3 shift valve 22 also shifts to the downshift position thereof, the communication between the ports 122k and 1221 is prevented. Under this condition, the line pressure is supplied to the port 124h of the 3—4 shift valve 24 from the oil conduit 124h of the 3—4 shift valve 24 from the oil conduit 412 via the oil conduit 454. This line pressure is supplied to the clutch C3 via the port 124g of the 3—4 shift valve and the oil conduit 442. That is, the clutch C3 is engaged by the oil passing through the orifice 662 in the oil conduit 454. Therefore, the engagement timing of the clutch C3 is determined by the flow sectional area of the orifice 662. As described before, the flow sectional area of the orifice 662 is set small as compared to that of the orifice 660, the engagement timing of the clutch C3 is delayed. That is, the engagement timing of the clutch C3 during 4—2 shifting is delayed as compared to that during 4—3 shifting. The reasons why the orifices 660 and 662 are set in the manner mentioned above is to decrease a difference between an engine revolution speed before and that after the shifting by elongating the period of time of neutral state which takes place temporarily in shifting so as to allow the engine to rapidly boost its speed in 4—2 shifting because a difference between an engine revolution speed before and that the 4—2 shifting is large as compared to that during 4—3 shifting. As a result, the optimum shifting performances required for both 4—3 shifting and 4—2 shifting are obtained.

It will now be understood that, according to the construction described above, the engagement timing of the clutch C3 during 4—3 shifting is determined by the orifice 660, while the engagement timing of the clutch C3 during 4—2 shifting is determined by the orifice 662. Therefore, the timing of the 4—3 shifting and that during the 4—2 shifting can be set optimum without relying on a complicated valve structure.

It will therefore be understood that the timings for the two shiftings, i.e., an optimum shifting from the n+2th speed ratio to the n+2th speed ratio and an optimum shifting from the n+2th speed ratio to the nth speed ratio can be set separately.
valve assuming the downshift position thereof during operation with the \( n \)th speed ratio, said first shift valve (22) assuming the upshift position thereof during the \( n+1 \)st speed ratio and the \( n+2 \)nd speed ratio; a second shift valve (24) having a downshift position thereof and an upshift position thereof, said second shift valve (24) assuming the downshift position thereof during operation with the \( n+2 \)nd speed ratio; said second shift valve (24) having a first port (124g) and a second port (124h), said first port (124g) being allowed to communicate with said second port (124h) when second shift valve (24) assumes the downshift position thereof, the first port (124g) communicating with the first friction unit (C3); and means (454) defining a first fluid supply passage having one end communicating with the second port (124h), being characterised in that said means (454) defining a first fluid supply passage has an opposite end communicating with said actuating fluid pressure generating means (O/P); a first orifice device (662) is fluidly disposed in said first fluid supply passage (454); means (452) are provided for providing a second fluid supply passage having one end communicating with a second port (124h) and an opposite end communicating with said actuating fluid pressure generating means (O/P) when said first shift valve (22) assumes the upshift position thereof; and a second orifice device (660) is fluidly disposed in said second fluid supply passage (452).

2. A hydraulic control system as claimed in Claim 1, being characterised in that said second orifice device (660) is larger in a flow sectional area than said first orifice device (662).

Patentansprüche

1. Hydraulisches Steuersystem für ein automatisches Getriebe, das von einen \( n \)-ten Übersetzungsverhältnis in ein \( n+1 \)-tes Übersetzungsverhältnis, ein \( n+2 \)-tes Übersetzungsverhältnis schaltbar ist, wobei \( n \) eine positive ganze Zahl ist, das automatische Getriebe eine erste Reibungseinheit und eine zweite Reibungseinheit (C3, C2) enthält, wobei die erste Reibungseinheit (C3) während des Betriebes im \( n \)-ten Übersetzungsverhältnis in Eingriff und die zweite Reibungseinheit (C2) außer Eingriff ist, die erste und zweite Reibungseinheit (C3, C2) während des Betriebes im \( n+1 \)-ten Übersetzungsverhältnis im Eingriff sind, die erste Reibungseinheit (C3) während des Betriebes im \( n+2 \)-ten Übersetzungsverhältnis außer Eingriff und die zweite Reibungseinheit (C2) im Eingriff ist, wobei das hydraulische System aufweist:

- eine Einrichtung (O/P) zur Erzeugung eines Betätigungsfluiddruckes;
- ein erstes Schaltventil (22), das eine Herounterschaltsstelle und eine Heraufschaltstellung besitzt, wobei das erste Schaltventil seine Herounterschaltsstellung während des Betriebes mit dem \( n \)-ten Übersetzungsverhältnis einnimmt und dieses erste Schaltventil (22) seine Heraufschaltstellung während des Betriebes im \( n+1 \)-ten Übersetzungsverhältnis und im \( n+2 \)-ten Übersetzungsverhältnis einnimmt;
- ein zweites Schaltventil (24), das eine Heroucherschaltsstelle und eine Heraufschaltstellung einnimmt, wobei das zweite Schaltventil (24) seine Heraufschaltstellung während des Betriebes im \( n \)-ten Übersetzungsverhältnis im Eingriff ist und dieses zweite Schaltventil (24) seine Heraufschaaltsstellung während des Betriebes mit dem \( n+1 \)-ten Übersetzungsverhältnis einnimmt;
- wobei das zweite Schaltventil (24) einen ersten Anschluß (124g) und einen zweiten Anschluß (124h) aufweist, wobei der erste Anschluß (124g) mit dem zweiten Anschluß (124h) kommunizierend verbunden ist, wenn das zweite Schaltventil (24) eine Herounterschaltsstellung einnimmt, wobei der erste Anschluß (124g) mit der ersten Reibungseinheit (C3) kommunizierend verbunden ist; und
- eine Einrichtung (454), die einen ersten Fluidzuführungskanal bildet, der ein Ende besitzt, das mit dem zweiten Anschluß (124h) kommunizierend verbunden ist, dadurch gekennzeichnet daß die Einrichtung (454), die einen ersten Fluidzuführungskanal bildet, ein gegenüberliegendes Ende aufweist, das mit der den Betätigungsfluiddruck erzeugenden Einrichtung (O/P) kommunizierend verbunden ist;
- eine erste Blendenvorratrichtung (662) strömungsverbunden im ersten Fluidzuführungskanal (454) angeordnet ist;
- eine Einrichtung (452) vorgesehen ist, um einen zweiten Fluidzuführungskanal zu bilden, der mit einem Ende mit einem zweiten Anschluß (124h) kommunizierend verbunden ist und mit einem entgegengesetzten Ende kommunizierend mit der den Betätigungsfluiddruck erzeugenden Einrichtung (O/P) verbunden ist, wenn das erste Schaltventil (22) seine Heraufschaltstellung einnimmt; und
- eine zweite Blendenvorratrichtung (660) strömungsverbunden in dem zweiten Fluidzuführungskanal (452) angeordnet ist.

2. Hydraulisches Steuersystem nach Anspruch 1, dadurch gekennzeichnet, daß die zweite Blendenvorratrichtung (660) einen größeren Strömungsquerschnitt aufweist als die erste Blendenvorratrichtung (662).

Revendications

1. Système de commande hydraulique pour une transmission automatique qui peut passer à un rapport de nième vitesse, un rapport de \( n+1 \)ième vitesse, un rapport de \( n+2 \)ième vitesse où \( n \) est un
nombre entier positif, la transmission automatique comprenant une première unité à friction et une seconde unité (C3, C2), la première unité à friction (C3) étant engagée et la seconde unité à friction (C2) étant libérée pendant un fonctionnement au nième rapport de vitesse, les premières et secondes unités à friction (C3, C2) étant engagées pendant un fonctionnement au n+1ième rapport de vitesse, la première unité à friction (C3) étant libérée et la seconde unité à friction (C2) étant engagée pendant un fonctionnement au n+2ième rapport de vitesse, le système hydraulique comprenant:

un moyen (O/P) pour produire une pression de fluide d’actionnement;
une première soupape de changement (22) ayant une position de rétrogradation et une position de passage à la vitesse supérieure, ladite première soupape de changement prenant sa position de rétrogradation pendant un fonctionnement au nième rapport de vitesse, ladite première soupape de changement (22) prenant sa position de passage à la vitesse supérieure pendant le n+1ième rapport de vitesse et le n+2ième rapport de vitesse;
une seconde soupape de changement (24) ayant une position de rétrogradation et une position de passage à la vitesse supérieure, ladite seconde soupape de changement (24) prenant sa position de passage à la vitesse supérieure pendant le n+1ième rapport de vitesse et le n+2ième rapport de vitesse;
ladite seconde soupape de changement (24) prenant sa position de passage à la vitesse supérieure pendant un fonctionnement au n+2ième rapport de vitesse;
ladite seconde soupape de changement (24) ayant un premier orifice (124g) et un second orifice (124h), ledit premier orifice (124g) pouvant communiquer avec ledit second orifice (124h) lorsque la seconde soupape de changement (24) prend sa position de rétrogradation, le premier orifice (124g) communiquant avec la première unité à friction (C3); et
un moyen (454) définissant un premier passage d’alimentation de fluide ayant une extrémité communiquant avec le second orifice (124h), caractérisé en ce que ledit moyen (454) définissant un premier passage d’alimentation en fluide a une extrémité opposée communiquant avec ledit moyen générateur de pression de fluide d’actionnement (O/P);
un premier dispositif à trou (662) est disposé de manière fluide dans ledit passage d’alimentation en fluide (454);
des moyens (452) sont prévus pour former un second passage d’alimentation en fluide ayant une extrémité communiquant avec un second orifice (124y) et un extrémité opposée communiquant avec ledit moyen générateur de pression de fluid d’actionnement (O/P) lorsque la première soupape de changement (22) prend sa position de passage à la vitesse supérieure; et
un second dispositif à trou (660) est disposé de manière fluide dans ledit second passage d’alimentation en fluide (452).

2. Système de commande hydraulique selon la revendication 1, caractérisé en ce que ledit second dispositif à trou (660) est plus grand, par son aire en section d’écoulement, qui ledit premier dispositif à trou (662).