US007275455B2

(12) United States Patent
Kennedy

(10) Patent No.: US 7,275,455 B2
(45) Date of Patent: Oct. 2, 2007

(54) AUTOMATIC GEAR SYSTEM

(75) Inventor: **Ian Duncan Kennedy**, Whitnash (GB)

(73) Assignee: **Luk Lamellen und Kupplungsbau Beteiligungs KG**, Buehl (DE)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 229 days.

(21) Appl. No.: **10/489,571**

(22) PCT Filed: **Sep. 10, 2002**

(86) PCT No.: **PCT/DE02/03349**

§ 371 (c)(1), (2), (4) Date: **Oct. 6, 2004**

(87) PCT Pub. No.: **WO03/025434**

PCT Pub. Date: **Mar. 27, 2003**

(65) **Prior Publication Data**

US 2005/0043139 A1 Feb. 24, 2005

(30) **Foreign Application Priority Data**

Sep. 12, 2001 (GB) 0121923.7

(51) **Int. Cl.**

F16H 59/60	(2006.01)
F16H 59/00	(2006.01)
F16H 61/00	(2006.01)
F16H 63/00	(2006.01)

(52) **U.S. Cl.** .... **74/335**; 74/336 R; 477/97

(58) **Field of Classification Search** .... 74/335, 74/336 R; 477/70, 76, 97, 98

See application file for complete search history.

(56) **References Cited**

U.S. PATENT DOCUMENTS

4,856,360	A *	8/1989	Yoshimura et al.	74/335
5,138,905	A *	8/1992	Kouno	477/80
5,899,829	A	5/1999	Salecker et al.	477/78
5,910,068	A	6/1999	Krauss et al.	477/109
5,925,086	A *	7/1999	Sakai et al.	701/66
6,070,117	A *	5/2000	Yamada et al.	701/50
6,145,398	A *	11/2000	Bansbach et al.	74/335
6,164,149	A *	12/2000	Ohmori et al.	74/336 R
6,295,884	B1	10/2001	Miyake et al.	74/335
6,393,928	B1	5/2002	Watanabe	74/336
6,524,215	B1 *	2/2003	Schmidt	477/3
2002/0046618	A1 *	4/2002	Yamashita et al.	74/335

FOREIGN PATENT DOCUMENTS

DE	195 04 847 A1	9/1995
DE	196 27 980 A1	1/1997
DE	197 34 023 A1	2/1998
DE	199 61 117 A1	7/2001
EP	0 038 113 B1	10/1981
EP	0 043 660 B1	1/1982

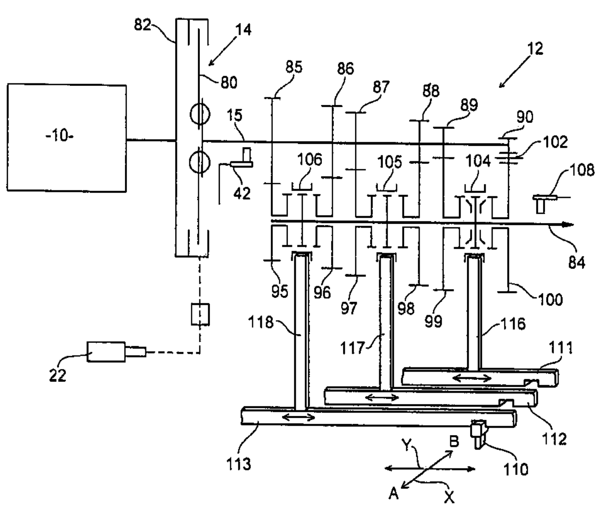
(Continued)

Primary Examiner—Tisha Lewis
(74) *Attorney, Agent, or Firm*—Simpson & Simpson, PLLC

(57) **ABSTRACT**

The present invention broadly comprises a method for the control of a gear engagement mechanism of an automated multiple-ratio transmission having at least one drive shaft and at least one output shaft, including assigning a synchronizer to at least one gear step, applying a load to the synchronizer in order to engage or disengage a gear step of the transmission, and varying the load that is applied to the synchronizer as a function of the service life of the transmission system.

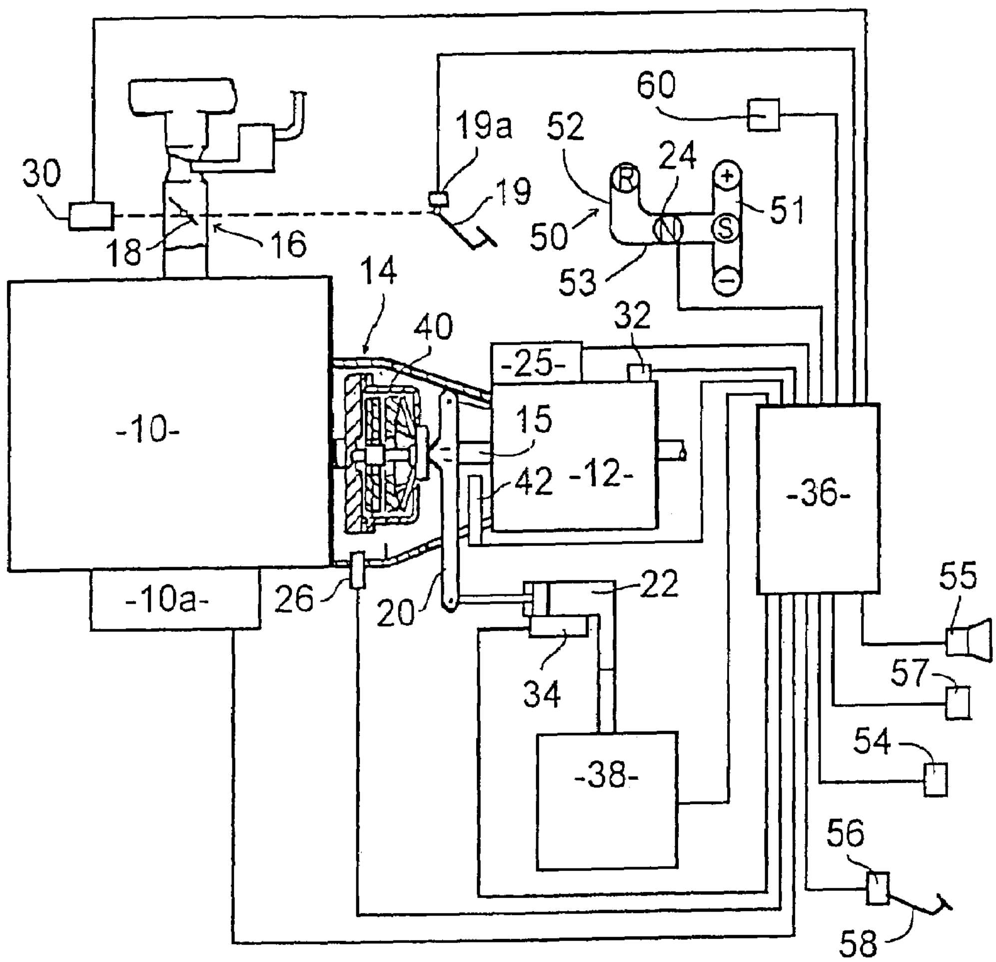
3 Claims, 4 Drawing Sheets



FOREIGN PATENT DOCUMENTS

EP	0 059 035 B1	9/1982
EP	0 101 220 B1	2/1984
EP	0 477 564 B1	4/1992
EP	0 976 955 B1	2/2000
EP	1 081 417 B1	3/2001
GB	2 308 413 A	6/1997
GB	2 354 295 A	3/2001
GB	2 354 296 A	3/2001
GB	2 358 443 A	7/2001
GB	2 360 557 A	9/2001
GB	2 369 654 A	6/2002
GB	2 369 656 A	6/2002
GB	2 369 657 A	6/2002
GB	2 370 611 A	7/2002
GB	2 370 865 A	7/2002
WO	WO92/13208	8/1992
WO	WO97/05410	2/1997
WO	WO97/10456	3/1997

* cited by examiner



60
19a
52
24
R
+
19
51
30
S
N
50
18
16
53
14
40
32
-25-
-10-
15
-12-
-36-
42
-10a-
26
20
22
55
57
34
54
-38-
56
58

Fig 1.

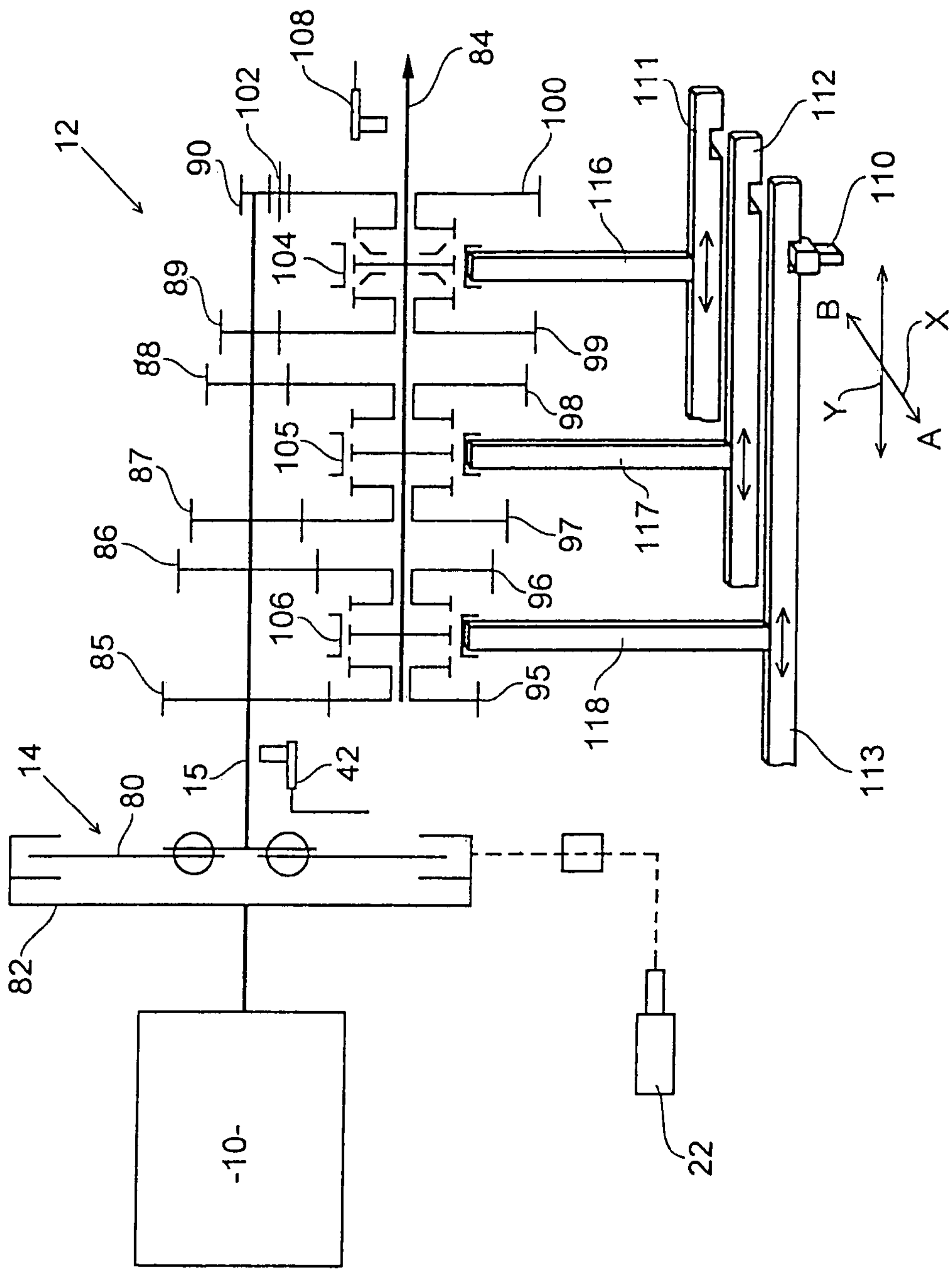
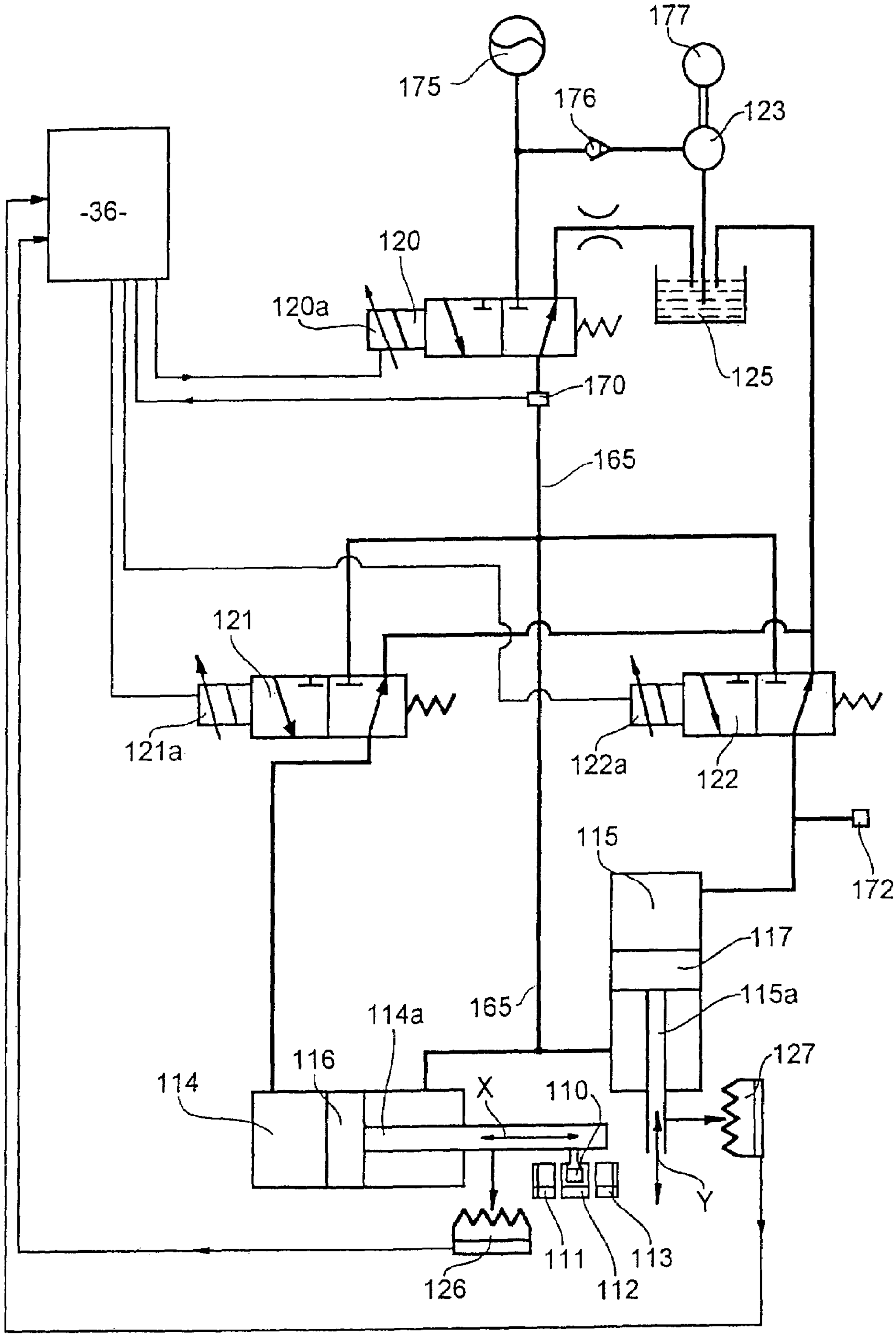

12
14
80
82
15
42
-10-
22
84
85
86
87
88
89
90
95
96
97
98
99
100
102
104
105
106
108
110
111
112
113
116
117
118
A
B
X
Y

Fig 2.



175
177
176
123
-36-
120
120a
125
170
165
121
121a
122a
122
115
172
117
115a
165
114a
116
114
X
110
127
Y
111
112
113
126

Fig 3.

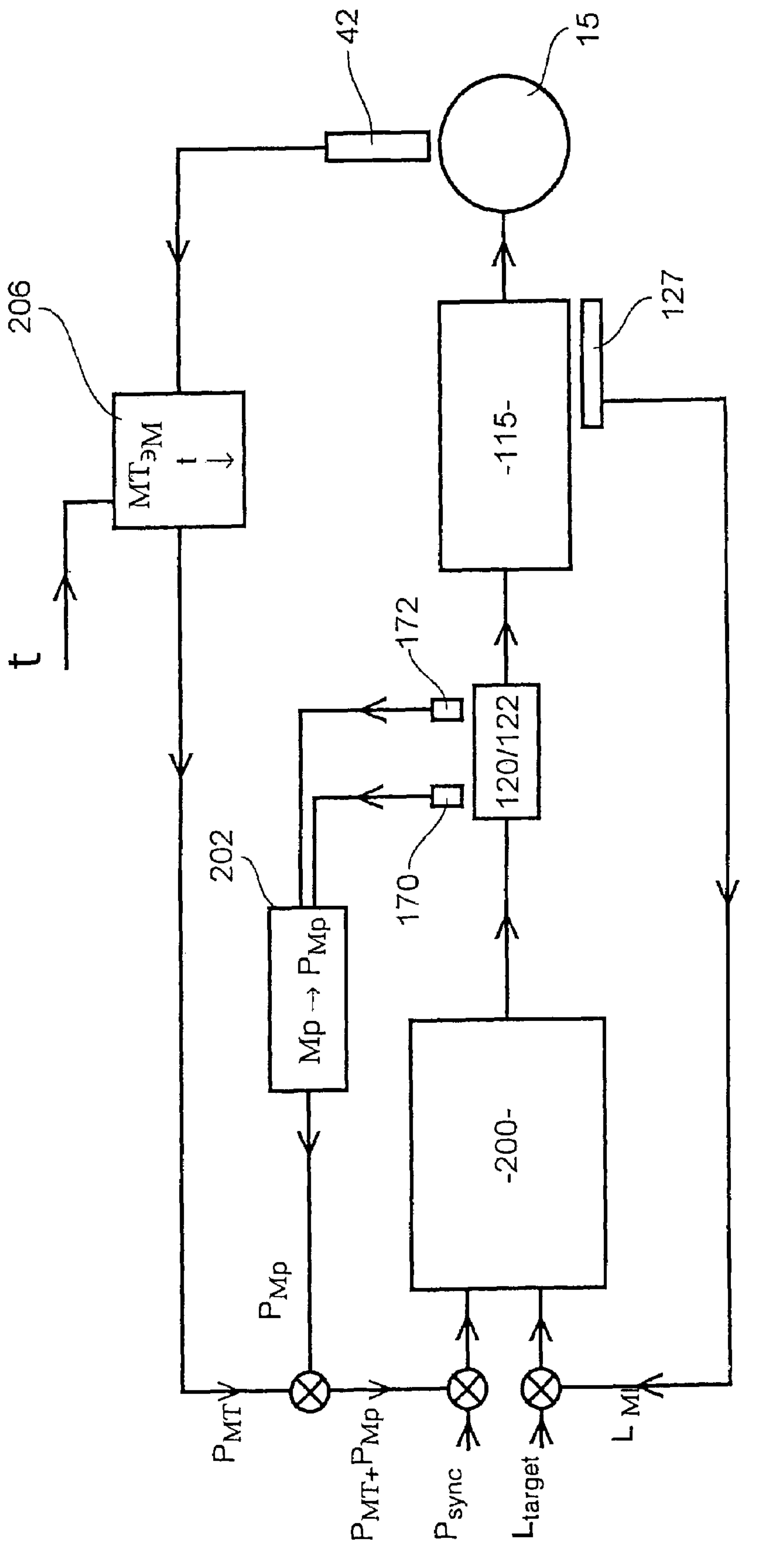

42
15
206
$MT_{\ni M}$
t
127
-115-
172
120/122
170
202
$Mp \to P_{Mp}$
-200-
P_{Mp}
P_{MT}
$P_{MT}+P_{Mp}$
P_{sync}
L_{target}
L_{MI}

Fig 4.

AUTOMATIC GEAR SYSTEM

CROSS REFERENCE TO RELATED APPLICATIONS

The present application is the U.S. national stage application pursuant to 35 U.S.C. §371 of International Application No. PCT/DE02/03349, filed Sep. 10, 2002, which application claims benefit of British Patent Application No. 0121923.7, filed Sep. 12, 2001, which applications are incorporated herein by reference.

BACKGROUND OF THE INVENTION

The present invention relates to automated transmission systems, and in particular automated manual, semi-automatic and fully automatic transmission systems that use a multiple-ratio transmission that has synchronizers on the gear steps.

In automated transmission systems that use multiple-ratio transmissions in which the engagement of the gears is regulated by synchronizers, as are disclosed in, for example, GB2308413; GB2354296; GB2354295; GB2358443; GB0105186.1; GB0029453.8; GB0026423.4; GB0025848.3; GB0025847.5; GB0029454.6; GB0025000.1; GB0024999.5; GB0026178.4; GB0027640.2; GB0028310.1; GB0031624.0; GB103312.5, whereby reference will be made exclusively to these disclosures and their content is incorporated in the disclosure content of the present application; the synchronizer is used to brake or accelerate the drive shaft of the transmission until the speed for the intended target gear is reached. Therefore, the synchronizer, as far as its action is concerned, is a friction device that is capable of transmitting a limited torque. The required torque is determined by the inertia of the drive shaft and by the time that is available for the complete synchronization. The synchronization time is a function of the magnitude of the required speed change. Traditionally, the synchronizers that are associated with the various gears have different performance capabilities that take into account the intended braking torque, the frequency of use and the average speed change.

It is desirable to regulate the torsional force that acts on the synchronizers in order to ensure that the torque acting on them, and consequently the synchronizer wear, do not become too great. Previously, this was achieved by regulating the force that acts on the synchronizer via an actuating mechanism, the actuating force corresponding to an optimal torsional force for the synchronizer that is empirically predetermined on the basis of steady load tests by the transmission manufacturer.

The torsional forces that act on the synchronizers change along with the speed change and also with the temperature changes of the transmission oil. Adjustments may be implemented via the actuation force that acts on the synchronizers in order to incorporate such changes. However, such changes can only be approximations and, while the regulation of the actuation force on the synchronizers is improved, it cannot be guaranteed that the torsional forces corresponding to the predetermined maximum synchronization forces prescribed by the manufacturer are not exceeded. Moreover, in order to implement such adjustments, additional sensors are required to record changes in the temperature and speed differentials.

Furthermore, other factors, such as the wear of the components mounted on the transmission drive shaft, and especially a driven clutch plate, and changes in the friction coefficient of the synchronizer that are not foreseeable and are not precisely applicable for the adjustments also have effects on the torque that is applied by the synchronizer and on the rate of change of the drive shaft speed.

SUMMARY OF THE INVENTION

According to one aspect of the present invention, a method for regulating a gear engagement mechanism of an automated multiple-ratio transmission, which has a drive shaft and at least one synchronizer, includes the application of a load to the synchronizer in order to engage or disengage a gear step of the transmission, whereby the load that acts on the synchronizer is calculated as a function of a transmission parameter that changes over time, for example, the rate of change of the drive shaft speed. For this purpose, the monitoring of the drive shaft speed directly via a transmission input speed sensor or the indirect determination of it via additional sensors and measured values, for example of gear information, at least one speed sensor, one clutch travel sensor, one engine speed sensor and/or one throttle flap position as well as the evaluation of stored and/or adaptable value groups of an engine characteristic map may be advantageous.

Because the change of a transmission parameter, for example, the speed of the drive shaft, may permit conclusions about the magnitude of the torque that acts on the synchronizer, the load that acts on the synchronizer may be regulated in this manner to ensure that the maximum synchronizer force that is prescribed by the manufacturer is not exceeded.

According to a preferred embodiment of the invention, an automated transmission system comprises a multiple-ratio transmission having a synchronizer by means of which the gear steps are connected in a driven manner and, between a drive shaft and a transmission output shaft, means for applying a load to the synchronizer in order to engage and disengage the gear steps, a closed-loop control circuit being provided in order to regulate the load that acts on the synchronizer as a function of the rate of change of the transmission parameter.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention is now described, merely for purposes of example, with reference to the accompanying drawings, of which:

FIG. **1** shows a semi-automatic transmission system that uses a hydraulic circuit according to the present invention;

FIG. **2** schematically represents a multiple-ratio transmission that is used in the transmission system in FIG. **1**;

FIG. **3** schematically shows a hydraulic control system for the gear engagement mechanism of the transmission that is represented in FIG. **2**; and

FIG. **4** schematically shows a control circuit for the hydraulic control system that is represented in FIG. **3**.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

FIG. **1** of the accompanying drawings shows an engine **10** having a starter and an associated starter circuit **1**a that is coupled by main drive friction clutch **14** via a transmission drive shaft **15** to a multiple-ratio, synchronized transmission **12** of the type having a countershaft. The engine is supplied with fuel via a throttle flap **16**, the throttle flap comprising a throttle valve **18** that is actuated by gas pedal **19**. The invention is equally applicable for gas or diesel engines with electronic or mechanical fuel injection.

Clutch **14** is actuated by a clutch fork **20** that is actuated in turn by a sequence-controlled hydraulic cylinder **22** that is under the control of a clutch actuator control mechanism **38**.

A gear selection lever **24** operates in a shift gate **50** having two legs **51** and **52** that are linked by a transverse track **53** that extends between the end of leg **52** and a position between the ends of leg **51**. Shift gate **50** defines five positions; "R" at the end of leg **52**; "N" in the center between the ends of transverse track **53**; "S" at the junction point of leg **51**. In leg **51** lever **24** is aligned at the starting point with central position "S". Position "N" of selection lever **24** corresponds to neutral; "R" corresponds to the selection of the reverse gear; "S" corresponds to the selection of a forward drive gear; the short-term movement of the lever into the "+" position represents a command that causes the transmission to shift up one gear step; and the short-term movement of gear lever **24** into the "−" position represents a command that causes the transmission to shift down one gear step.

The positions of lever **24** are recorded by a series of sensors, for example microswitches or optic sensors, that are disposed around shift gate **50**. The signals of the sensors are supplied to an electronic control unit **36**.

An output of the control unit **36** controls a gear engagement mechanism **25** that engages the gear steps of transmission **12** according to the movement of selection lever **24** by the driver of the vehicle.

In addition to the signals of gear selection lever **24**, control unit **36** receives signals from:

Sensor **19***a*, which indicates the degree to which gas pedal **19** is depressed;

Sensor **30**, which indicates the degree to which throttle valve **18** is opened;

Sensor **26**, which indicates the engine speed;

Sensor **42**, which indicates the speed of the driven clutch plate;

Sensor **34**, which indicates the position of the sequence-controlled clutch cylinder;

and

Sensor **32**, which indicates the selected shift step.

Control unit **36** uses the signals of these sensors to control the actuation of clutch **14** while starting out from the idle position and during gear change, as is described in patent descriptions of EP0038113, EP0043660, EP 0059035, EP0101220 and WO92/13208, whose contents are expressly included in the disclosure content of the present invention.

In addition to the aforementioned sensors, control unit **36** also receives signals from a vehicle speedometer **57**, from ignition lock **54** and from brake light switch **56**, which is a part of the main braking system of the vehicle, for example, foot brake **58**.

A buzzer **52** is connected to control unit **36** in order to warn the driver of the vehicle or indicate to him if certain operating states arise. In addition to or instead of buzzer **52**, a flashing warning light or other indication means may be used. A gear indicator **60** is also provided in order to indicate the selected gear.

As illustrated in FIG. **2**, an output shaft **84** from transmission **12** is installed parallel to drive shaft **15**. A series of gears **85** to **90** is mounted on drive shaft **15** in order to turn with it. A matching series of gears **95** to **100** are attached to the output shaft **84** for rotation in relation to said first series. Gears **85** to **89** and gears **95** to **99** are disposed in mating pairs and thus harmonized in their sizes in order to provide the different gear steps; gears **85** and **95** provide a fifth gear; gears **86** and **86** provide a fourth gear; gears **87** and **97** provide a third gear; gears **88** and **98** provide a second gear; gears **89** and **99** provide a first gear. Another gear engages between gears **90** and **100** to reverse the direction of rotation and provides a reverse gear.

Synchronizers **104**, **105**, **106** are located between gears **99** and **100**; **97** and **98**; and **95** and **96**, respectively. The axial movement of synchronizer **104** to the left, as represented in FIG. **2**, thereby engages gear **99** in a rotary connection with output shaft **84**, while the axial movement of synchronizer **104** to the right engages gear **100** in a rotary connection with output shaft **84**. In a similar manner, the axial movement of synchronizer **105** engages **97** or gear **98**, depending on the selection, in a rotary connection with output shaft **84**, and the axial movement of synchronizer **106** engages gear **95** or gear **96**, depending on the selection, in a rotary connection with output shaft **84**.

Speed sensors **42** and **108** are available to monitor the speed of transmission drive shaft **15** and transmission output shaft **84** and provide signals that are proportional to said speeds and are sent to electronic control unit **36**.

The gear engagement mechanism **25** comprises three shift rails **111**, **112**, **113** that are mounted parallel to each other for the axial directional movement. Each shift rail **111**, **112**, **113** is connected via a shift fork **116**, **117**, **118** to another synchronizer **104**, **105**, **106** so the movement of the shift rails **111**, **112**, **113** in an axial direction causes the engagement of one of the gears that are assigned to the corresponding synchronizer **104**, **105**, **106**, and the axial movement of the shift rails **111**, **112**, **113** in the opposite axial direction causes the engagement of the other gear that is assigned to corresponding synchronizer **104**, **105**, **106**.

A shifter element **110** is mounted for a movement in a first direction X transverse to the axes of shift rails **111**, **112**, **113** and mounted in a second direction Y, which represents an axial movement with respect to shift rails **111**, **112** and **113**. The shifter element **110** can therefore be moved in a selected direction X along a neutral level A-B so that it can be latched with one of the shift rails **111**, **112**, **113** and can then engage in a selected shift rail. Shifter element **110** can then be moved in a shift direction Y in order to move the engaged shift rail **111**, **112**, **113** axially in one of the two directions in order to engage one of the gears that is connected to it.

As represented in FIG. **3**, the shifter element **110** is moveable in the selected direction X via a first actuator **114**, which is actuated by fluid pressure, along neutral level A-B in order to align shifter element **110** with one of shift rails **111**, **112** or **113** and thereby select a pair of gears that is connected with synchronizer **104**, **105**, **106**, which in turn is controlled by shift rail **111**, **112** or **113**. The shifter element **110** can be moved in the shift direction Y via a second actuator **115** actuated by fluid pressure in order to move the shift guide **111**, **112** or **113** axially in one of the two directions in order to engage one of the gears that is connected with synchronizer **104**, **105**, **106**, which in turn is controlled by shift rail **111**, **112** or **113**.

Actuators **114** and **115** each include a double-acting plunger piston having an engaging piston rod **114***a* or **115***a*, respectively, that is connected with shifter element **110**. Engagement rod **114***a* is situated on one side of a piston **116** of actuator **114** so the effective surface on the rod-end side of piston **116** is smaller than on the opposite, head side. The same is true for actuator **115**; engagement rod **115***a* sits on one side of a piston so the effective surface on the rod-end side of piston **117** is smaller than on the opposite, head side.

The supply of hydraulic fluid to the rod-end side and head-end side of pistons **116** and **117** is controlled by three valves **120**, **121** and **122** that are actuated by solenoids. Valve **120** is an on-off valve that connects the rod sides of pistons **116** and **117** and proportional flow control valves **121** and **122** to the pressurized fluid from a hydraulic source that comprises a pump **123** and an accumulator **175** or connects them to compensating tank **125** via main fluid supply line **165** for the discharge. Valves **121** and **122** can connect head-end sides of pistons **116** and **117** to main control valve **120** or compensating tank **125**. A pressure measurement converter **170** is provided in main supply line **165** between main control valve **120** and valves **121** and **122** and rod-end sides **116** and **117**, and a pressure measurement converter **172** is provided between the head-end side of piston **117** and valve **122**. Valves **120**, **121** and **122** are controlled by electronic control unit **36** in order to apply the appropriate pressure to the opposing sides of pistons **116** and **117** in order to control the movement of pistons **116** and **117** and of attached engagement rods **114***a* and **115***a* in order to select and engage the required gear in the manner that is disclosed in WO97/05410.

Potentiometers **126** and **127** are connected with engagement rods **114***a* or **115***a* in order to provide a signal that marks the position of the attached engagement rod. The signals of potentiometer **126** and **127** are supplied to control unit **36** in order to provide an indication of the position of engagement rods **114***a* and **115***a* for each of the gear steps of the transmission and also in order to mark the position of engagement rod **115***a* if shifter element **110** is located in neutral level A-B from FIG. **2**. In this way, the transmission system can be calibrated so that predetermined position signals that come from potentiometers **126** and **127** correspond to the engagement of each of the gears of transmission **15**.

Measured results of potentiometer **126** and **127** may then be used by a closed-loop control system to control valves **120**, **121** and **122** in order to move engagement rods **114***a* and **115***a* to the predetermined positions to engage the desired gear.

In a first position of main control valve **120**, high pressure accumulator **175** is connected via main supply line **165** to the rod-end side of pistons **116** and **117** of actuator **114** or **115**, respectively, and valves **121** and **122**.

Valves **121** and **122** can then be controlled in order to selectively connect head-end sides of pistons **116** and **117** to accumulator **175** or compensating tank **125**. If the head-end sides of piston **116** and **117** are connected to accumulator **175**, the pressure that acts on both sides of pistons **116** and **117** is the same, but because of the difference in the effective surfaces of the rod-end sides and the head-end sides of pistons **116** and **117**, pistons **116** and **117** are moved to the right or downward as illustrated in FIG. **3**.

If the head-end sides of pistons **116** and **117** are connected via valves **121** or **122** to compensating tank **125**, the pressure difference around pistons **116** and **117** moves piston **116** and **117** to the left and to the right, respectively, as illustrated in FIG. **3**, the fluid from the head end side of pistons **116** and **117** being pressed out into compensating tank **125**. The movement of shifter element **110** can therefore be controlled in order to bring about the engagement of the desired gear by controlling main valve **120** in order to connect main supply line **165** to accumulator **175** and by operating valves **121** and **122** in order to move shifter element **110** accordingly in directions X and Y, respectively.

When the gear change is completed, main control valve **120** is switched in such a manner as to connect main supply line **165** to compensating tank **125**, which allows the return of the fluid from the rod-end side of piston **116** and **117** and the head-end side of pistons **116** and **117** via valves **121** and **122**.

In order to maximize the service life of synchronizers **104**, **105**, **106**, it is desirable to limit the loads that act on them. This may be achieved by skillful operation of valves **120**, **122**, in that valves **120**, **122** are rapidly switched between a position in which the head-end side of piston **117** is connected to compensating tank **125** and a position in which the head-end side of piston **117** is connected to accumulator **175** in order to precisely control the pressure difference around piston **117**. Other means, such as a proportional pressure control valve or the method that is disclosed in GB 0024999 or GB 0025000, whose content is expressly included in the disclosure content of the present invention, may alternatively be used in order to control the pressure difference around piston **117** and as a result the load that acts on the synchronizer that is assigned to the selected gear.

As illustrated in FIG. **4**, if a gear change is initiated when clutch **14** has been disengaged, control unit **36** generates a position signal L_{AB}, which corresponds to the position on neutral plane A-B, and a pressure signal P_0, which corresponds to the force required to move synchronizer **104**, **105**, **106**, which is assigned to the gear engaged at this moment, from the gear engagement position to the neutral level. Position signal L_{AB} and pressure signal P_0 are fed to a controller **200**, which generates a signal in order to excite valves **120** and **122** in order to apply a pressure differential around actuator **115** in order to move synchronizer **104**, **105**, **106** the gear engaged at this time is disengaged. Because the drive and output shafts **42**, **84** of transmission **12** are synchronized, the loads on synchronizer **104**, **105**, **106** are negligible. So the pressure difference on sliding actuator **115** can be maximized in order to achieve a rapid disengagement of the gear engaged at this point in time. Position sensor **127**, which is attached to actuator **115**, indicates a status signal to control unit **36**, which is used to modify position signal L_{AB} by adding or subtracting a correction factor L_{MI}, which is calculated as a function of the difference between the actual position of positioning member **115** and the position that is required for reaching the neutral level A-B so the shifter element **110** is moved swiftly and precisely to neutral level A-B.

Valves **120** and **121** are then skillfully operated in order to move actuator **114** to a position in which shifter element **110** ends up engaged with the shift rail **111**, **112**, **113** that is assigned to the target gear. On the other hand, this may also be done in that the maximum pressure difference around actuator **114** is used, the position signal that is generated by control unit **36** is in turn modified by a closed-loop feedback control circuit, the signal of position sensor **126** being used in the manner described above.

Control unit **36** then generates a position signal L_{target} and pressure signal P_{target}, which correspond to the target gear. Pressure signal P_{target} may initially be high in order to bring the synchronizer rapidly into frictional engagement, but pressure signal P_{target} is then weakened to P_{Sync} in order to limit the torsional force that acts on synchronizer **104**, **105**, **106**. P_{Sync} is a predetermined value that is a function of the type of synchronizer **104**, **105**, **106** and is typically different from one synchronizer **104**, **105**, **106** to the other. Pressure measurement converters **170** and **172** measure the pressure difference around sliding actuator **115** and provide a signal P_{Mp} that is used to correct pressure signal P_{Sync} via a closed-loop feedback circuit **202**. Position sensor **127** also provides a closed-loop feedback control of position signal L_{target} in the manner described above.

Moreover, according to the present invention, the signal that comes from drive shaft speed sensor **42** is processed in order to provide a signal that corresponds to the rate of change of the speed of drive shaft **15**.

This is converted into another pressure signal P_{MT}, which corresponds to the difference between the actual rate of change of the drive shaft speed and a predetermined optimal rate of change of the drive shaft speed for a given synchronizer **104**, **105**, **106** and which brings about a further correction of the pressure signal via a closed-loop feedback control **206**. In this way, the actual application of the actuator load on the synchronizer is monitored so as to ensure that the synchronization forces that are prescribed by the manufacturer are not continually exceeded. Because the rate of change of the speed of drive shaft **15** reflects the torque that acts on synchronizer **104**, **105**, **106**, closed-loop feedback control **206** also corrects deviations that occur due to changes in the moment of inertia of drive shaft **15**, changes in the friction of drive shaft **15** on the basis of the viscosity of the transmission oil and changes in the speed differential between drive shaft **15** and output shaft **84**.

As depicted in FIG. **4**, pressure correction signal P_{MT} may be adjusted exclusively or in conjunction with the change in the speed of the transmission input shaft in relation to the temperature of the transmission oil. Another input t is applied to the closed-loop feedback circuit, the input t being an estimate of the transmission oil temperature, which is based on the coolant and intake air temperatures of the motor vehicle. This estimated value of the transmission oil temperature can be used to modify the correction factor P_{MT} in order to include the transmission oil temperature and thus its viscosity. For precise comparisons, it may be advantageous to use a sensor for the transmission oil temperature.

According to an alternative embodiment of the invention, the drive shaft feedback loop, instead of being used to form a pressure correction factor P_{MT}, may be used to incrementally increase or decrease the pressure signal, this being a function of whether the rate of change in the speed of the drive shaft is above or below the predetermined optimal value. That means that the load which is applied to the synchronization unit is increased by an increment if the rate of change of the speed of drive shaft **15** is above the pre-determined optimum value.

According to another embodiment of the present invention, the rate of change in the speed of drive shaft **15** is used instead of the pressure signals that come from measurement converters **170** and **172**, whereupon there is no need for measurement converter **172** and feedback loop **202**.

The rate of change in the correction of the drive speed may be used according to the present invention in order to determine independent individual settings for different operating strategies or operating modes, for example for the use of a lower synchronization speed in the saver switching operating mode, in order to extend the service life of the transmission; meanwhile, greater synchronization speeds are permitted for a better output in sport operating mode. Furthermore, different parameters may be used over the service life of the transmission. For example, the optimum rate of change of the drive shaft speeds can be raised after the gear starts running.

While the embodiments that have been described above use hydraulic actuators, the present invention is equally applicable for other forms of actuators, for example, for pneumatic or electric actuators, as described, for example, in DE19504847, WO97/10456 or DE19734023, their disclosures being included herewith for purposes of referral.

Of course, instead of a hydraulic drive, also actuation means for the engagement and disengagement of gears and electric drives for the engagement and disengagement of the clutch are usable that may be controlled or regulated accordingly by specification of the electrical energy. Thus, for example, for specification of the load on the at least one synchronizer, the current, the voltage, a pulse width or the like of the actuation element may be controlled or regulated. Appropriate systems for a transmission actuation of this type are known, for example from DE 196 27 980 A1.

The patent claims submitted along with the application are formulation proposals without prejudice for the attainment of ongoing patent protection. The applicant reserves the right to claim additional feature combinations that so far are only disclosed in the description and/or drawings.

References used in the dependent claims point to the further formation of the subject matter of the main claim by the features of each dependent claim; they are not to be understood as renunciation of the attainment of a separate, concrete protection for the feature combinations of the referred dependent claims.

Because the subject matter of the dependent claims may form separate and independent inventions with respect to the state of the art on the priority date, the applicant reserves the right to make them the subject matter of independent claims or separation statements. They may furthermore also include independent inventions that have a configuration independent of the subject matters of prior dependent claims.

The exemplary embodiments are not to be understood as a limitation of the invention. Rather, numerous amendments and modifications are possible within the context of the present publication, especially such variants, elements and combinations and/or materials as may be inferred by one skilled in the art with regard to the resolution of the problem using, for example, a combination or modification of individual features or elements or methodological steps that are described in connection with the general description and embodiments as well as the claims and that are contained in the drawings and, using combinable features, lead to a new subject matter or to new methodological steps or methodological sequences, even if they pertain to manufacturing, testing and operating method.

What is claimed is:

1. A method for the control of a gear engagement mechanism of an automated multiple-ratio transmission having at least one drive shaft and at least one output shaft, comprising:

assigning a synchronizer to at least one gear step;

applying a load to the synchronizer in order to engage or disengage a gear step of the transmission; and,

varying the load that is applied to the synchronizer as a function of a transmission parameter, in which the load that acts on the synchronizer is controlled as a function of the difference of a to-be-detected value of the transmission parameter and a pre-determined optimal value of the transmission parameter.

2. The method as described in claim **1**, in which the load that acts on the synchronizer is incrementally controlled, the load being increased by an increment if the rate of the transmission parameter is below the predetermined optimal value, and the load being reduced by an increment if the rate of the transmission parameter is above the predetermined optimal value.

3. The method as described in claim **1**, in which the predetermined optimal rate changes from a first synchronizer to at least one other provided synchronizer.

* * * * *