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(54) **CVT control system has regime change clutches controlled independently of variator**

(57) A hydraulic control system (70) for a multi-regime CVT includes regime-change clutches which are controlled independently of the variator so that a smooth change of regime may be effected both at synchronous ratio and at transmission ratios other than the synchronous ratio.

The system may comprise two electro-hydraulic pressure control valves 99,100, and respective clutch control circuits 90,97 each containing two electrically operated solenoid valves 106,107 and 109,110 which can be switched to connect each of the regime change clutches for hard-fill or soft-fill as required.

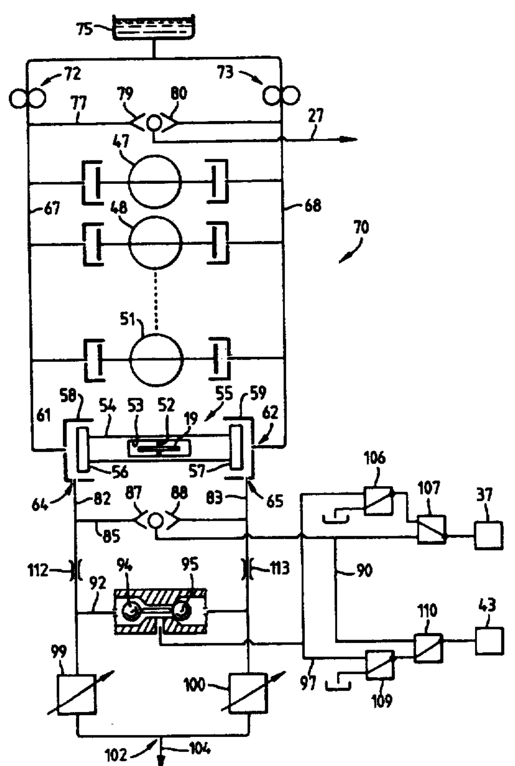

Fig.1

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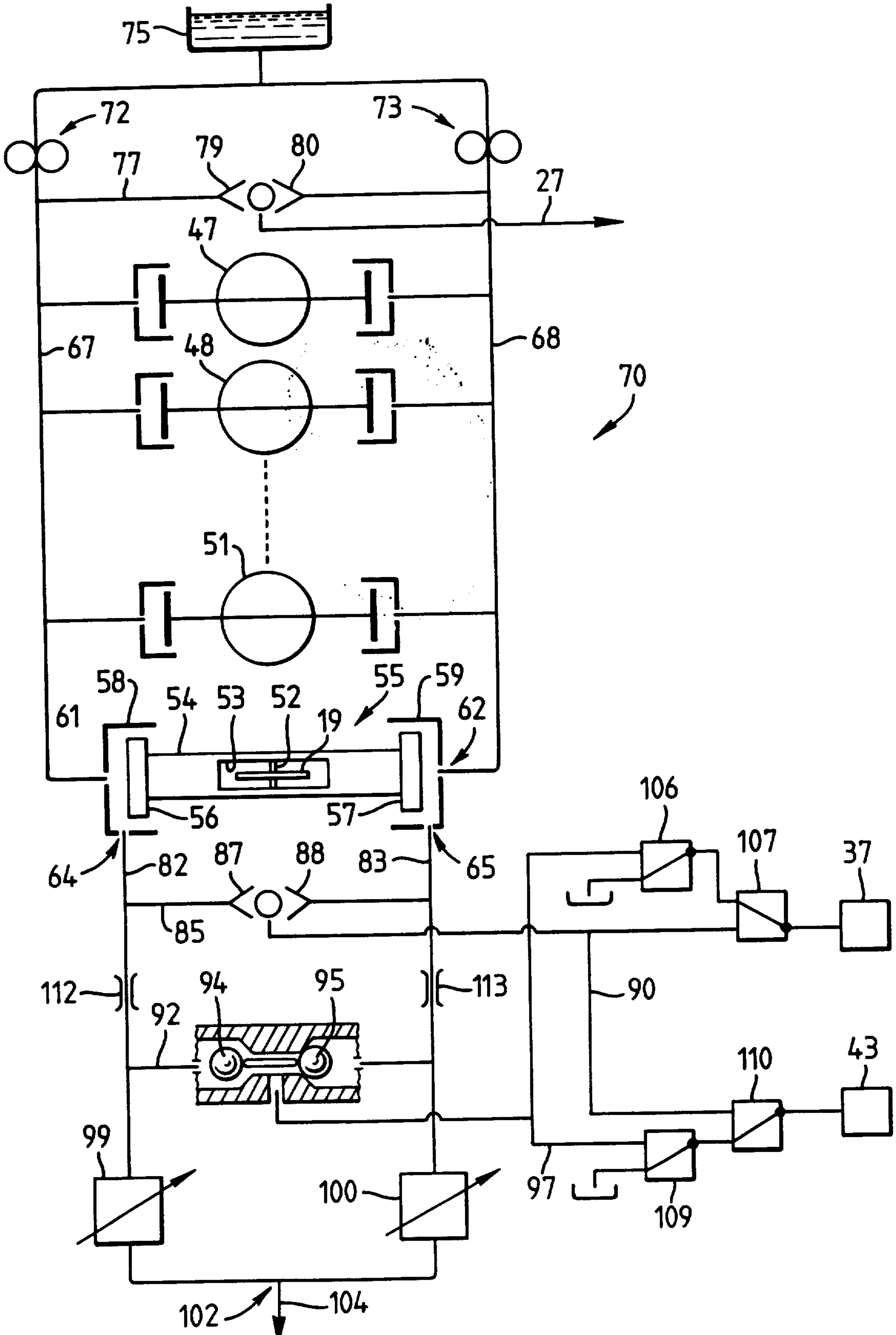
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Fig.1

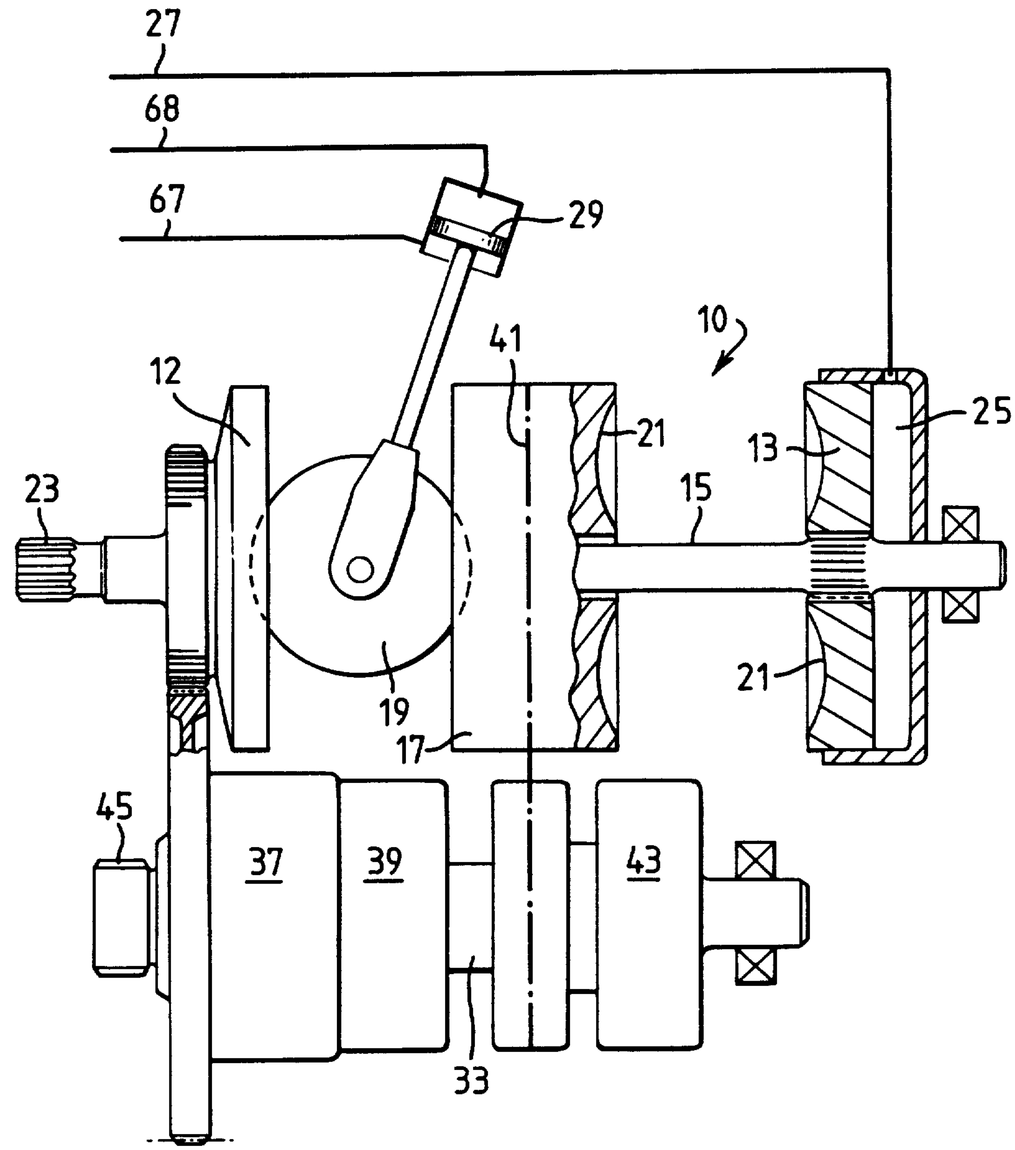

Fig. 2

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CVT CONTROL SYSTEM

The present invention relates to continuously-variable-ratio transmissions (CVTs) *e.g.* for use in an engine-driven vehicle, and to hydraulic control systems for such CVTs.

Transmissions are known which use a variator of the toroidal-race rolling-traction type to give the continuously-variable-ratio performance and clutches to move the transmission between one or other of two operating regimes.

The clutches are designed with interleaving friction plates actuated by hydraulic pistons. To avoid excessive drag when disengaged, the plates are positively separated by 'push off' springs incorporated in the clutch.

When the low regime clutch is engaged, *e.g.* for neutral, reverse and low forward speeds of the vehicle, the drive from the variator to the transmission output shaft is via a mixing epicyclic gear set in which the planet carrier is driven by the input shaft, the variator output disc drives the sun, and the epicyclic annulus is connected with the transmission output shaft.

With the rollers set at their highest speed ratio positions, the influence of the sun wheel dominates to drive the transmission output shaft in reverse. As the rollers are moved away from this position, they pass through a "geared neutral" setting in which the equal and opposite effects of the sun and the planet carrier cancel one another out to give a zero drive effect. Thereafter, as the rollers move to operate the variator at a progressively lower speed ratio, the forward drive from the planet carrier dominates to an increasing extent. Thus when the variator ratio reaches its lower limit (i.e. minimum reverse effect), the sun, planet carrier and annulus all rotate in unison. This results in the two components of the high regime clutch also rotating at the same speeds as one another and the transmission is said to be operating at synchronous ratio.

It will readily be understood that when this latter condition has been established, the low regime clutch can disengage at the same time as (or after) the high regime clutch engages to effect a regime change with minimal slip, shock or wear.

In high regime operation, the transmission output shaft is driven via a fixed ratio chain from the variator output disc and movement of the variator rollers back towards their highest speed ratio positions will enable the transmission to achieve increasingly higher forward speed ratios right up to deep overdrive.

It will be clear that in these known systems, a synchronous change of regime can only occur at one specific transmission ratio since it is only at this point that the oncoming clutch has no relative motion across its elements and can be engaged without substantial risk of shift shock. This is true whether the change is from low regime to high regime as above discussed, or in the contrary sense. However, as in practice the clutches take a finite time to fill and engage, the fill process must be started correspondingly early if the CVT is to provide a smooth continuously varying transmission ratio.

The strategy currently adopted for this purpose in these systems engages the clutch in two phases.

In the first phase, low pressure fluid is used as the transmission approaches synchronous ratio to 'soft-fill' the clutch to a pressure just capable of overcoming the 'push off' springs and closing the friction plates. The finite oil flow required for this purpose is accessed from the lubrication flow at a point downstream of the system control valves to provide a high volume low pressure flow of fluid to the clutch-actuating pistons. Since the low pressure applied to the clutch is not sufficient to create significant clutch capacity, soft fill can be started at any convenient time provided only that the clutch is full as the transmission reaches synchronous ratio.

In the second phase, once the transmission ratio is within acceptable tolerance of synchronous ratio and the clutch has been soft-filled as above described, the hydraulic supply is switched to 'hard fill' the clutch at a sufficiently higher pressure to fully engage the clutch and move the transmission into high regime operation. This second phase of the process requires very little oil flow and so is rapid with minimum disruption to pressure control.

As already stated above, if the sequence has been judged correctly, a perfect regime shift will result, as it will if the clutch fill process has been completed early because the system can then wait for synchronism. However if the fill is late, as sometimes occurs, then the transmission will have already passed synchronous ratio before any action can be taken thereby creating an almost impossible situation for satisfactory regime change. In these latter circumstances, hard-filling of the clutch will occur at a significant transmission ratio error and this results in a noticeable mechanical shock on the change from one transmission regime to the other.

According to the present invention, in a hydraulic control system for a multi-regime CVT of any convenient type, the regime-change clutches are controlled independently of the variator whereby a smooth change of regime may be effected both at synchronous ratio and at transmission ratios other than the synchronous ratio.

Conveniently, regime control is effected using absolute pressures in the hydraulic control system whereas variator control is effected using differential pressures.

In a preferred embodiment of the invention, the hydraulic control system is divided into a first circuit which at the given time is operating at a so-called trailing pressure accessible to soft-fill the clutches and a second circuit which at the given time is operating at a so-called control pressure accessible to hard-fill the clutches.

According to a preferred feature, control electronics are used to manage the regime shift, directly determining the shift rate in the event that regime shift commences at a transmission ratio other than synchronous ratio in which case the transmission ratio is brought to synchronous ratio by virtue of the relevant regime clutch becoming fully operational.

The invention also includes multi-regime CVTs (*i.e.* CVTs with two or more regimes) incorporating the new control system.

An embodiment of the invention will now be described, by way of example only, with reference to the accompanying schematic drawings in which:

Figure 1 shows a circuit diagram for a hydraulic control system in accordance with the present invention; and

Figure 2 shows, in outline form only, the CVT to be controlled by the system of Figure 1.

Referring first to Figure 2, the CVT 8 shown there includes a variator 10 of the toroidal-race rolling-traction type comprising two input discs 12,13 (the latter splined to the variator shaft 15 for limited axial movement along the shaft), a central output disc 17, and two arrays of piston-controlled rollers engaging with the input and output discs to transmit torque between the discs in response to the torque demand on the variator. For simplicity, only one of the six rollers, the so-called master roller 19, is shown in Figure 2.

As is known from GB 2227287, for example, it is important that the roller pistons be aligned so that they are substantially tangential to the centre circle of the imaginary toruses 21 of which the rotor races form part, but with a small inclination known as the "camber" or "caster" angle.

The variator input shaft 15 is powered at one end 23 from the engine (not shown) of the host vehicle while an axial load is imposed on the input disc at the other end of the variator by a pressure chamber 25 filled with high pressure hydraulic fluid from the same line 27 as the control pressure fluid for roller piston 29. The trailing pressure fluid for piston 29 is provided from a relatively low pressure source 31 and it will be noted that the roller responds to the differential pressure across its control piston rather than to the absolute pressure values in lines 27 and 31.

For low regime operation, *e.g.* as described in the introductory portions of this application, the transmission output shaft 33 is powered from the variator input shaft via gearing 35, a low regime clutch 37 and an epicyclic gear set 39 in known fashion.

For high regime operation, shaft 33 is instead powered from the variator output disc 17 via a drive chain 41 and a high regime clutch 43.

Reference numeral 45 indicates the output end of shaft 33, *e.g.* for connection with the host vehicle's differential and road wheels.

The master roller 19 and the five "slave" rollers 47-51 and their associated control pistons and cylinders appear again in Figure 1 as do the two regime clutches 37 and 43, both of which are fitted with push-off springs rated to withstand a plate-closing pressure of up to 3 bar.

As will be seen from Figure 1, the axle 52 of the master roller 19 is mounted in the cavity 53 of the hollow shaft 54 of a double-acting piston 55. This piston is formed with opposed piston heads 56, 57 which are both free to slide under hydraulic load within coaxial cylindrical caps 58, 59 and to rotate about the longitudinal axis of the shaft 54.

In a modification (not shown), item 55 is replaced by a single-headed design of double-acting piston *e.g.* as disclosed, for example, in GB 2227287 and depicted, for convenience only, in the diagrammatic representation of Figure 2.

Returning to the double-ended arrangement of Figure 1, the hydraulic fluid inlets 61, 62 and outlets 64, 65 for the master piston are formed in the end and side walls of the associated cylinder caps 58, 59 and pressure lines 67, 68 ensure that the various slave pistons behave in exactly the same way as master piston 29 so that all six variator rollers are continuously maintained at the same orientation as one another.

Turning now to the hydraulic control system 70, this comprises two independent oil pumps 72, 73 delivering hydraulic fluid from a sump 75 at equal rates of flow to the lines 67, 68 referred to above. A cross-connection 77 between these two lines communicates by way of a 'higher-pressure-wins' arrangement of non-return valves 79 and 80 with the flow line 27 in Figure 2.

The outlets 64, 65 for the master piston end caps 58, 59 feed left-hand and right-hand pressure lines 82, 83. These are interconnected by a cross-connection 85 which communicates by way of a 'higher-pressure-wins' arrangement 87, 88 with the hard-fill circuit 90 for clutches 37, 43. A second cross-connection 92 communicates by a 'lower-pressure-wins' arrangement 94, 95 with the soft-fill circuit 97 for the two clutches and with the trailing side of the roller pistons as already explained with reference to Figure 2.

Reference numerals 99, 100 indicate two electro-hydraulic pressure control valves whose action in controlling the regime shift of the transmission 8 will be discussed below.

Downstream of these two valves, the pressure lines 82, 83 combine at 102 from whence a connection 104 provides back-pressure fluid for general lubrication of the transmission.

Turning now to clutch control circuits 90, 97 it will be noted that each contains two electrically-operated solenoid valves 106, 107 and 109, 110 which can be switched to connect each of the clutches 37, 43 for hard-fill or for soft-fill as required. From the situation illustrated in Figure 1, for example, switching the valve 106 will connect the low regime clutch 37 to the soft-fill circuit 97 whereas switching the valve 107 instead, will connect the low regime clutch 37 to the hard-fill circuit 90. Valves 109 and 110 operate in analogous fashion to valves 106 and 107 but in respect of the high regime clutch 43.

The system is completed by two 1 bar restrictor plates 112, 113 located in lines 82, 83 between the two cross-connection 85, 92.

In operation of the illustrated embodiment, suppose, by way of example only, that initially clutch 37 is operable and clutch 43 is to be brought into operation in place of clutch 37 to effect a regime change. Then, as compared with the situation illustrated in Figure 1 (neither clutch operative), solenoid valve 107 will have been switched so as to connect clutch 37 with the hard fill circuit 90.

Turning now to the pressure control valves 99, 100, in a typical case the pressure control valve 99 initially is receiving zero current and the valve 100 is receiving a ½ amp current. This means that the line pressure immediately upstream of valve 99 will be at roughly back-pressure (2 bar) and the next step is to switch valve 109 to fill the line between the valve 109 and clutch 43 with low pressure oil.

To soft-fill clutch 43 and close the clutch plates in readiness for a regime change, the currents in valves 99, 100 are raised to 0.1 amps and 0.6 amps respectively to increase the adjacent line pressures typically from 2 bar to 3.1 bar (line 82) and from 10 bar to 11.1 bar (line 83). The pressure in line 82 is now sufficient to soft-fill the high-regime clutch 43 at a rate determined by the control current for valves 99, 100. After a set time, typically ½ second, say, the control electronics will assume that soft-filling of the clutch 43 has been completed and the system is held in readiness for the next phase.

This latter is commenced when the control electronics increase the control valve currents at 99, 100 to 1 amp and 1.5 amps respectively, raising the adjacent line pressures to 5 bar (line 82) and 13 bar (line 83). The pressure in line 82 is now sufficient to cause the high-regime clutch to generate capacity and this moves the transmission to the synchronous ratio. The initial part of the regime-change process is completed when the control electronics measures the transmission ratio to be synchronous.

It is to be noted that in all the situations discussed above, the operating currents for the two pressure control valves are increased by exactly the same amounts so that the **difference** in pressure between the two lines 82, 83 remains at 8 bar. This means that the roller control piston settings in the variator are unaffected by what is happening at the regime clutches.

As already described in an earlier portion of the application, bringing both the regime-shift clutches into full operation ensures that the transmission is operating at synchronous ratio and it is at this point that the control electronics has to 'decide' from the information it is receiving about the transmission, the engine speed and the throttle pedal setting whether to return the transmission to low regime operation or to switch over to high regime operation.

A decision to return to low regime operation will simply involve reversing each of the steps discussed above (at all times maintaining the 8 bar pressure difference between lines 82, 83).

It should be noted, however, that if the decision is to change the transmission from one regime to the other, then, when this change of regime occurs, the operating characteristics of the variator will be reversed so that what was the control pressure side of the roller control piston will now be the trailing pressure side and vice versa. This change requires that at the cross-over point of the variator's operational cycle, the "control" and "trailing" pressures lie temporarily at one and the same value.

Thus, if it is confirmed that a change of regime is still appropriate, by maintaining clutch 43 engaged and disengaging clutch 37, the first necessary step is that the valve currents are both raised to a same value, typically 2 amps, so as temporarily to 'cut out' the variator 10 from the transmission before switching valves 106, 107 to the positions illustrated in Figure 1 to disengage clutch 37. During the subsequent release of clutch 37, the current in valve 100 is reduced to zero amps while that in valve 99 is reduced to ½ amp so as to preset the pressure differential across the variator to that required for operation in the next regime.

Further regime changes at either clutch will be carried out in an analogous fashion to those described above.

It is to be noted that when the valves 99, 100 have zero control current and the pressure differential across the variator is zero, the presence of restrictor plates 112, 113 (or their functional equivalent) is important as they act to ensure a one bar pressure difference between the soft-fill and the hard-fill lines. This means that what is at the time the hard-filled clutch can be retained at a sufficiently high pressure to maintain the clutch engaged while what is at the time the soft-filled clutch can be maintained at a sufficiently low pressure to prevent the clutch plates from closing against the opposing action of the clutch push-off spring.

It is further to be noted that in the event of an emergency overload situation, the pressure peaks occurring as a result of the hydraulic end-stop effect in the relevant cylinder cap 58, 59 will be passed on by lines 67, 68 to the other control cylinders and to the variator end load chamber 25. However, as this momentary peaking will not occur in the downstream lines 82, 83 controlling the regime clutches, these latter will remain unaffected and can slip, if necessary, to relieve the excess load on the variator.

CLAIMS

1. A hydraulic control system for a multi-regime CVT of any convenient type, comprising regime-change clutches which are controlled independently of the variator whereby a smooth change of regime may be effected both at synchronous ratio and at transmission ratios other than the synchronous ratio.

2. A control system as claimed in Claim 1 in which regime control is effected using absolute pressures in the hydraulic control system whereas variator control is effected using differential pressures.

3. A control system as claimed in Claim 1 or Claim 2 in which the hydraulic control system is divided into a first circuit which at the given time is operating at a so-called trailing pressure accessible to soft-fill the clutches and a second circuit which at the given time is operating at a so-called control pressure accessible to hard-fill the clutches.

4. A control system as claimed in any preceding claim in which control electronics are used to manage the regime shift, directly determining the shift rate in the event that regime shift commences at a transmission ratio other than synchronous ratio in which case the transmission ratio is brought to synchronous ratio by virtue of the relevant regime clutch becoming fully operational.

5. A multi-regime CVT incorporating a control system as claimed in any preceding claim.

6. A control system or multi-regime CVT substantially as hereinbefore described with reference to and/or as illustrated in the accompanying drawings.

The Patent Office

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Application No: GB 9608147.6
Claims searched: 1 to 6

Examiner: Jim Calvert
Date of search: 11 July 1996

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.O): F2D(DCA,DCG)

Int Cl (Ed.6): F16H 61/38, F16H 37/00

Other: Online: WPI

Documents considered to be relevant:

Category	Identity of document and relevant passage	Relevant to claims
X	EP0507329A2 (MAZDA) See pages 7 to 9	1,4,5

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category.	P	Document published on or after the declared priority date but before the filing date of this invention.
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