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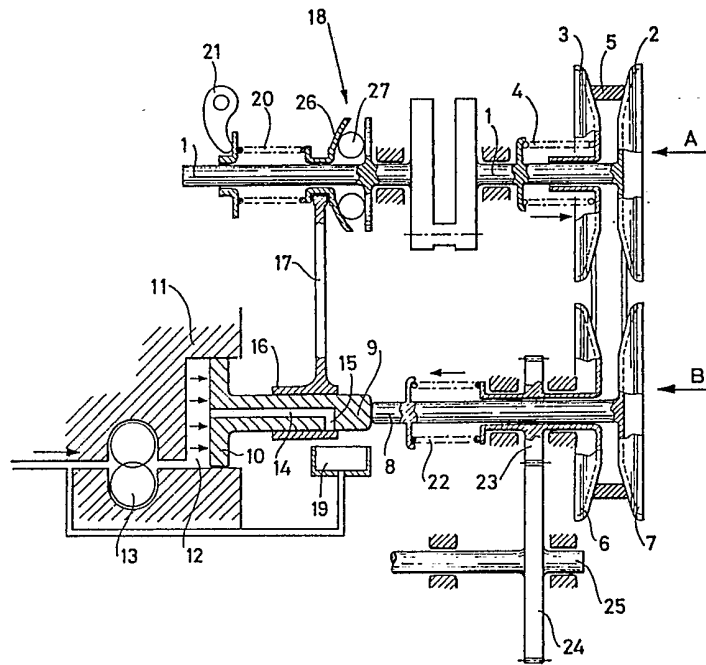
(58) Field of search  
**F2D**

(54) **Variable-transmission-ratio  
mechanical drive with V-belt and  
expandible sheaves**

(57) Driving sheave A has half-sheave 2 fixed to the driving shaft 1 and half-sheave 3 biased by spring 4. Driven sheave B has half-sheave 6 which is axially fixed and half-sheave 7 biased towards 6 by spring 22 but driven in the opposite sense by piston 10. Liquid is pumped to chamber 12 by pump 13 and escapes through orifice 15 controlled by sleeve 16. The position of sheave 16 is determined by governor 18 on the drive shaft 1 which is biased by spring 20 whose pressure is adjusted by the accelerator control lever through lever 21. Thus only one of the movable half-sheaves is positioned by a servo-mechanism in accord with the speed and load of the engine.

In an alternative construction the servo mechanism coupled to governor 18 acts only on the half-sheave 3 (reversed in position).

**Fig.1**



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Fig. 1

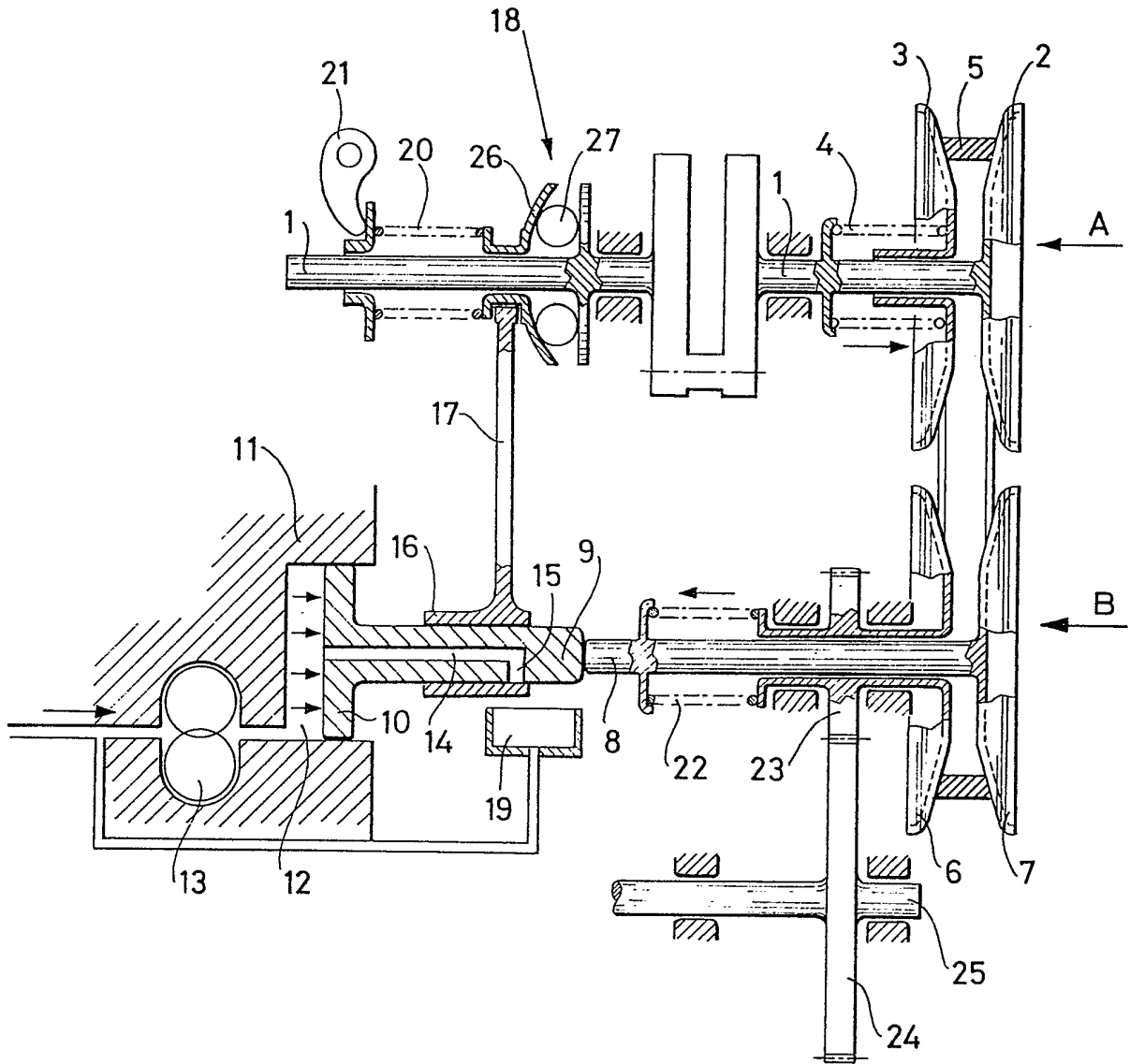
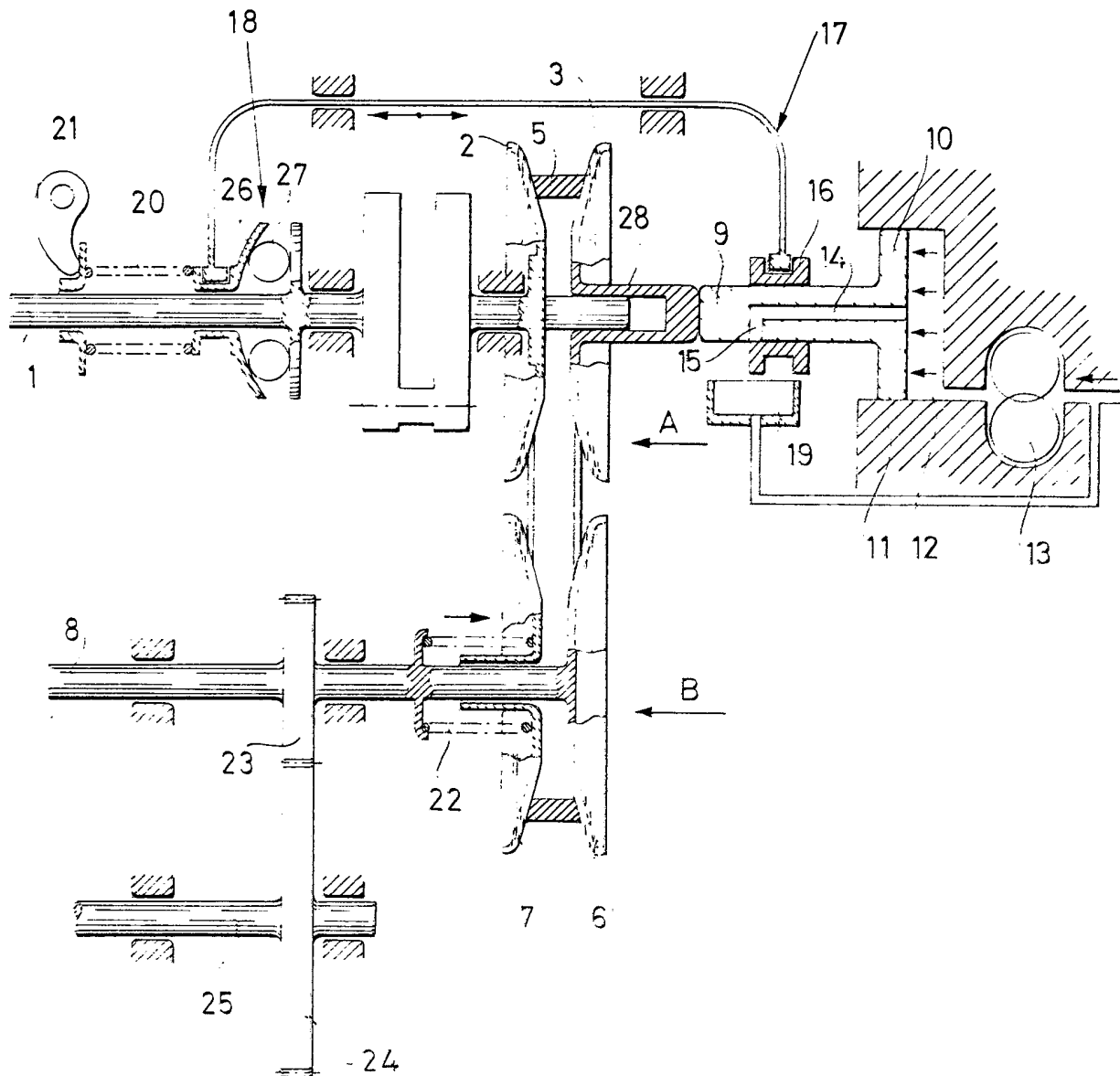


Fig. 2



## SPECIFICATION

**Variable-transmission-ratio mechanical drive with v-belt and expansible sheaves**

5 This invention relates to transmissions in which the transmission ratio varies continuously and are of the type with V-belt and expansible sheaves, more particularly with the ratio which automatically varies as a function of both the speed and the load.

10 It is known that such transmissions can vary continuously the transmission ratio by adopting comparatively simple constructional arrangements.

15 In the practical applications on motor bicycles and such-like, the variation of the transmission ratio is achieved automatically, generally by a device of the centrifugal class which acts directly upon the driving sheave and causes the latter to become closed against the bias of the spring which is active on the  
20 driven sheave, this action takes place by the agency of the tension of the belt. The result is a previously established correspondence between the speed (rpm) of the engine and the transmission ratio; this correspondence is modified only slightly by the  
25 magnitude of the transmitted torque (belt effect), in the sense of counteracting the action of the centrifugal device as the torque is increased. Such a result is far from being satisfactory since it is not possible to maintain the lowest gear ratios while exploiting  
30 the full engine power to climb steep slopes and whenever a top pickup is desired, even when the vehicle speed is comparatively high, or, conversely, it becomes difficult to pass to higher gear ratios when a silent run under economy-run conditions is  
35 desired. Attempts towards offsetting this drawback have been made by adding mechanical devices responsive to the transmitted torque, but with poor results, however, because very high loads would be required, which are difficult to embody constructionally, for counteracting, throughout the entire range  
40 of operative rpm values the effect of the centrifugal device: moreover, such devices introduce into the regulation system considerable frictional forces which, being added to those of the centrifugal device  
45 itself, reduce the sensitivity of the latter device too much.

50 In the practical applications on motor cars, an approach is known which is torque sensitive and consists of a pneumatic servomechanism actuated by the negative pressure downstream of the carburettor throttle, said pressure being variable as the accelerator pedal position is varied, but the direct mechanical control of the centrifugal device on the driving sheave remains unaltered and thus the high  
55 frictional forces also remain unaffected. More sophisticated approaches are also known with a regulation which is completely made with intermediary members and in which both the expansible sheaves are controlled by fluid-actuated hydraulic rams, the pressure of which is modulated as function  
60 of the engine rpm and the required load, the operation of such mechanisms being fully responsive to the expected. Such a contrivance however is too intricate to be adopted on small vehicles.

65 In addition, inasmuch as these approaches retain

their functional feature, which is common also the mechanical approaches discussed above, and which consists in obtaining a determined ratio through the balance of the forces applied to the two sheaves

70 both from the outside and from the belt, they are considerably influenced by the inevitable variations of the internal frictional forces of the entire regulation and drive-transfer assembly, the variations of the viscosity of the working fluid and the dimensional variations due to the inevitable machining allowances of all their components parts and especially  
75 of the resilient members, so that their characteristic can vary considerably both due to mass production and the conditions of practical use. Such devices  
80 thus require accurate and expensive checking operations during construction.

This invention has the tasks of overcoming the defects of the automatic mechanical control transmissions without incurring the intricacy of the  
85 integral hydraulic control system and, in addition, the invention aims to overcoming the aforesaid difficulties inherent in maintaining the specifications in a mass production and in use in order to become enabled to apply the transmission system to small  
90 vehicles.

To solve this problem, according to the invention, a transmission system is suggested having mechanical control and a transmission ratio variable continuously of the V-belt and expansible sheave type,  
95 each sheave comprising a fixed half-sheave and a movable half-sheave, said system being characterized in that only the movable half-sheave of either of said expansible sheaves is held in the position corresponding to the transmission ratio as determined by the preselected law of dependence on the engine rpm and load, by means of a servomechanism which automatically varies the position, whereas on the movable half-sheave of the other expansible sheave the axial load is applied of a member  
100 which is adapted to provide the adherence of the belt to both said expansible sheaves.

By so doing, the transmission ratio which is obtained is, therefore, no longer consequent to the balance of the axial loads which are applied to both sheaves, in combination with the "belt effect", but is only and exclusively a consequence of the relative positioning as established with accuracy for the two half-sheaves of the sheave which is controlled by the servomechanism in question.

115 More particularly, such servomechanism can be controlled, also as a function of the engine rpm and of the position of the accelerator control of the same engine, by a centrifugal governor having a biasing spring which can be variably, preloaded, the spring being servoed to the accelerator control and the preselected transmission ratio is obtained by the positioning of the collar of said governor through a follower hydraulic mechanisms which determines the position of said either half-sheave. The physical  
120 law which binds the variables, speed, throttle angle of aperture and transmission ratio, is previously established as a function of the features of the engine and the vehicle and is embodied in the particular characteristic of the centrifugal governor  
125 which is made properly variable as the throttle

aperture angle varies, by a biasing spring the flexibility of which is variable as a function of the stroke and also of the preload.

The sensitivity of the regulation system provided by the invention is thus made independent of the friction between the belt and the sheaves, of the axial relative sliding friction of the half-sheaves and the viscosity variation of the motive fluid. The physical law of variation of the transmission ratio is thus influenced only by the constructional characteristics of the mechanical regulation system, said characteristics being easily controlled. A variation of the preselected law of variation of the ratio can be achieved only by adopting belts of a different length; such as variation can be corrected during the initial calibration operations for the device and/or during practical use.

For a better understanding of the invention, two embodiments have been diagrammatically shown by way of example only in the accompanying drawings, wherein:

*Figure 1* is a constructional layout of an embodiment in which the servomechanism controls the driven sheave and

*Figure 2* is a constructional layout of another embodiment in which the servomechanism controls, instead, the driving sheave.

In the embodiment shown in *Figure 1*, the transmission system comprises a driving sheave A driven to rotation by a driving shaft 1 and composed of a fixed half-sheave 2, an axially slidable half-sheave 3 and a resilient member 4 which is active in the sense of mutually approaching the two half-sheaves 2 and 3, squeezing and tensioning the V-belt 5.

By means of the V-belt 5 the driven sheave B is caused to rotate and is composed of an axially fixed half-sheave 6 and an axially slidable half-sheave 7 integral with a shaft 8. The half-sheave 7 is urged by a spring 22 so as to take the closed position (close to the fixed half-sheave 6), against the contrary bias exerted via the tension of the belt 5 by the spring 4 active against the movable half-sheave 3 of the driving sheave A.

The bias of the spring 22 is, moreover, counteracted by the push of a hydraulic ram 10 subjected to the pressure of the motive fluid contained in the chamber 12 of the fixed cylinder 11, through the stem 9 of a piston 10 and the shaft 8. The chamber 12 is fed with a pressurized liquid by a pump 13. The fluid can exit the chamber 12 through a duct 14 formed through the stem 9 and opens to the outside through a radial port 15. The port 15 can be shut by a valve having the shape of a bushing 16 slidable over the stem 9. The bushing 16 can be shifted by the arm 17 of a movable beel 26 subjected to the thrust of the masses 27 of a centrifugal governor 18, carried for rotation by the driving shaft 1.

The shift of the bell 26 is counteracted by a spring 20, the preload of which can be varied from the outside via a lever 21 the movement of which is linked to the position of the accelerator control lever (not shown).

The outline of the internal surface of the bell 26 and spring 20 of the governor 18 have a shape and a characteristic, respectively, such that they provide

the preselected physical law of the variation of the rpm as a function of the engine torque, as outlined hereinbefore.

The displacement of the piston 10 as caused by the pressure of the motive fluid, originates, through the shaft 8, the shifting of the half-sheave 7 towards the positions corresponding to the lower gear ratios, overcoming the bias of the spring 22.

To every position of the bell 26, as a function of the engine rpm and the position of the accelerator control lever, there corresponds univocally a position of the bushing 16 and thus of the piston 10.

The piston 10, in fact, finds its balance when the port 15 is so widely open as it is necessary to deliver the entire rate of flow of the pump 13 and the pressure drop through the port 15 is such as to originate a resistive force which is equal to the bias of the spring 22 minus the load acting upon the slidable half-sheave 7. Thus, the half-sheave 7 must follow the axial shift of the bushing 16. The liquid delivered through port 15 is received in a tank 19 from which it is drained away by the pump 13.

The drive is received at the outside by a shaft 25 through the couple of gears 23 and 24, of which gear 23 is integral with the fixed half-sheave 6 of the driven sheave B.

In the embodiment shown in *Figure 2* the transmission system essentially comprises the same component parts as in *Figure 1*, with the exception that the servomechanism is active on the driving sheave A.

Like parts corresponding to those shown in *Figure 1* are connoted with the same numerals in *Figure 2*. The fixed half-sheave 2 of the driving sheave A is once again integral with the mainshaft 1, whereas the movable half-sheave 3 is integral with a shaft 28. Shaft 28 is kept in contact with the stem 9 of the hydraulic ram 10, slidable in the fixed cylinder 11, by the thrust of the movable half-sheave 3 urged axially by the pull of the belt 5 as originated by the bias of the spring 22 which is active on the movable half-sheave 7 of the driven sheave B. Also in this case the variable-capacity chamber 12 is defined between the piston 10 and the cylinder 11, the chamber being fed by the pump 13 with a pressurized liquid. The liquid can exit the chamber 12 through the duct 14 and the radial port 15 and it is collected in the tank 19 from which it is drained away by the pump 13. The port 15 is throttled by the bushing 16 connected via the arm 17 to the movable bell 26 of the centrifugal governor 18, the latter being rotated by the mainshaft 1.

The shift of the bell 26 is resisted by the spring 20, which can be preloaded variably by means of the lever 21, the movement of the latter being served to the position of opening of the accelerator control lever.

In this case a single resilient member 22, is enough instead of the two (4 and 22) shown in *Figure 1*, since the relative bias serves also to provide the positioning of the transmission system in the lowest gear ratio when the engine is stopped.

The operation of the transmission system according to *Figure 2* is very much the same as that of the previous example on taking into account, however,

that in the case of Figure 2 an rpm increase of the engine commands the closure of the driving shave A instead of opening the driven sheave B.

Also in the example of Figure 2 the drive is collected at the outside by a shaft 25 via a couple of gearwheels 23 and 24.

On the basis of what has been described above, the sensitivity of the governor is conditioned only by its internal frictional forces and not also by those which oppose the variation of the drive-transfer ratio. The constructional simplification of the servomechanism is evident since it contains only a single-effect hydraulic ram. Lastly, inasmuch as the centrifugal regulator 18 can be small sized, also the force required to vary the preload of its spring 20 is small, so that the preload variation can be obtained manually and simply.

Due to its properties as outlined above the transmission system according to the invention lends itself particularly to being applied to motor bicycles and such like, driven by internal-combustion engines.

It is understood that the invention can be embodied variously from the constructional standpoint without departing from its basic idea. For example, the centrifugal governor 18 can also be of a non-mechanical type such as of an electrical or a hydraulic type and can also be influenced by other variables or it can be subjected to the direct action of the motor bicycle driver.

#### CLAIMS

1. A transmission system with mechanical control and variable ratio with continuity of the kind having a V-belt and expansible sheaves, each of which comprises a fixed half-sheave and a movable half-sheave, characterized in that the position of only one of the movable half-sheaves is determined as a function of the rpm and the load of the engine by a servomechanism automatic of positioning, the other movable half-sheave being subjected only to the axial load of a member adapted to ensure the adherence of the belt to said expansible sheaves.

2. A transmission system according to claim 1, characterized in that said positioning servomechanism is of the hydraulic type and comprises a governor adapted to determine, as a function of the rpm and the load, the position of an outlet valve for the motive fluid of the servomechanism and a hydraulic piston moved by said motive fluid, pressurized by a specially provided pump, said piston acting upon said movable half-sheave through a stem in which there has been formed the outlet port for the motive fluid as controlled by said valve.

3. A transmission system according to claim 2, characterized in that said governor is of the centrifugal mass type with biasing springs with characteristic variable as a function of the load by a control for the spring preload directly connected to the fuel-feed control for the engine, the power of which is transmitted by said transmission system.

4. A transmission system according to claim 2, characterized in that the hydraulic piston of the servomechanism controls the movable half-sheave

of the driven sheave against the bias of a resilient member which is active upon said movable half-sheave itself.

5. A transmission system according to claim 2, characterized in that the hydraulic piston of the servomechanism controls the movable half-sheave of the driving sheave and is counteracted via the tension pull of the belt by the load of the resilient member which is active upon the movable half-sheave of the driven sheave.

6. A transmission system according to claim 2, characterized in that the governor is of a centrifugal mass type with biasing springs and that the spring of the governor is manually adjusted by the vehicle driver in parallel with the control of the feeding of the fuel to the engine.