THE DESCRIPTION OF THE "JUPITER" HYDROMECHANICAL CONTINUOUSLY VARIABLE SPEED TRANSMISSION

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The internal combustion motor, that looks back to a past of nearly 100 years — if we suppose the gas engine of the French Lenoir as being its ancestor — has kept its characteristics also in development. Although the torque-characteristic of modern internal combustion motor is very much improved, it does not satisfy the operating claims of an automobile today. Therefore the problem of speed changing was simultaneously born with internal combustion-engine driven automobile. During the last decades many types of transmissions were developed but as soon as the territory of use of the internal combustion motor grew wider and it became of greater importance in road traffic as well as in railway traction, various types of transmissions were created. Up to now gear wheel transmissions were best suited and have been mostly used, more complicated are planetary systems which are chiefly applied because of their easy automatization. At last, especially for heavier vehicles, mechanical transmissions combined with hydraulics, and later, the purely hydraulic transmissions came forward. Although some types of complete hydraulical systems solve the problem of infinitely variable speed changing but with unsatisfactory efficiency, that means with great losses.

The JUPITER transmission solves the problem of infinitely variable speed changing by two coupled planetary gear sets and two hydraulic pumps and — motors.

The principal arrangement of transmission, the coupling of planetary gears and hydraulic aggregates (pump and hydraulic motor) are illustrated in Fig. 1. The 1st, 3rd and 5th speeds can be produced by suitable coupling of the two planetary gear sets, while the 1st and 3rd speeds, furthermore the 3rd and 5th ones are bridged over by the hydromecanically operated infinitely variable 2nd and 4th speeds. In this way the whole speed changing range is infinite-variably realized by the hydraulically operated 2nd and 4th speeds. In the 5th speed, *i. e.* in direct transmission neither planet-gears nor hydraulics operate.

The advantage of the just discussed transmission to the so far known and manufactured other types is that mechanical transmission to great part overcomes the speed-changing range. There are three speeds entirely mechanically operated: the 1st (low) one for starting, the 3rd the middle or relax speed and the 5th the direct (high) speed.

In the variable 2nd and 4th speeds the power of the motor branches into two directions in the transmission, into mechanical and hydraulical parts. But even at these speeds the mechanical percentage becomes greater than the hydraulic one. Hydraulic transmission that consists of pump and hydraulic motor takes part only in infinite r. p. m. changing. So power, inducted hydraulically to planetary gears is equal to zero at the beginning of variable 2nd and

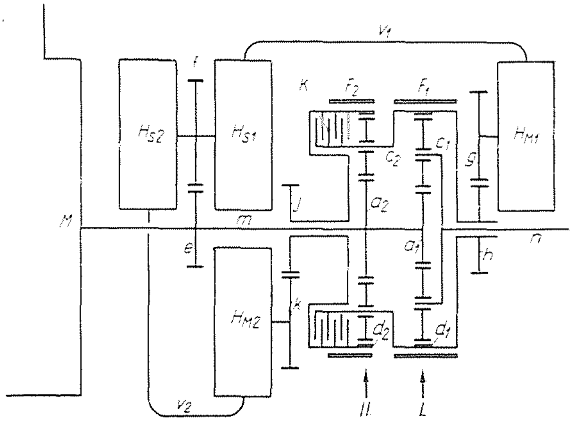

Fig. 1

4th speeds. At the end — depending on decreasing ratio — it only means a certain part of input power. In power-splitting operated hydraulics and the 1st, 3rd and 5th exclusively mechanical speeds assure a very good efficiency to the transmission in the whole speed changing range. The mechanical efficiency of planetary gears is excellent because no "blind power" operates in planetary systems, *i. e.* the rolling power of gears is always equal to the transported power.

The transmission assures the possibility of braking with the motor — generally used on motor vehicles — at any speed ratio, because pumps can be operated as hydraulic motors and the motors as hydraulic pumps.

The 2nd and 4th variable speed ratios operate hydromechanically from the start of the vehicle until its highest speed. Starting aid is given by the mechanically shifted 1st speed, the ratio of which corresponds to the beginning ratio of the 2nd speed. The 1st speed is able to overcome the steepest gradients and is suitable for permanent slow traffic. The ratio of mechanically working 3rd speed is the same as the end-ratio of the variable ratio 2nd speed and the beginning ratio of the variable 4th speed. The mechanical 3rd speed overbridges the time between two speed changings and is suitable for mastering average gradients. In the 5th speed the vehicle can run in direct transmission (1:1) with the highest velocity.

The solving of the problem of reverse-speed on various vehicles will be discussed later on. The principal design of the transmission is illustrated in Fig. 1. In the 1st speed of the two coupled planetary gears but the I. set works. The a_1 sun-gear is driven by the motor, the d_1 annulus gear is locked by F_1 band-brake and the c_1 planet carrier rotates with its arm the (secondary) cardan-shaft with a revolution reduced by the maximal reduction of the planetary gear.

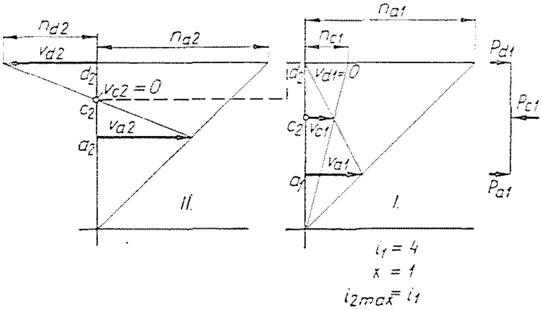

Fig. 2

The speed-, and revolution-scheme is to be seen in Fig. 2 together with the free-wheeling II. planetary gear. The dimensions of gears applied to gear-sets expressed by comparative numbers are the following.

In the I. planetary-set the diameter and number of the teeth on sun-gear a_1 and planet b_1 are the same therefore as $a_1 = b_1$. The diameter of planet-carrier $c_1 = 2\ a_1$, that of annulus is $d_1 = 3\ a_1$. To simplify the calculation the radii of gears will be:

$$r_{a1} = 1 \quad r_{c1} = 2 \text{ and } r_{d1} = 3$$

The diameter of sun-gear is, in the II. planetary set, $a_2 = 1\frac{2}{3} \cdot a_1$.

The diameter of planet is $b_2 = 2/3 \cdot a_1$.

The diameter of planet-carrier is $c_2 = 2\frac{1}{3} \cdot a_1$.

The diameter of annulus gear is $d_2 = 3 \cdot a_1$.

Radii as $r_{a1} = 1$, so $r_{a2} = 1\frac{2}{3}$, $r_{c2} = 2\frac{1}{3}$ and $r_{d2} = 3$.

The power of the driving motor is $N_M = 1$, revolutions per minute are $n_M = 1$, driving torque is $M_M = 1$, so the power is not expressed by HP-s but is 716,2 times larger.

On shifting the 1st speed the engine drives the sun-gear of I. planetery gears by the input shaft m, brake F_1 locks annulus gear d_1 while arm c_1 drives secondary (output) shaft n. The mechanical relations of planetary gears are the following in accordance with speed and revolution schemes of Fig. 2.

Circumferential velocities:

$$V_{a1} = 1, \; V_{c1} = 0{,}5 \text{ and } V_{d1} = 0$$

revolutions:

$$n_{a1} = 1, \; n_{c1} = 0{,}25 \text{ and } n_{d1} = 0.$$

The speed ratio which is to be had directly from the diameter of the gears of the planetary system by the following equation:

$$i_1 = \frac{d_1}{a_1} + 1 = \frac{3}{1} + 1 = 4$$

further it will be called the ground ratio of the I. planetary gear.

Tangential forces:

$$P_{a1} = 1, \; P_{c1} = 2 \text{ and } P_{d1} = 1; \; (P_{d1} = P_{a1} \text{ and } P_{c1} = P_{a1} + P_{d1})$$

torques:

$$M_{a1} = P_{a1} \cdot r_{a1} = 1, \; M_{c1} = P_{c1} \cdot r_{c1} = 4 \text{ and } M_{d1} = P_{d1} \cdot r_{d1} = 3$$

so:

$$M_{a1} : M_{c1} : M_{d1} = 1 : 4 : 3 \quad M_{c1} = M_{a1} + M_{d1}$$

$$M_{c1} = i_1 \cdot M_{a1} = 4 \text{ and } M_{d1} = (i_1 - 1)\, M_{a1} = 3$$

As for the rest M_{c1} is the torque transmitted to secondary shaft n, and M_{d1} is the reactive moment of planetary gear I. at ratio i_1.

Power outputs:

$$N_{a1} = M_{a1} \cdot n_{a1} = 1 \text{ driving}$$

$$N_{c1} = M_{c1} \cdot n_{c1} = 1 \text{ driven}$$

$$N_{d1} = M_{d1} \cdot n_{d1} = 0.$$

The input power of engine $N_a = N_M$ on sun-gear a_1 equals to output power on c_1 planet carrier.

The 2nd speed operates hydromechanically variably changed between mechanically shifted 1st and 3rd speeds. Therefore, maximal ratio of 2nd speed equals to 1st speed ratio ($i_{2\,max} = i_1$) and continuously decreases until ratio of 3rd speed.

Minimal ratio of 2nd speed is $i_{2\,min} = i_3$

On shifting the 2nd speed first the F_1 brake of I planetary gear will become unlocked by which the d_1 annulus gear is freed. Simultaneously, the H_{S1} hydraulic pump will be coupled, that is driven, by the motor through e and f gears. The pump feeds the hydraulic motor with the high-pression oil delivered on the v_1 pipe. The motor rotates the d_1 annulus gear of 1st planetary gear through g and h gears. The variable speed revolution-regulation of hydraulic motor is possible from zero up to the suitable revolution by controlling the hydraulic pump, during constant revolution of the driving motor. In the 2nd

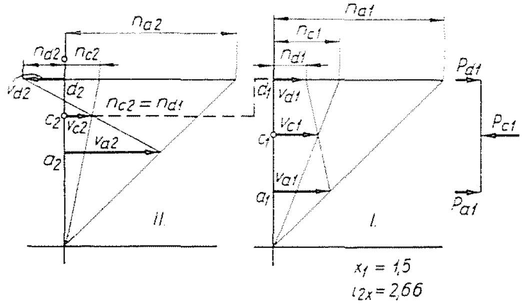

Fig. 3

speed ratio the I. planetary gear becomes dual-driven and power-splitting is carried out. The a_1 sun-gear is driven by the power-conveying in the mechanical branch-off, the d_1 annulus gear by the hydraulic one. It is evident that at the beginning of variable 2nd speed when $i_{2\,max} = i_1$ and the H_{M1} oil-motor and the coupled d_1 annulus are still standing, transmitted power is zero in the hydraulic line. However, as H_{M1} oil-motor slowly begins to rotate the d_1 annulus, hydraulically transmitted power will grow, simultaneously the power necessary for driving the a_1 sun-gear transferred in the mechanical line diminishes, but the sum of the power transported in the two lines remains the same and equals the power of the motor. In the whole r.p.m.-range of the variable 2nd speed only the I. planetary gear works with hydromechanical dual-driving. At the beginning of the 2nd speed when $i_{2\,min} = i_1$ the mechanical circumstances of I. planetary gear correspond to its 1st speed. Therefore, in that case the speed and revolution schemes shown in Fig. 2 are available for the I. planetary gear. The II. planetary gear rotates freely in the whole range of 2nd speed.

In Fig. 3 the speed and revolution scheme of the working I. and the free rotating II. planetary gear are shown, on a determined point of variable 2nd speed. The I. planetary gear is 1,5 times accelerated by the H_{S1}, H_{M1} hydraulic pump and motor. In the following discussion it will be expressed by x acceleration factor. The speed ratio of I. planetary gear belonging to it will be designated by i_{2x} in the 2nd speed to distinguish it from the ground ratio of planetary gear. The ground ratio is: $i_1 = \frac{d}{a} + 1$. According to it, the ratio of double-drive accelerated planetary gear can be expressed, in a general case, by the following equation:

$$x \cdot i_x = i,$$

in ground ratio

$$x = 1 \text{ and } i_x = i$$

In the foregoing case

$$x_1 \cdot i_{2x} = i_1$$

$$i_{2x} = i_1/x_1 = 4/1,5 = 2,66$$

that means the n output shaft is driven by the motor through the m shaft of the gear-box 2,66-times slower than the motor rotates, with a 2,66-times higher torque. From the unified speed and revolution scheme the mechanical circumstances of I. planetary gear can be fixed, but at the intermedial ratio of 2nd speed power-splitting will be of the greatest interest. The a_1 sun-gear of I. planetary gear is driven by the motor. As the ratio of planetary gear has been diminished x-times, the power transmitted in mechanical split-line will be x-part of the motor-power, so

$$N_{a1} = N_M/x_1 = 1/1,5 \cdot N_M = 0,66\ N_M.$$

the power inducted by the hydraulic motor on the d_1 annulus of I. planetary gear is:

$$N_{d1} = N_M \left(1 - \frac{1}{x_1}\right) = (1 - 1/1,5) \cdot N_M = 0,33\ N_M$$

The variable 2nd speed lasts till mechanical shiftable 3. speed, that means the d_1 annulus of the I. planetary gear is so far accelerated by the hydraulic motor till the d_2 annulus of the free rotating II. planetary gear stops. These circumstances are shown in the upper part of Fig. 4.

The unified speed and revolution scheme of the working I. and free-wheeling II. planetary gears can be seen in Fig. 4. From these it can be stated that r.p.m. of d_2 annulus of the II. planetary gear will be $n_{d2} = 0$ only when the acceleration of the I. planetary gear is $x_1 = 2,083$. The minimal beginning ratio of the variable 2nd speed is:

$$i_{2\,\min} = i_1/x_1 = 4/2,083 = 1,92.$$

The mechanically inducted power of the motor on the a_1 sun-gear of the I. planetary gear is:

$$N_{a1} = N_M/x_1 = N_M/2{,}083 = 0{,}48\ N_M$$

and hydraulically inducted power of the motor on the d_1 annulus of the I. planetary gear is:

$$N_{d1} = N_M(1-1/x_1) = N_M(1-1/2{,}083) = 0{,}52\ N_M$$

$$N_{a1} = N_{d1} + N_M \qquad 0{,}48 + 0{,}52 = 1{,}0.$$

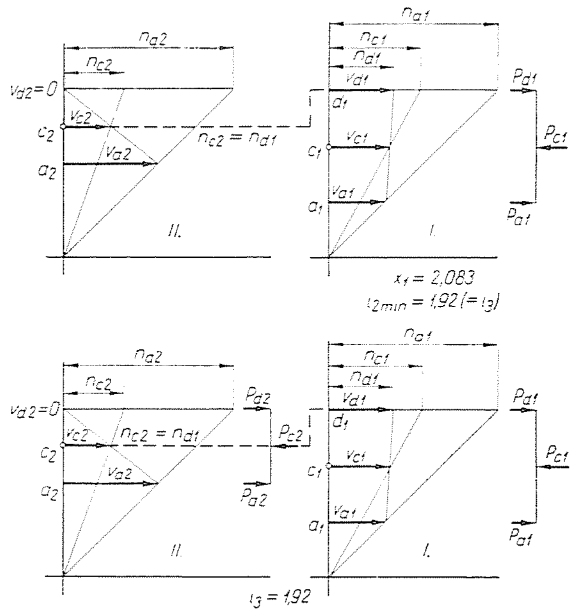

Fig. 4

The change of power-splitting in function of x acceleration factor of planetary gear conforms to the laws of hyperbola-function. The equations above give the possibility of exactly calculating the power which is splitted into mechanical and hydraulical parts at any i_{2x} ratio of the variable 2nd speed. The minimal, the so-called end-ratio of the variable 2nd speed conforms with the ratio of the mechanically shifted 3rd speed. The shifting of the 3rd speed is carried out after putting out the H_{S1} oil-pump and the H_{M1} hydraulical motor by locking the d_3 annulus of the II. planetary gear. The locking of d_2 annulus gear happens with the F_2 brake. The working of the brake is slipless because at the end of the 2nd speed the d_2 annulus gear is already standing, $i_{2\ min} = i_3$. The unified speed- and revolution scheme of the two planetary gears coupled to each other is the same as in the foregoing case, but its working is totally different, as both of the two planetary gears operate. In Fig. 4 the wheel tangential forces are already shown below. The power of the motor is inducted in the two coupled planetary gears on the sun-gear a_1 of I. planet and on a_2 of the II. planet. The II. planetary gear is of single action, it gets the motor-power only on the sun-gear a_2 which is delivered by the c_2 planet carrier to the d_1 annulus of I. planet. So the I. planetary gear is of dual-driving in the 3rd speed too. The ground ratio of the I. planet ($i_1 = 4$) is diminished purely mechanically by the coupled II. planet until ratio of 3rd speed, $i_3 = 1{,}92$.

The mechanical circumstances of the I. planetary gear:
tangential speeds:

$$V_{a1} = 1; \; V_{c1} = 1{,}042 \text{ and } V_{d1} = 1{,}083$$

revolutions:

$$n_{a1} = 1; \; n_{c1} = 0{,}52 \text{ and } n_{d1} = 0{,}36$$

tangential forces:

$$P_{a1} = 0{,}48; \; P_{c1} = 0{,}96 \text{ and } P_{d1} = 0{,}48$$

torques:

$$M_{a1} = 0{,}48; \; M_{c1} = 1{,}92 \text{ and } M_{d1} = 1{,}44$$

$$M_{a1} : M_{c1} : M_{d1} = 1 : 4 : 3$$

$$M_{c1} = M_{a1} + M_{d1}$$

power output:

$$N_{a1} = 0{,}48 \cdot N_M; \; N_{c1} = N_M \text{ and } N_{d1} = 0{,}52 \, N_M.$$

Mechanical circumstances of the II. planetary gear:
tangential velocities:

$$V_{a2} = 1{,}66; \; V_{c2} = 0{,}83 \text{ and } V_{d2} = 0$$

revolutions:

$$n_{a2} = 1; \; n_{c2} = n_{d1} = 0{,}36 \text{ and } n_{d2} = 0$$

tangential forces:

$$P_{a2} = 0{,}31; \; P_{c2} = 0{,}62 \text{ and } P_{d2} = 0{,}31$$

torques:

$$M_{a2} = 0{,}52; \; M_{c2} = 1{,}44 \text{ and } M_{d2} = 0{,}92$$

the reaction moment of the annulus gear which is taken up by F_2 brake,

$$M_{d2} = (i_3 - 1) \, M_M; \; M_{d2} = M_{c2} - M_{a2}$$

power outputs

$$N_{a2} = N_M - N_{a1} = 1 - 0{,}48 = 0{,}52$$

driving

or

$$N_{a2} = M_{a2} \cdot n_{a2} = 0{,}52 \cdot 1{,}00 = 0{,}52$$

$$N_{c2} = M_{c2} \cdot n_{c2} = 1{,}44 \cdot 0{,}36 = 0{,}52 \text{ driven}$$

$$N_{d2} = M_{d2} \cdot n_{d2} = 0.$$

The mechanically operated 3rd speed need not be shifted if the motor disposes of a reserve of power to considerably accelerate the vehicle. For the rest this speed is accounted for as relaxation speed which gives the H_{S2}, H_{M2} second hydraulic aggregate enough time to begin to operate efficiently.

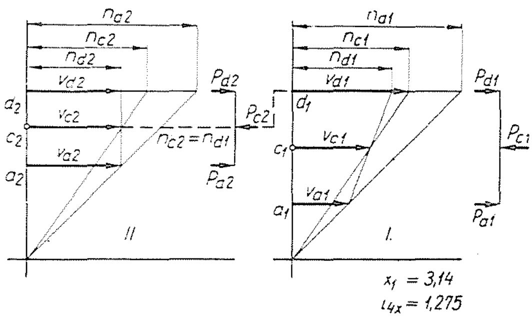

Fig. 5

In the case above, as it is directly shifted from the variable 2nd speed to the variable 4th speed the second hydraulic aggregate must be put in action before ending the 2nd speed ratio. Then all advantages of the variable speed transmission can be exploited to accelerate the vehicle.

The shifting of the variable 4th speed is the following. After unlocking brake F_2 (supposing 3rd speed to be shifted) d_2 annulus becomes free. Further diminishing of ratio is carried out by the second H_{S2}, H_{M2} hydraulic pump and motor that rotates the d_2 annulus of the II. planetary gear by the j and k gears. So, in the 4th speed the motor drives the a_2 sun-gear of the II. planet and a_1 sun-gear of the I. planet through the m shaft, while the hydraulic motor drives the d_2 annulus of the II. planetary gear. At the beginning of the 4th speed the ratio of the two planets is $i_{4max} = i_3$, then the hydraulic motor accelerates the revolutions of the d_2 annulus until it reaches that of the motor, $i_{4min} = 1$.

At the beginning of the 4th speed until $i_{4max} = i_3$ the unified speed and revolution schemes of the two planetary gears fit Fig. 4. Let us examine the mechanical circumstances of the planets in this case too, at any i_{4x} ratio that falls between i_{4max} and i_{4min}.

In Fig. 5 the speed and revolution scheme of the two planetary gears is laid out at $x_1 = 3{,}14$-times acceleration referring to the I. planetary gear. Then the ratio of the variable 4th speed is:

$$i_{4x} = \frac{i}{x_1} = \frac{4}{3{,}14} = 1{,}275$$

The mechanical circumstances of the II. planetary gear:
tangential velocities:

$$V_{a2} = 1{,}66\,; \; V_{c2} = 1{,}66 \text{ and } V_{d2} = 1{,}66$$

revolutions:

$$n_{a2} = 1\,; \; n_{c2} = 0{,}715 \text{ and } n_{d2} = 0{,}555$$

tangential forces:

$$P_{a2} = 0{,}205\,; \qquad P_{c2} = \frac{r_{d1}}{r_{c2}} P_{d1} = \frac{3}{2{,}33} \cdot 0{,}32 = 0{,}41 \text{ and } P_{d2} = 0{,}205$$

torques:

$$M_{a2} = P_{a2} \cdot r_{a2} = 0{,}205 \cdot 1{,}66 = 0{,}34$$

$$M_{c2} = P_{c2} \cdot r_{c2} = 0{,}410 \cdot 2{,}33 = 0{,}96$$

$$M_{d2} = P_{d2} \cdot r_{d2} = 0{,}205 \cdot 3{,}00 = 0{,}62$$

$$M_{a2} + M_{d2} = M_{c2}^{?}$$

power outputs:

$$N_{a2} = M_{a2} \cdot n_{a2} = 0{,}34 \cdot 1{,}00 = 0{,}34$$

$$N_{c2} = M_{c2} \cdot n_{c2} = 0{,}96 \cdot 0{,}71 = 0{,}68$$

$$N_{d2} = M_{d2} \cdot n_{d2} = 0{,}62 \cdot 0{,}55 = 0{,}34$$

N_{a2} is the power transmitted in the mechanical line, and N_{d2} is the power transmitted in the hydraulical line. The $N_{c2} = N_{a2} + N_{d2}$ hydromechanically transmitted power is conveyed by the c_2 arm of the II. planetary gear to the connected d_1 annulus of the I. planet.

The mechanical terms of the I. planetary gear:
tangential forces:

$$P_{a1} = M_{a1}/r_{a1} = 0{,}32\,; \; P_{c1} = 0{,}64 \text{ and } P_{d1} = 0{,}32$$

tangential velocities:

$$V_{a1} = 1\,; \; V_{c1} = 1{,}57 \text{ and } V_{d1} = 2{,}07$$

revolutions:

$$n_{a1} = 1\,;\; n_{c1} = 0{,}785 \text{ and } n_{d1} = 0{,}715$$

power outputs:

$$N_{a1} = M_M/x_1 = 1/3{,}14 = 0{,}32 \quad (= M_{a1} \cdot n_{a1})$$

$$N_{c1} = N_M = 1{,}0 \qquad (= M_{c1} \cdot n_{c1})$$

$$N_{d1} = N_{c2} = 0{,}68 \qquad (= M_{d1} \cdot n_{d1})$$

$$N_{d1} = N_{c1} - N_{a1}$$

torques:

$$M_{a1} = N_{a1}/n_{a1} = 0{,}32/1{,}0 = 0{,}32$$

$$M_{c1} = i_{4x} \cdot M_M \qquad M_{c1} = N_{c1}/n_{c1} = 1{,}0/0{,}785 = 1{,}28$$

$$M_{d1} = N_{d1}/n_{d1} = 0{,}68/0{,}715 = 0{,}96$$

tangential forces:

$$P_{a1} = M_{a1}/r_{a1} = 0{,}32\,;\; P_{c1} = 0{,}64 \text{ and } P_{d1} = 0{,}32.$$

Summing up the above-mentioned: in the two planets following power-splitting occurs and in the variable 4th speed at $i_{4x} = 1{,}275$ ratio: the a_2 sun-gear of the II. planet operates in the mechanical line $N_{a2} = 0{,}34$. N_M and the a_1 sun-gear of the I. planet with $N_{a1} = 0{,}32$. N_M power output, $N_{a2} + N_{a1} = 0{,}66\ N_M$ and the d_2 annulus of the II. planet works in the hydraulical line with $N_{d_2} = 0{,}34 \cdot N_M$ power output.

The power for the n secondary shaft transmitted on to the planetary gears is:

$$N_{c1} = N_{a1} + N_{a2} + N_{d2} = 1{,}0 \qquad (= N_M)$$

If the powers to be determined are those transmitted exclusively in power-splitting in the two connected planetary gears, the procedure is the following one:

If $x_1 = 2{,}083$, so at $i_{4\max} = i_3$ it only gives mechanically transmitted power.

If $x \leqq 2{,}083$ e. g. in the case above where $x_1 = 1{,}275$ so the power transmitted in mechanical split-line is:

$$N_{\text{mech}} = N_M\left(1 - \frac{1}{x_1}\right) = N_M\left(1 - \frac{1}{3{,}14}\right) = 0{,}66\ N_M$$

and the hydraulically transmitted power is

$$N_{\text{hydr}} = N_M/x_1 = 1/3{,}14 \qquad N_M = 0{,}34\, N_M \qquad (= N_{d2})$$

In the result found above as $N_{m\text{-}ch} = N_{a1} + N_{a2}$ the powers transmitted on the sun-gears of the I. and II. planet are not to be established one by one. But when instead of the x acceleration factor sufficient for the end-ratio we introduce the $-x_2 = 2$ acceleration factor referring to the II. planetary gear, so the half of $N_{c2} = 0{,}68 \cdot N_M$ driven power-output will be inducted in the hydraulical line on d_2 annulus gear:

$$N_{d2} = N_{c2}/x_2 = 0{,}68/2 \cdot N_M = 0{,}34\ N_M$$

at the intermediate ratio of the 4th speed.

On the minimal ratio of variable 4th speed the second hydraulic motor (H_{M2}) accelerates d_2 annulus of II. planetary gear, until its revolutions per minute equal those of the motor, respectively of those of a_1 and a_2 sun-gear.

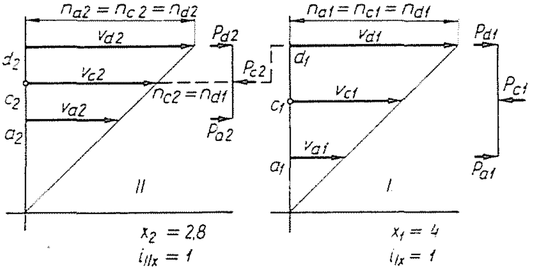

Fig. 6

The minimal ratio of the 4th speed is $i_{min} = i_5 = 1$, the acceleration of the II. planet $x_{2max} = 2{,}8$ the end-acceleration of the system referring to the I. planetary gear is $x = 4$.

That means the ratio of the II. planet at the end of the 4th speed is:

$$i_{IIx} = \frac{i_{II}}{x_2} = \frac{2{,}8}{2{,}8} = 1$$

for the I. planetary gear:

$$i_{Ix} = \frac{i_I}{x_I} = \frac{4}{4} = 1$$

The assembled velocity and revolution scheme of both planetary gears at the end of the 4th speed is shown in Fig. 6. The ratio of planets is 1 : 1, there is no roll-down in it, in spite of the fact that power-splitting takes place in the same way as in the case of the intermediate ratio, mentioned above.

The mechanical terms of the II. planetary gear:

tangential velocities:

$$V_{a2} = 1{,}66, \; V_{c2} = 2{,}2 \text{ and } V_{d2} = 3$$

revolutions:

$$n_{a2} = n_{c2} = n_{d2} = 1$$

tangential forces:

$$P_{a2} = 0{,}16\,, \qquad P_{c2} = \frac{r_{d1}}{r_{c2}} P_{d1} = \frac{3}{2{,}33} \cdot 0{,}25 = 0{,}32 \text{ and } P_{d2} = 0{,}16$$

torques:

$$M_{a_2} = P_{a2} \cdot r_{a2} = 0{,}16 \cdot 1{,}66 = 0{,}27$$

$$M_{c2} = P_{c_2} \cdot r_{c2} = 0{,}32 \cdot 2{,}33 = 0{,}75$$

$$M_{d2} = P_{d2} \cdot r_{d2} = 0{,}16 \cdot 3{,}00 = 0{,}48$$

$$M_{a2} + M_{d2} = M_{c2}$$

power outputs:

$$N_{a2} = M_{a2} \cdot n_{a2} = 0{,}27$$

$$N_{c2} = M_{c2} \cdot n_{c2} = 0{,}75$$

$$N_{d2} = M_{d2} \cdot n_{c2} = 0{,}48$$

$$N_{a2} + N_{d2} = N_{c2}.$$

The mechanical relations of the I. planetary gear:

tangential velocities:

$$V_{a1} = 1, \; V_{c1} = 2 \text{ and } V_{d1} = 3$$

revolutions:

$$n_{a1} = n_{c1} = n_{d1} = 1$$

power outputs:

$$N_{a1} = M_M/x_1 = 1/4 = 0{,}25 \quad \text{mechanical split line}$$

$$N_{c1} = N_M = 1$$

$$N_{d1} = N_{c2} = N_{c1} - N_{a1} = 0{,}75 \text{ hydromechanical}$$

$$N_{a1} + N_{d1} = N_{c1} = N_M \qquad \text{split line}$$

torques:

$$M_{a1} = N_{a1}/n_{a1} = 0{,}25$$

$$M_{c1} = N_{c1}/n_{c1} = 1{,}00$$

$$M_{d1} = N_{d1}/n_{d1} = 0{,}75$$

$$M_{a1} + M_{d1} = M_{c1}$$

tangential forces:

$$P_{a1} = M_{a1}/r_{a1} = 0{,}25$$

$$P_{c1} = 2\,P_{a1} = 0{,}5$$

$$P_{d1} = P_{a1} = 0{,}25$$

On the minimal ratio of 4th speed the power transmitted on mechanical split-line:

$$N_{a1} = 0{,}25 \cdot N_M$$

$$N_{a2} = 0{,}27 \cdot N_M$$

$$N_{a1} + N_{a2} = 0{,}52 \cdot N_M$$

the power transmitted in the hydraulic split-line:

$$N_{d2} = 0{,}48 \cdot N_M$$

so

$$N_M = N_{c1} = N_{a1} + N_{a2} + N_{d2}.$$

The end-ratio of the variable 4th speed equals the ratio of the 5th speed $i_5 = 1$, just that its connecting is different and the transmission is a mechanical one. The shifting of the 5th direct speed is carried out by locking the K multi-disc clutch that is built in between the c_2 arm and d_2 annulus of the II. planetary gear. At the same time the operation of the second hydraulic motor is stopped because there is no need of further acceleration of the planetary gears. In direct transmission the whole planet-system rotates with the same revolutions as the motor. The direct speed practically has no losses. The velocity and revolution scheme is the same as the one shown in Fig. 6. The torque transmitted by the K clutch $M_K = 0{,}48\,M_M$, its locking happens slipless as at the end of the 4th speed the revolutions of the c_2 arm and d_2 annulus are equal $n_{c2} = n_{d2} = n_m$.

The speed changing range of the variable speed hydromechanical transmission can be extended even beyond the 5th (direct) speed also to the overdrive. In such a case the coupling of the planetary gears is the same as that of the 4th speed, *i. e.* its controlling, too. At the overdrive one must only care for the ability of the 2. hydraulic motor to make rotate the d_2 annulus of the II. planetary gear with a higher r. p. m. than at which the motor rotates.

Fig. 7 shows the unified velocity and revolution scheme in overdrive-speed.

The ratio of overdrive is chosen in such a way that the speed of the vehicle shall be 25% higher than in the 5th (direct) speed. Then the accelerating factor of the I. planetary gear is $x_1 = 5$ and the ratio of overdrive:

$$i_{\text{fast}} = \frac{i}{x_1} = \frac{4}{5} = 0{,}8$$

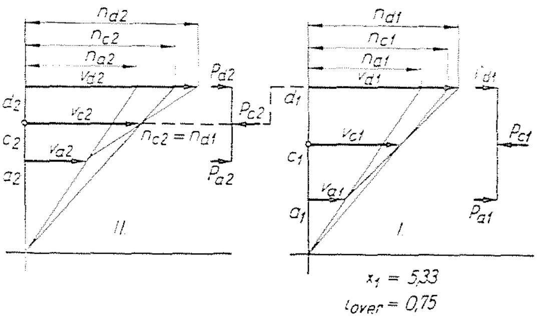

Fig. 7

The mechanical relations of the II. planetary gear in overdrive are: tangential velocities:

$$V_{a2} = 1{,}66\,;\; V_{c2} = 1{,}87 \text{ and } V_{d2} = 2{,}4$$

revolutions:

$$n_{a2} = 1\,;\; n_{c2} = 1{,}33 \text{ and } n_{d2} = 1{,}52$$

(n_{d2} is at the same time the r. p. m. of hydraulic motor)

tangential forces:

$$P_{a2} = 0{,}133,\; P_{c2} = 0{,}266 \text{ and } P_{d2} = 0{,}133$$

torques:

$$M_{a2} = 0{,}2\,;\; M_{c2} = 0{,}6 \text{ and } M_{d2} = 0{,}4$$

$$M_{c2} = M_{a2} + M_{d2}$$

power outputs:

$N_{a2} = M_{a2} \cdot n_{a2} = 0{,}2$ inducted in mechanical split line

$N_{c2} = M_{c2} \cdot n_{c2} = 0{,}8$ hydromechanical

$N_{d2} = M_{d2},\; n_{d2} = 0{,}6 \qquad N_{c2} = N_{c2} + N_{d2}$

N_{d2} is the power output of the hydraulic motor.

The mechanical relations of the I. planetary gear in overdrive: tangential velocities:

$$V_{a1} = 1;\ V_{c1} = 2,5 \text{ and } V_{d1} = 4$$

revolutions:

$$n_{a1} = 1,\ n_{c1} = 1,25 \text{ and } n_{d1} = 1,33 \text{ (the same as } n_{c2})$$

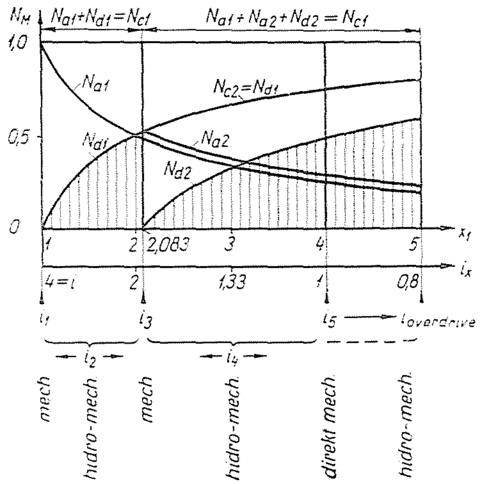

Fig. 8

tangential forces:

$$P_{a1} = 0,2;\ P_{c1} = 0,4 \text{ and } P_{d1} = 0,2$$

torques:

$$M_{a1} = 0,2,\ M_{c1} = 0,8 \text{ and } M_{d1} = 0,6 \text{ (the same as } M_{c2})$$

$$M_{c1} = M_{a1} + M_{d1}$$

power outputs:

$N_{a1} = 0,2 \cdot N_M$ inducted in mechanical split line

$N_{c1} = 1,0 \cdot N_M$ hydromechanical

$N_{d1} = 0,8 \cdot N_M$ hydromechanical ($N_{d1} = N_{c2}$)

The driven power output is the sum of the power inducted by the hydraulic motor on the two sun-gears a_1, a_2 in mechanical split-line, and on the d_2 annulus of the II. planetary gear:

$$N_{c1} = N_{a1} + N_{a2} + N_{d2}\,; \; N_{c1} = N_M.$$

In the power-splitting diagram of the transmission the whole revolution range will be seen, beginning from the 1st speed up to the overdrive. On the ordinata axis the power outputs are measured, on the abscissa axis the x_1 acceleration factors, respectively the corresponding i_x ratios. From the power diagram the power transmitted on each element of the planetary gears is to

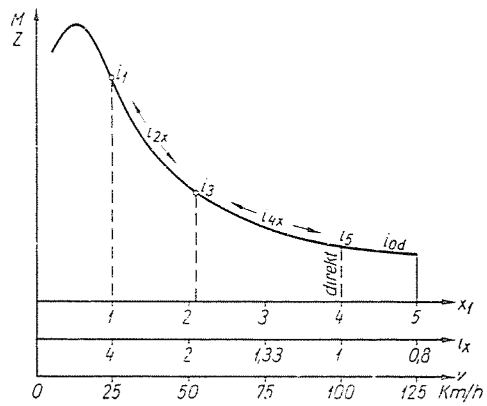

Fig. 9

be established at every ratio. It is also to be seen that in the whole revolution range of the transmission the mechanically transmitted power output surpasses the hydraulic one. In the diagram each speed-grade is pointed out, especially the 1st, 3rd and 5th (direct) speeds of purely mechanical power transmission. They are to be operated even then, when the hydraulic system is unable to operate on account of a defect.

In Fig. 9 the torque respectively the tracting force diagram of the transmission is shown as a function of the speed. Passing over the losses, this curve is identical to the ideal tracting-force diagram. The tracting-force diagram made regarding the losses — as is later stated by calculations too — does not essentially differ from it. In the following table the working (+) and not working (—) elements are compiled in each speed. Here all possible speed-changings are shown, the mechanical and the hydraulic ones as well.

	I. planet	F_1	II. planet	F_2	$H_{S_1} \cdot H_{M_1}$	$H_{S_2} \cdot H_{M_2}$	K	Power transmission
$i_1 = 4{,}0$	+	+	—	—	—	—	—	mechanical
$i_2 = 4—1{,}9$	+	—	—	—	+	—	—	hydromech.
$i_3 = 1{,}9$	+	—	+	+	—	—	—	mechanical
$i_4 = 1{,}0—1$	+	—	+	—	—	+	—	hydromech.
$i_5 = 1{,}9$	—	—	—	—	—	—	+	mechanical
$i_{fast} = 0{,}8$	+	—	+	—	—	+	—	hydromech.

If we want to reach the highest velocity of the vehicle in a short time we continuously accelerate from the start, we only let the continuous variable hydromechanical 2nd and 4th speeds operate, at the end we shift to a direct drive.

	I. planet	F_1	II. planet	F_2	$H_{S_1} \cdot H_{M_1}$	$H_{S_2} \cdot H_{M_2}$	K	power transmission
$i_2 = 4—1{,}9$	+	—	—	—	+	—	—	hydromech.
$i_4 = 1{,}9—1{,}0$	+	—	+	—	—	+	—	hydromech.
$i_5 = 1{,}0$	—	—	—	—	—	—	+	mechanical

The transmission overdrive is used only in exceptional cases, as the 5th (direct) speed works with the highest efficiency. The mechanical efficiency of the transmission is higher than that of the hydraulic ones. The simple planetary gears built into the transmission operate with excellent mechanical efficiency, as the rolling-power of gears equals the transmitted power. It is the same in the case when both of the planetary gears operate together, even then a blind-power is not created. The mechanical efficiencies are to be easily established from the power-splitting diagram shown in Fig. 8. The mechanical efficiency of one working planetary gear makes 95%, two items working together give 90%, let us take the efficiency of the hydraulic aggregate in average as 78%.

In the 1st speed only the I. planetary gear operates, the power-transmission is mechanical, so the efficiency of one operating planetary gear will be $\eta_1 = 95\%$.

In the variable 2nd speed only the I. planetary gear operates but in a double transmission. At the beginning of the 2nd speed when $i_{2\max} = 4 \; (= i_1)$ the transmission is entirely mechanical the efficiency of one working planetary gear is $\eta_{2max} = 96\%$. The 1st hydraulic motor drives the d_1 annulus faster and faster in order to continuously diminish the ratio. Therefore the power transmitted in hydraulic split-line grows more and more while the power output mechanically conveyed on the a_1 sun-gear diminishes.

The sum of both equals the power of the motor

$$N_{a1} + N_{d1} = N_M \quad (= N_{c1}).$$

The highest hydraulic power transmission is at the end of the 2nd speed when the ratio is $i_{2min} = 1{,}9$ $(= i_3)$, $N_{d1} = 0{,}52 \cdot N_M$ and the mechanical power-transmission is $N_{a1} = 0{,}48\ N_M$.

The efficiency of the hydraulical part is:

$$0{,}52 \cdot 78 = 40{,}5\%.$$

The efficiency of the mechanical part is:

$$0{,}48 \cdot 96 = 46{,}0\%.$$

The summarized efficiency is:

$$\eta_{2min} = 86{,}5\%.$$

The ratio of the 3rd speed is $i_3 = i_{2min} = 1{,}92$. In the power transmission both the planetary gears operate purely mechanically.

The summarized efficiency of both planetary gears is $\eta_3 = 92\%$.

At the beginning of the 4th speed the ratio is $i_{4max} = i_3$. Then both the planetary gears still operate mechanically, so the summarized efficiency is $\eta_{4max} = 92\%$ $(= \eta_3)$.

In order to diminish the ratio continuously, the d_2 annulus is driven by the 2nd hydraulic motor faster and faster until it reaches the revolutions of the motor. So the ratio is $i_{4min} = 1$ $(= i_5)$ and the hydraulic power transmission part becomes maximal $N_{d2} = 0{,}48 \cdot N_M$.

Mechanical part of transmitted power is	$N_{a1} + N_{a2} = 0{,}52\ N_M$
Power transmission of hydraulic part	$0{,}48 \cdot 78 = 37{,}5\%$
Power transmission of mechanical part	$0{,}52 \cdot 92 = 48{,}0\%$
Summarized efficiency	$\eta_{4min} = 85{,}5\%$.

The ratio of the 5th (direct) speed is $i_5 = 1$. Then the planetary gears rotate together without roll-down because K clutch locks them. The mechanical efficiency is theoretically $\eta_5 = 100\%$.

In overdrive both planetary gears operate and the power transmitted in the hydraulic line reaches its maximum $N_{d2} = 0{,}6\ N_M$, the mechanical part is $N_{a1} + N_{a2} = 0{,}4\ N_M$.

Power transmission of hydraulic part	.....	$0{,}6 \cdot 78 = 47\%$
Power transmission of mechanical part	...	$0{,}4 \cdot 92 = 37\%$
Summarized efficiency		$\eta_{fast} = 84\%$

The calculation detailed above justifies the mechanical efficiency if the "Jupiter" continuous variable hydromechanical speed transmission is higher

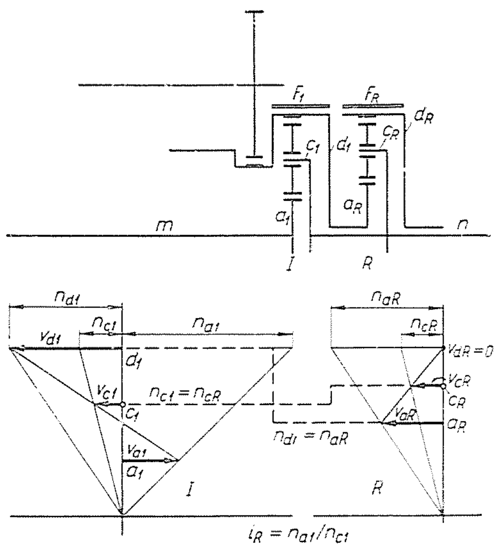

Fig. 10

than that of the purely hydraulical system. $\eta_{\min} = 84\%$ and $\eta_{\max} = 100\%$. But if the examination is extended to the operation in traffic, too, the result will be still better. As in the 3rd speed with 92% efficiency and especially in the 5th speed with 100% efficiency a very large time of operating is needed while in the variable 2nd and 4th speeds the vehicle operates in a comparatively shorter time — ignoring a few specially long slopes — these two variable speed ranges are generally used for effective acceleration of the vehicle.

Based on the above-given considerations the mechanical efficiency of "Jupiter" transmission in traffic operation is to be estimated on an average at 96%. The diagram of mechanical efficiency of the transmission is to be seen in Fig. 11. The operating time of each speed is not shown there as they are considerably influenced by the type of the vehicle, its specific HP-weight and numerous other operating circumstances. In case of one speed-step the reverse gear of "Jupiter" transmission can be solved according to Fig. 10.

In this case the d_1 annulus of the I. planetary gear is connected to a_R sun-gear of the reverse planetary gear while its arm c_r is coupled to n secondary shaft and the d_R annulus is locked by F_R brake. The operation of the reverse speed is mechanical.

The reverse speed of vehicles running in both directions (special motor-vehicles, motor-railcars, motor-locomotives) is solved by a spur-gear or a bevel-gear reversing mechanism coupled after the transmission.

The technical exposition of "Jupiter" transmission detailed above refers to a mechanism of a lower speed-range. The torque amplifying from the

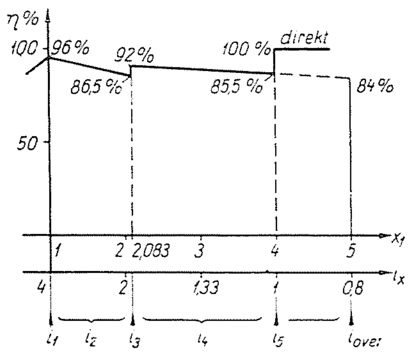

Fig. 11

primary shaft to the secondary shaft was just 4 times as much till the direct speed, and regarding the overdrive ratio, too, it was 5 times as much. That lower speed-range is principally used on auto-cars and motor rail-cars.

The "Jupiter" transmission is able to operate in an essentially wider speed-range too.

The speed-range of economical working is determined principally by the use of simple planetary gears and by the growing of the hydraulically transmitted part of power.

The power output of hydraulic motor is introduced to the planetary gears through annulus gears, the sun-gear is driven by the motor in all cases. The power mechanically transmitted on the sun-gear (*e.g.* at a torque-amplifying 6 times as much) is to reach with two coupled planetary gears with $\sqrt{6} = 2{,}45$ times as much accelerating one by one:

$$N_a = N_M/x = N_M/2{,}45 = 0{,}41 \cdot N_M$$

further the power output transmitted hydraulically is:

$$N_d = N_M \left(1 - \frac{1}{x}\right) = N_M \left(1 - \frac{1}{2{,}45}\right) = 0{,}59 \cdot N_M$$

that is the 59% of the motor power.

As we want to get the possibly highest mechanical efficiency, the power transmitted hydraulically must be minimized as much as possible or operated at such a speed-range in which the vehicle is only running for a short time.

It refers especially to the starting and commence-accelerate working period of heavy vehicles where the impulse of the slow running vehicle is small, so changing to the next higher speed-ratio causes difficulties. Based upon these considerations the entirely continuous variable speed operation of "Jupiter" transmission is limitable to a lower speed-range too. The changing of one or more mechanical speed-grades applied in the higher speed-range does not make any difficulty, and so the speed-range is to be enlarged economically even to 1 : 10. On city-autobuses — with this transmission — the frequent speed-changing is left out which is an advantage not only for the motor and for the car-driver but it increases the velocity, the security and the economy to a very high degree.

Summary

The above-given paper explains the action of a new dual-planetary geared transmission. It solves the problem of variable speed-changing by hydrostatical motors operating in split-power.

With this method the mechanical power-transmission surpasses the hydrostatic transmission over the whole range that assures a very high mechanical efficiency for the transmission. The shifting of the two epicyclic gear-trains makes it possible, that at certain points of the whole velocity range the transmission can be exclusively mechanical. So the 1st, 3rd and 5th (direct) speed ratios operate mechanically and as a consequence of the power-splitting the maximal power transmitted hydrostatically is not more than the half of the output of a driving engine.

The "Jupiter" transmission of variable gear-ratio was planned for various automobiles, railcars and locomotives.

The author (inventor) made an application on the transmission by the Hungarian Patent Office under № 9886.

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