Hydro-static Torque Converter for Motor Vehicles

(Full range and fully automatic)*

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This report refers to a hydro-static power transmission system for passenger cars, busses, motor-lorries and the like. Its operating characteristics are the following abilities: (1) to deliver steplessly variable output speed-ratios ranging from under-drive to over-drive in forward direction and also under-drive in backward direction; (2) to change automatically the speed-ratio while keeping the engine running at the minimum specific fuel consumption; (3) to develop a constant driving speed by automatic speed-ratio control; (4) to produce maximum output power for driving the engine at a given throttle opening, affording maximum acceleration at starting, or passing ahead, and also hill-climbing with maximum possible speed; (5) to set free hydro-static coupling at idle engine running or at inertia vehicle running; (6) to secure variable speed-ratio so as to keep constant the engine speed affording the braking power effectively at all car speeds. These functions may be achieved by the combination of manual manoeuvre and automatic control.

1. Introduction

Hydro-static torque converters ordinarily used in motor vehicles are able to run in the under-drive range up to the directly-coupled drive. For the over-drive and reverse drive ranges, additional gear trains are to be called upon. The de-clutching is obtained by the slip running. The stepless increasing of running speed is fulfilled with lower efficiency, requiring another reduction gear train to overcome increased road resistance.

The hydro-static torque converter, reported here, can afford stepless speed control for full range, namely, under-drive, direct-coupling and over-drive ranges. By handling a single control lever, driving forwards, declutching (neutral) and also driving backward can be brought out positively. Thus, the driving engine may run at ease, resulting in complete combustion with clean exhaust and much lower rate of fuel consumption.

2. Construction

As shown in Fig. 1, the engine torque is coupled with the input shaft I driving the input pump P at its end. In the under-drive range of forward drive, most part of the input power is directly transmitted to the output shaft D. The working fluid is pumped out at the speed ratio $1-x$, where x denotes the speed ratio of the output shaft to the input shaft. The working fluid is supplied to the control motor

Fig. 1 Full-range steplessly variable speed hydro-static torque converter (without reverse gear)

M converting the pumping power into the driving power, which is transmitted to the output shaft D through the clutch F_1 and then to the output casing C. Thus, we see the torque converter consists of the input pump P and the control motor M, both being of axial plunger type. In the under-drive range, M and C are coupled together, and in the over-drive range, the control motor M is coupled with the input shaft through the clutch F_2 whereas the clutch F_1 is set free. Thus, the input pump is actuated as a driving oil motor, resulting in higher speed of the casing C together with the output shaft than the input shaft, and consequently the speed ratio x becomes more than unity.

The input pump P works at the running speed ratio y to the input shaft which is negative in the under-drive range and posi-

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tive in the over-drive, as shown in Fig.1. The speed ratio z of the control motor M increases up to unity in the under-drive range, which remains constant in the over-drive range. The swash-plate inclination θ and the ratio $e/l = \sin \theta$ range from negative to positive value, as shown in Fig.1. The positive and negative pressure sides in the working fluid circuit remain unaltered, but the direction of the fluid flow is reversed.

In the backward drive, the control motor is driven by the input shaft working as an oil pump; and the working fluid is supplied to the input pump to work as an oil motor running in reversed direction, the speed ratio y being less than -1 . As shown in Fig. 3, the reverse drive may be achieved in another way, namely, the input shaft is driven in negative direction by the sun-and-planetary gear, and the driving torque is mechanically transmitted to the output shaft with reduced speed ratio, the hydro-static drive being left open. Thus, the power transmission efficiency in reversed drive becomes much higher in mechanical drive instead of hydro-static drive (See Fig.2).

3. Power Transmission Efficiency

Let ω denote the angular velocity of the input shaft, T_1 the input torque, η_p the hydraulic power efficiency for the combination of the input pump and the control motor, T_0 the output shaft torque, T_p the pump torque, T_m the control motor torque, η_m the mechanical efficiency of the power transmission as a whole and x the speed ratio of output shaft to input shaft. The output power is given by $T_0\omega x$ and the input power by $T_1\omega$. Then, we get for the over-drive range

$$T_0\omega x = \eta_m \{T_1\omega - T_p \cdot (x-1)\omega \cdot (1-\eta_p)\}$$

which gives

$$\eta = \frac{1}{1/\eta_m + (1-1/x)(1-\eta_p)} \quad (1)$$

where the ratio $\tau_p = T_p/T_1 = \eta/x$.

Similarly, we get

$$\eta = \frac{1}{1/\eta_m + (1-x)(1-\eta_p)} \quad (2)$$

for the under-drive range.

In the reverse drive, the pump $T_m\omega\eta_p$ is recovered to drive the output shaft in the backward direction amounting to $T_p(1+|x|)\omega$, where $|x|$ denotes the absolute value of negative speed ratio. Thus, we get

$$\tau_m\eta_p = \tau_p(1+|x|)$$

The output power $T_0|x|\omega$ is equal to $\{T_1\omega - T_m\omega(1-\eta_p)\}\eta_m$ and we get

$$\eta = \tau_0|x| = \{1 - \tau_m(1-\eta_p)\}\eta_m$$

$$= \{1 - \tau_0(1-\eta_p) \frac{1+|x|}{\eta_p}\}\eta_m$$

$$= \{1 - \frac{1+|x|}{|x|} \frac{1-\eta_p}{\eta_p}\}\eta_m$$

which gives

$$\eta = \frac{1}{1/\eta_m + (1/\eta_p - 1)(1/|x| - 1)} \quad (3)$$

Put $\eta_p = 0.70$ and $\eta_m = 0.95$, and we get the efficiency curves as fine full lines in Fig. 2, whereas the thick full lines indicate the experimental results obtained from the actual torque converter. The differences may be due to the variable values of η_p and η_m instead of constant values in the calculated evaluation.

Fig. 2 Power transmission efficiency

4. Change-over Clutch

When the speed ratio x approaches unity, namely, direct-drive, the peripheral speeds of the dog-clutches F_1 and F_2 equally approach each other, resulting in easy change-over by manual or remote control, as shown in Fig. 4.

5. Stepless Speed Control

As shown in Fig. 1, 3 or 4, the speed ratio is controlled by changing the inclination angle θ of the swash-plate actuated by the working oil pressure through the servo-valve. Approaching the position of direct-drive, the swash-plate assumes vertical position and the distance e becomes null and the tip of the actuator piston rod pushes the vertical crosshandle to the left to push down the bellow 2. Then, the inside working oil will be forced out to push the change-over clutch to the left connecting the control motor to the input shaft. Thus, the over-drive running may be started. By and by, the top of the cross-handle may slip out of contact with the tip of the actuator piston rod, and the bellows 2 at the left begins to assume the natural height. Similar actuation will take place in reversed direction, namely, from over-drive to under-drive.

Fig. 3 Full-range steplessly variable speed hydro-static torque converter (with reverse gear)

Fig. 4 Change-over clutch for speed ratio control

Fig. 5 General arrangement

6. Automatic Speed Ratio Control

As shown in Fig. 5, the positions D, R, N and P of the control lever correspond to the positions of the control servo-valve. At the neutral position N, the high pressure side and the low pressure side of the working oil circuit is neutralized through the by-pass valve (Fig. 12), resulting in declutched state. Forward or backward drive will be started by transferring the control lever to the corresponding positions D or R. For the sake

of smooth starting, the by-pass clutch valve is applied.

Push the brake pedal, then engine-brake function will take place at first, ensuring wheel braking by degrees.

Surplus engine output may be expected by pushing down the accelerator pedal, but the usual economical engine running will be restored by releasing the pedal. This economical running of the engine is expected to adjust the speed ratio to match the engine output with the road resistance power, keeping the least fuel consumption rate. Thus, the engine is kept running at ease, resulting in clean exhaust.

7. Pre-set Working Oil Pressure

The governor oil pressure is produced by shunting the pre-set working oil pressure. This working oil is made of use, at the same time, to feed the working fluid to the main hydro-static circuit for the supplementary feed to compensate for the leakage. As shown in Fig. 6, the flow rate Q_s from the supply pump (shown in Fig. 5) is shunted to return, amounting to Q_r . The difference $Q_a = Q_s - Q_r$ is expected to establish the constant oil pressure designed. The amount of Q_r is controlled by the relief valve, which is positioned by the mutual action between the supporting spring and the working oil pressure on the bearing surface of the plunger. Cavitation may be prevented by the suction of the supply pump.

Fig. 7 shows the characteristics of

Fig. 6 Relief valve for pre-set pressure

Fig. 8 Discharge from relief valve $Q_s - Q_r$

the relief valve. Let x denote the displacement of the relief valve, a the flow area which depends upon the value of x , C the discharge coefficient, ϕ the inclination of flow, k the spring constant, x_0 the initial deflection of the balance spring and A the area subjected to the oil pressure. Then we have the following relations:

$$P \cdot A = \frac{\gamma}{g} Q v \cot \phi + k(x_0 + x)$$

$$Q = C \cdot a \cdot \sqrt{2g \frac{P}{\gamma}}$$

where γ stands for the density of oil and the flow velocity is given by

$$v = Q/a$$

Combining these relations, we get the oil pressure as follows:

Fig. 7 Characteristics of spool valve

Fig. 9 Pre-set pressure versus engine speed

$$P = k \cdot (x_0 + x) / \{ A - 2C^2 a \cdot \cot \phi \}$$

Thus, we see x and P are increased in proportion to Q , as shown in Fig. 7.

The pre-set oil pressure P is to be constant independently of the flow rate Q proportional to the running speed. Such requirement was realized by the design shown in Fig. 6. The main improvements are (1) the throttle in front of the surface subjected to the oil pressure reduces or suppresses the vibration of the relief valve provoked by the pulsation of the pressure from the supply pump; (2) utilization of the input shaft governor pressure checks the increase of the pre-set oil pressure in proportion to the input shaft speed and saves the input energy to drive a special oil pump; (3) switching the drain from the relief valve to the suction of the supply pump suppresses the cavitation provoked at higher driving speed of the supply pump and reduces the driving power by the effect of supercharging of the supply pump.

Experimental results are shown in graphs Q_a and Q_s in Fig. 8, which enables us to find the amount of the return oil Q_r . Fig. 9 shows the results of such betterments, the deviation of the pre-set pressure being reduced to ± 1 kg/cm² for the designed pressure of 7 kg/cm².

8. Governor Pressure of Input and Output Shafts

The governor pressure of the input shaft corresponds to the running speed of the driving engine and is used (1) to actuate on-off the by-pass valve for starting; (2) to actuate on-off the band brake for reversed drive (see Fig. 4); (3) to actuate the relief valve for the pre-set oil pressure (see Fig. 6). The governor pressure of the output shafts is used (1) to indicate the point of direct drive and (2) to actuate the pilot valve for the speed ratio controller (see Fig. 4).

As shown in Fig. 10, the oil pressure p_s is throttled down to p_g and then to atmospheric pressure $p_0 = 0$ through the variable throttle. Then we have

$$c_o a_o \sqrt{2g \frac{p_s - p_g}{\gamma}} = c_x a_x \sqrt{2g \frac{p_g - 0}{\gamma}}$$

where c stands for the discharge coefficient, a the flow area and γ the specific weight of the working oil. Thus, we get

$$p_g = p_s / \{ 1 + (\frac{c_x \cdot a_x}{c_o \cdot a_o})^2 \}$$

As shown in Fig. 10b, the centrifugal force $\frac{W}{g} r \left(\frac{2\pi N}{60} \right)^2$ is balanced with the fluid pressure $A \cdot p_g$, which is utilized for the actual governors of input and output shaft. Fig. 11 shows that the governor pressure increases with engine speed, and the input or output shaft speed. The governor pres-

sure for the input shaft varies in two steps, namely, due to the weight W_2 in usual running and W_1 in starting to actuate the by-pass clutch valve accurately.

Fig. 10 Principle of governor pressure

Fig. 11 Governor pressures

9. By-pass Clutch Valve

By-pass device between the higher and lower pressure sides in the hydrostatic torque converter circuit is utilized for idle running and smooth starting. Fig. 12a shows such a by-pass clutch valve. The governor pressure of the input shaft pushes the valve to the left. The left end of the spring is seated on the secondary valve supported by the pre-set pressure to the right seating on the stopper. The axial distribution of the oil pressure, as shown in Fig. 12c, along the hollow passage of the by-pass valve results in the effect of the by-pass valve being pushed to the left. At lower running speed

Fig. 12 By-pass clutch valve

of the input shaft or engine shaft, the by-pass valve is pushed to the right and the high pressure working fluid flows out freely into the lower pressure side. When the speed of the input shaft is increased up to 600~900 rpm, the by-pass valve begins to close halfway and consequently the hydro-static torque converter works as half-clutched. At the engine speed 1600 rpm, the governor pressure pushes the by-pass valve to the left against the pressure on the secondary valve and the hydro-static torque converter begins quickly the usual working. The working position of the by-pass valve is clearly shown in Fig. 13.

When the control lever is shifted to the neutral position N from the driving position D (see Figs. 12a and 4), the pre-set pressure pushes the secondary valve to the right as far as the stopper and, consequently, the by-pass valve is opened and the hydro-static torque converter ceases its function, namely, it is de-clutched.

To start steep climb, the driver sets the control lever to the position D' instead of N (see Fig. 4), then the declutching state will continue up to a higher running speed and consequently a stronger engine torque may be generated. Then, the clutch lever is set back from D' to D to clutch on. Thus, the car starts with a much stronger engine torque.

Fig. 13 Design of by-pass valve

10. Control of Speed-ratio

As shown in Fig. 4, the actuator piston controls the speed ratio α by titling the swash-plate in order to regulate the plunger stroke of the control motor. The pilot piston is subjected to the governor pressures, one on the left side and the other on the right side, and the pilot piston will take the position corresponding to the speed difference between the input and output shafts. Thus, the servo-piston will be displaced to supply pressure oil to the corresponding side of the actuator piston, resulting in a change of the inclination of the swash-plate. Soon after, the servo-piston is displaced back by the motion of the actuator piston to assume the neutral position. Thus, the supply of the working pressure oil is cut off to the actuator piston, and the swash-plate will assume a new position and the corresponding new speed ratio will be maintained. For the sake of quick response, the pressure oil is supplied from the higher pressure side of the working fluid circuit of the hydro-static torque converter, which remains unchanged both in under-drive and over-drive running. The interconnecting lever between the pilot piston and the intermediate rod is bent, as shown in Fig. 14, instead of being straight. Thus, symmetrical displacement of the pilot valve about the neutral position is secured along the corrected displacement curve represented by a thick line instead of a fine dotted line.

Fig. 14 Correction of actuator pilot valve displacement

11. Inertia Running

During the road driving, the inertia running may be initiated by shifting the control lever from the drive position D to the neutral position N, then the preset pressure oil will flow into the secondary valve chamber. This oil pressure being always higher than the governor pressure of the input shaft, the clutch valve may be pushed to the right (see Fig. 12a). Thus, the high and low pressure sides of the hydro-static circuit are interconnected, causing a declutched state followed by idle running of the car by its inertia power.

12. Engine-braking

Suppose the driver shifts his foot from the acceleration pedal to the brake pedal and presses it by degrees. Then the brake oil pressure generated will push the spool valve to the left halfway, as shown in Fig. 15. Then, the governor pressure of the input shaft will actuate the economical drive piston just in reversed way, closing tightly the throttle valve to the suction of the driving engine. Thus, the engine-braking will take place most effectively. By pushing the brake pedal further, the spool valve is farther pushed to the left, connecting the brake oil directly to the wheel brake cylinders.

When the driver's foot is removed from the brake pedal, the compressed restitution spring will bring back the spool valve to the right end, resuming the ordinary engine driving configurations. The engine-braking power may be kept constant, since the speed ratio α decreases with the decreasing of the car speed, which means much more effective engine-braking can be expected than ordinary engine-braking systems with engine throttling only (Fig. 16).

Furthermore, the driver may push down the throttle pedal, the acceleration of the car will be commenced from reduced speed ratio α , resulting in a smooth and effective

Fig. 15 Engine-brake manipulation

Fig. 16 Effect of engine braking on speed ratio

acceleration. If the driver tilts the speed control lever positively, the effect of the engine-braking will be doubled (see Fig. 4).

13. Reversed Driving

Position the control lever at N and continue idle running for a while, until the dog clutch comes into contact with under-drive side. Shift the control lever from N to R. Then the band brake will be actuated by degrees on the reverse gear. Thus, the car starts gradually backwards.

At the same time, the by-pass clutch, shown in Fig. 12, is shifted to the right by the pre-set oil pressure, which is changed over by shifting the control lever from N to R (Fig. 4), and the pre-set oil pressure shifts the by-pass valve to the right, rendering the hydro-static transmission powerless. The pre-set oil pressure cuts off the supply to the governor of the output shaft and the dog clutch continues to connect the rotor of the control motor to the output casing (Fig. 17).

Fig. 17 Reverse drive

14. Automatic Control for Economical Running

Suppose a gasoline motor car driving with constant opening of the throttle valve. The car speed will be varied by the effect of road resistance between the maximum torque point and the maximum output point. Between these two limits, there will be the maximum economy point, namely, minimum specific fuel consumption point. Such maximum economy points will be found for various openings of the throttle valve. The locus of such points may be called the maximum economy line, along with the maximum torque line and the maximum horse power line.

The required horse power to drive a car is consumed to overcome the air resistance, hill-climbing resistance and acceleration resistance. In order to keep economical running at all driving speeds, the speed ratio x of the propeller shaft speed to the engine speed is to be controlled automatically. Suppose that the engine speed and the throttle opening are kept constant corresponding to the economical running, as mentioned above, the speed ratio x must range from under-drive to over-drive to match the road resistance. Thus, the fuel economy is secured and the engine runs at ease with clean exhaust and longer life. Naturally, the driver may press the throttle pedal further, when the road resistance is increased for hill-climbing or higher speed driving. In any case, the driver may put his foot freely from the throttle pedal, leaving it to the automatic speed ratio control.

Fig. 18 shows the characteristics of a gasoline engine with throttle opening and engine running speed. At the throttle

Fig. 18 Performance of gasoline engine

opening 35° , for example, the engine may run with maximum economy at the speed 2350 rpm, the specific fuel consumption being 220 g/PS_h. It may increase to 285 g/PS_h at the engine speed 4350 rpm with the same throttle opening, which corresponds to the car speed 100 km/h with direct coupling for reduced road resistance. Such road resistance may be met with increase of speed ratio up to $x = 1.98$ with reduced a fuel consumption, reduced by 30%, as shown in Fig. 19.

The automatic speed ratio control mechanism, reported here, is affected by the mutual relation with the car speed given by the governor pressure of the output shaft to the engine throttle opening given by that of the input shaft, and also by the mutual relation of the inclination θ of the swash-plate with the speed ratio x (see Fig. 1). Fig. 20 shows the running characteristics for the same gasoline car above mentioned. Thick lines indicate the mutual relations of the car speed with the throttle opening in order to drive on inclined roads with various car speeds, keeping the economical fuel consumption condition by the automatic speed ratio control. The dotted lines show the running speeds on the same inclined roads with the top, second or third gearing. Referring to Fig. 18, we see the fuel economy cannot be expected from gasoline cars with three speed change gear boxes.

Once the relation between the speed ratio and the economical running is confirmed, further improvements may be expected on the effects of acceleration and hill-climbing feelings. Thus, we get sev-

Fig. 19 Economical running versus speed ratio

Fig. 20 Speed ratio versus fuel economy

Fig. 21 Actual speed ratio control

eral designs for automatic speed ratio control device. An example of such improvements is shown in Fig. 21. The fuel economy depends upon the combination of gearing with the inclination of road and driving car speed, 30 % fuel being saved at running speed 100 km/h on level road.

15. Conclusion

The new hydro-static torque converter, reported here, indicates the driving characteristics as shown in Fig. 22, actual deviation from the economical line may be due to the deviation of pre-set pressure as shown in Fig. 9.

This kind of hydro-static torque con-

Fig. 22 Economical running with hydro-static torque converter

verter with automatic speed ratio control mechanism for fuel economy enables us to run the engine at much lower running speeds, resulting in longer life of the engine itself, clean exhaust and lower specific fuel consumption. Further an automatic speed ratio control mechanism may be designed which can keep maximum engine output for racing cars or maximum engine torque for steep hill climbing cars, if necessary.