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Characteristics of Multiple Range Hydromechanical Transmissions

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IN ORDER TO compete successfully with the commercially established devices for power transmission, any new device which expects to penetrate the market to a large extent must provide some advantages which are not obtainable with the conventional devices.

Hydromechanical transmissions must offer something to the user which is an improvement over devices currently available. What can a hydromechanical transmission offer that others cannot, and what does it mean to the user? Hydromechanical transmissions can be efficient and practical, infinitely variable devices offering the capability of optimizing for minimum exhaust emissions, maximum fuel economy, or maximum power availability.

Advantages to be gained by using infinitely variable hydromechanical transmissions lie in the fact that the powerplant and transmission can be considered as one system, and programmed to operate under the most advantageous conditions. The optimizing benefits can only be realized if the transmission has high efficiency.

The hydromechanical transmission is a split-power-path type device which transmits a portion of the power through a direct mechanical path, and a portion of the power through positive displacement hydraulic motor pumps. The portion of power transmitted hydraulically is determined by the gearing configuration and by adjusting the displacements of the

hydraulic units. Ratio change is accomplished by adjusting the relative displacements and, therefore, the relative speeds of the hydraulic units. The ratio of output speed variation available through a single gear train is a function of the amount of power which can be transmitted hydraulically. Decreased hydraulic power results in decreased range per gear train and, therefore, a larger number of gear trains are required to achieve a given overall speed and torque variation for the transmission.

There are tradeoffs between the improved efficiency and system capability of increased number of ranges on the one hand, and its increased complexity on the other. The more ranges there are, the more planetary gearing and clutches are required. Also, the more ranges there are, the more complicated the controls become. At the same time, the loads on gears and bearings decrease, and the size and/or stress level of the hydraulic units can be reduced.

It can be shown that a hydromechanical transmission can be constructed having gearing and clutch complexity similar to a corresponding powershift transmission.

COMPARISONS

For a qualitative comparison, it can be said that for a multi-range transmission, the gear and clutch section will compare

ABSTRACT

The purpose of this paper is to show the advantages of multirange hydromechanical transmissions and to show some basic relationships of primary design parameters. Hydromechanical transmissions can be designed so that they maintain a high efficiency over a wide range of torque/speed variations. The amount of volumetric loss and, therefore,

slip, can be reduced to a minimum; therefore, it is possible to control the speed of the powerplant by proper ratio control.

The powerplant and transmission can be considered as one system which is permitted to operate, under optimum conditions, irrespective of the road load or speed of the vehicle.

in cost to that of a powershift transmission having a similar number of ranges. The controls will also be similar in cost.

The relative cost of the multirange hydromechanical and a powershift transmission will, therefore, depend on the cost of the two hydraulic units, as compared to a torque converter assembly, with its attendant overrunning clutch and lockup clutch.

The increase in first cost must be offset by more advantageous system capability. Some of these advantages are:

ENGINE SPEED CONTROL - Being infinitely variable and virtually without internal slip, it is possible to program the transmission system so that the engine operates at constant speed or at some number of preselected speeds which can be varied in accordance with the vehicle power demands. For example, a truck system can be programmed to operate the engine at some lower speed during city and cross-country flat-land operation and at rated speed when maximum power is required by the vehicle, such as in mountainous operations. A further refinement of the control system can provide for variation in the transmission ratio such that the engine is operated at whatever speed will give the maximum fuel economy and/or minimum emissions for that particular combination of vehicle speed and power demand. This ability to program engine speed basically independent of vehicle speed is a benefit afforded by none of the previously existing transmission types.

NO INTERRUPTION IN POWER FLOW - As all range changes are stepless and conducted with overlapping clutch sequencing (on-going clutch engages prior to disengaging off-going clutch), there is no interruption in power transmission to the vehicle drivetrain. Not only does this increase drivetrain component life by removing the shock loads imposed by the absorption of engine inertia during shifts, it also frees the vehicle operator from all of the problems associated with rapid variation in torque application during inclement weather conditions. This feature also allows maximum grade start-ups without vehicle roll-back. In fact, the operator can hold the vehicle at rest on a grade without utilizing the service brakes (assuming adequate drive wheel traction) with the engine at idle, due to the infinite ratio feature of the system. In this condition, there is virtually no power requirement on the engine other than that created by the transmission boost pump which is utilized for lube and control functions. Oftentimes, torque converter/powershift transmissions are utilized for holding the vehicle on a grade, but the adverse effect created by the attendant generation of heat is a well-known problem to transmission manufacturers.

FULL ENGINE BRAKING - Maximum engine friction can be obtained for vehicle braking regardless of vehicle speed simply by programming the transmission system to maintain maximum engine speed during the periods of closed throttle operation.

AUXILIARY BRAKING - Auxiliary braking may be achieved by the partial application of nonsynchronous clutches. This requires substantial heat rejection capability and increases the controls complexity.

OPERATIONAL CAPABILITIES

The operational capabilities can provide additional advantages unique to the type of engine system employed.

DIESEL ENGINES - The high cost and complexity of diesel engine fuel systems today are due, in large part, to engine manufacturers' increasing requirements for additional control features, such as torque shaping, variable advance, and aneroid fuel limiting. If the engine is required to operate only at one or two speeds, however, these features can all be greatly simplified or, in most cases, eliminated, with virtually no sacrifice in engine performance or emissions.

On turbocharged engines, the benefits of limiting the operating speed range are even greater. Not only will the turbo-charged acceleration lag be greatly reduced but the turbo-charger can be matched so as to operate at a much higher efficiency than is possible when it is required to operate over the entire engine speed range.

Many of the compromises presently necessitated in engine design, such as valve timing, vibration damping, and combustion chamber design, can also be greatly reduced by the application of a transmission system which will reduce the operating speed requirements.

TURBINE ENGINES - The major advantage obtained when coupling a turbine engine to this type of transmission system is that the required complexity and, therefore, cost of the turbine and controls are greatly diminished. A single shaft turbine becomes feasible for vehicle motive power which, in addition to reducing engine complexity, provides some major operational advantages. Prime among these is the ability to obtain engine braking at a power level in excess of the engine rating. Operational economy can be improved by programming the transmission control for a predetermined schedule of turbine-operation conditions. Since the transmission is capable of controlling the speed of the engine, the fuel control problems are still further simplified.

KINETIC ENERGY DEVICES - Recent studies have indicated that kinetic energy propulsion systems have advantages in mass transportation vehicles. These systems require an infinitely variable, high-efficiency transmission for practicality. Such requirements can be met with multirange hydromechanical transmissions.

SYSTEM EFFICIENCY - To accomplish the advantages just described, the transmission must have a high efficiency so that the potential gains in overall system performance are not offset by internal losses. Although it is unlikely that any transmission which circulates power hydraulically will ever be as efficient as a straight mechanical transmission, the overall engine/transmission system can be made more efficient due to the improved operating conditions afforded the engine by the infinitely variable transmission system, provided the transmission efficiency remains sufficiently high throughout the majority of the vehicle-operating ranges.

Since the efficiency of a hydromechanical transmission depends primarily on the percentage of power transmitted hydraulically, it must be kept to a minimum.

The mathematical relations governing torque and speed

variation are such that the quantity of hydraulic power transmitted is a function of ratio, and increases very rapidly with a small increase in ratio. In order to limit the quantity of hydraulic power for a required overall torque and speed ratio, the transmission must go through several narrow speed and torque variation ranges, with synchronous shifts between ranges to accomplish a stepless variation over the entire range.

For minimum percent of hydraulic power:

1. Each of the several ranges must be equal in ratio. With a given overall torque variation consisting of "n" ranges, each range has to be

$$\sqrt[n]{R_{HM}} \quad (1)$$

2. The shift from one range to another must occur under synchronous conditions, without interruption of power flow.

3. The design must be such as to avoid overspeeds of gears, particularly planet pinions.

NOISE LEVEL REDUCTION - An area which is receiving more attention due to increasingly stringent government regulations is the noise level of any proposed transmission system. The noise produced by hydrostatic transmissions and, to a lesser degree, hydromechanical transmissions, is almost wholly due to the hydraulic system. The noise level varies as a function of the load under which the hydraulic system operates. It is apparent that for any transmission system utilizing hydraulic power, the noise level is a function of the amount of power transmitted hydraulically. With hydrostatic and hydromechanical transmissions which transmit a high percentage of power hydraulically, it is not feasible to operate the hydraulic unit at moderate ratings, nor is it feasible to tolerate any reduction in efficiency. On the other hand, multirange hydromechanical transmissions may be operated at moderate ratings with little penalty in efficiency.

A great many combinations of hydraulic systems and planetary assemblies can be conceived to produce the desired speed and torque variation. It is beyond the scope of this paper to evaluate the relative merits of the hundreds of combinations which have appeared in the patent and technical literature during the past 80 years or so.

This discussion will be limited to one particular type of planetary/hydraulic combination which permits the use of small-size hydraulic units, and affords the simplest possible starting and reverse conditions.

Before going ahead with the description of the three-range transmission, it seems desirable to review the general principles and nomenclature underlying power-splitting transmissions.

The definition of terms and mathematical relationships are herein described.

LOCKED REACTION MODE

In this mode of operation, there is no power transmitted hydraulically, because the variable unit, which is connected

to the input, rotates at zero displacement, and, therefore, prevents the rotation of the fixed displacement unit by blocking its fluid output.

The power is transmitted entirely by the mechanical path. The gear loads and torques which are produced within the planetary assembly are, therefore, a function solely of engine torque and planetary ratios. No torque is added or subtracted hydraulically.

The recirculative mode results in planetary gear loads and torques which are higher than those produced by the engine, and the nonrecirculative mode results in gear loads and torques which are lower than those produced by the engine.

RECIRCULATIVE MODE

During the interval when the power is transmitted in a recirculative manner, the entire engine power is delivered to the planetary. The fixed hydraulic unit, which is connected to the reaction member, now acts as a pump. It delivers hydraulic power to the variable unit, connected to the input upstream of the planetary, which now acts as a motor. Therefore, the hydraulic system adds power to the input of the planetary. This results in increased output torque, and, at the same time, the gear and bearing loads within the planetary are increased above the level produced by engine torque. Thus the planetary system internally recirculates more than the engine input horsepower. Specifically, this equals the engine horsepower plus the percentage of power recirculated to the input hydraulically. The sum of the losses in both systems is therefore greater, and the overall transmission efficiency during this mode of operation is lower than during the nonrecirculative mode now described.

NONRECIRCULATIVE MODE

During the interval when the power is transmitted in a nonrecirculative manner, the total power is divided by the planetary into power transmitted by the mechanical system, and power transmitted by the hydraulic system. Thus, the mechanically transmitted power is less than 100%, and the balance is transmitted hydraulically.

The variable hydraulic unit, which is connected to the input upstream of the planetary assembly, acts as a pump in this mode of operation, and absorbs some of the input horsepower; and the fixed hydraulic unit, which is connected to the reaction member of the planetary, acts as a motor and adds power to the output of the planetary.

The gear and bearing loads in the planetary assembly are, therefore, less than those produced by the engine torque, since the torque is divided into a mechanical and hydraulic path before it reaches the planetary.

Hydrostatic Range - R_H . All power is transmitted hydraulically. Power-splitting planetary operates in a locked condition and is disconnected from the engine. This is torque-limited range, during which the transmission delivers the maximum torque beyond which the wheels will slip. During the constant-torque operation, the horsepower which can be

transmitted by the wheels, therefore, increases from zero at the start of R_H to 100% of engine power at the end of R_H .

Total Hydromechanical Range - R_{HM} . Power transmitted by a combination of mechanical and hydraulic means. R_{HM} may consist of several ranges, R , which may have equal speed and torque variation ratio. R equals the range of output torque and speed variation, during which the variable hydraulic unit goes from full displacement in one direction, to zero, and over center to full displacement in the other direction.

The planetary reaction unit goes simultaneously from maximum speed in one direction, to standstill, and then to maximum speed in the other direction. Each range is further divided into two subranges:

$$R_R \text{ and } R_{NR} \quad (2)$$

where:

Subrange R_R = recirculative, power transmitted by recirculation of hydraulic power

Subrange R_{NR} = nonrecirculative, power transmitted by dividing load between mechanical and hydraulic system.

Relation of Mechanically and Hydraulically Transmitted Portions of Power - The power relations in the recirculative mode are:

$$X_R = \frac{\text{Output speed at end of } R_R}{\text{Output speed at start of } R_R} \quad (3)$$

Total power circulated mechanically within the planetary, including recirculating power:

$$(\text{Input hp}) (X_R) \quad (4)$$

Power recirculated hydraulically within the planetary:

$$(\text{Input hp}) (X_R - 1) \quad (5)$$

The power relations in the nonrecirculative mode are:

$$X_{NR} = \frac{\text{Output speed at end of } R_{NR}}{\text{Output speed at start of } R_{NR}} \quad (6)$$

Power transmitted hydraulically:

$$(\text{Input hp}) \left(\frac{X_{NR} - 1}{X_{NR}} \right) \quad (7)$$

Power transmitted mechanically:

$$(\text{Input hp}) \left(1 - \frac{X_{NR} - 1}{X_{NR}} \right) \quad (8)$$

The maximum power requirement of the hydraulic system is, therefore:

$$(\text{Input hp}) (X_R - 1) \quad (9)$$

At midrange, theoretically, zero horsepower is transmitted hydraulically.

A multirange transmission of three hydromechanical ranges is shown in Fig. 1.

In order to simplify the description of the functions of this device, we will start from point A in Figs. 2 and 3. This mode is the locked reaction mode and occurs at point A when the variable unit is on zero displacement and, therefore, locks the ring gear against rotation in each range. At this point, the output has a speed as indicated by point B, and the entire power is transmitted mechanically.

The gear and bearing loads are, therefore, a function of engine torque and planetary gear ratios, and the entire power is transmitted mechanically.

Since it is desired to vary the output speed, while retaining a constant input speed and horsepower, it becomes necessary to add torque to the planetary when the speed of its output member is decreased with respect to point B, and subtract torque from the planetary when its output speed is increased with respect to that point, because the gearloads within the planetary have a constant relation with respect to each other. Thus, if we wish to increase the torque and decrease the speed of the output carrier, the displacement of the variable hydraulic unit is increased from zero in a direction which allows the planetary reaction unit (which is shown as a fixed displacement unit in the schematics) to rotate under the influence of the drive reaction, and act as a pump.

It supplies fluid under pressure to the variable input unit, which acts as a motor, thus adding its torque to the input shaft. The input shaft now transmits not only the engine torque, but also the added hydraulic torque to the planetary, which for this reason operates at a higher level of load than at point A, and the carrier output torque, therefore, increases with its decrease in speed.

As the displacement of the variable input hydraulic unit is further increased, the planetary reaction unit increases its speed, the carrier output speed decreases, and its torque increases.

The increase in planetary gear loads causes an increase in the output carrier torque (Fig. 2). During this time, the power transmitted by the hydraulic system from the reaction to the input is recirculated within the planetary. This mode of operation is, therefore, called the recirculative mode, and, due to the simultaneous rise in pressure, displacement of the variable input unit, and speed of the fixed planetary reaction unit, it is possible to recirculate more than the input horsepower. This is solely a function of the ratio R_R in the definition of terms. As this ratio increases, the size of the hydraulic unit increases very rapidly, and the efficiency deteriorates.

In going to higher output speed from the locked reaction

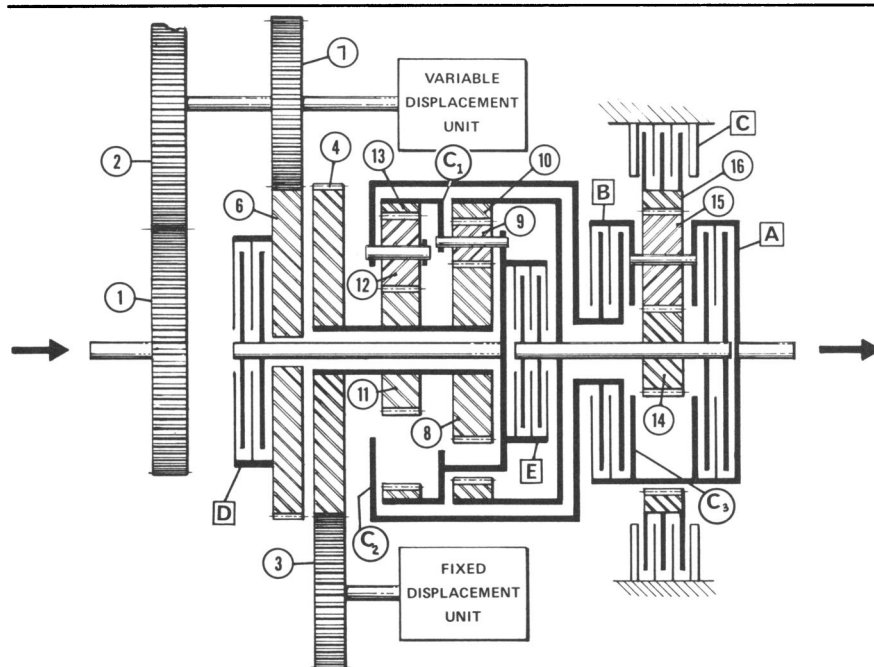


Fig. 1 - Three-range transmission

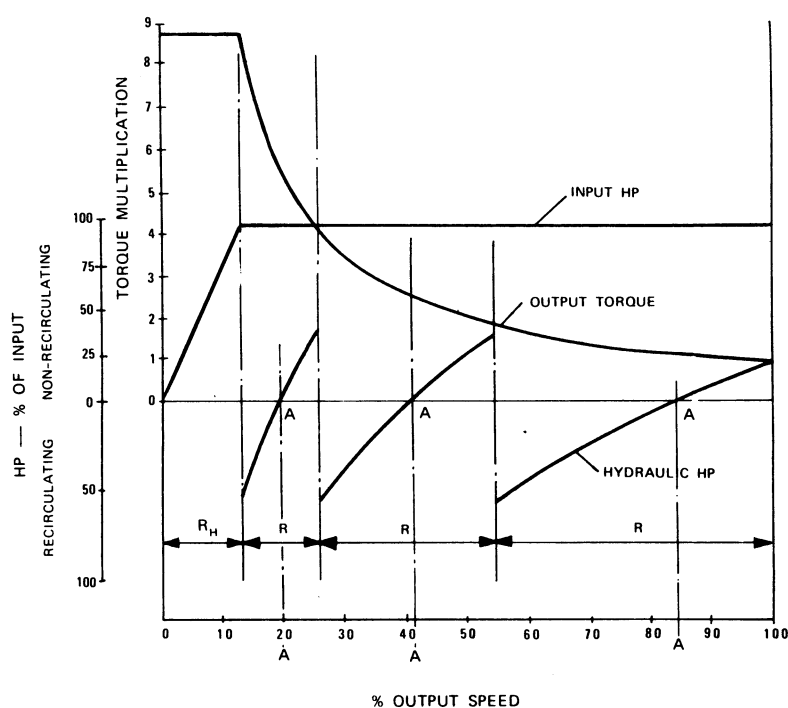


Fig. 2 - Three-range transmission - horsepower and torque at rated input

point, the variable input unit is stroked in the opposite direction and becomes a pump, thus absorbing some of the input torque, and drives the fixed planetary reaction unit which now acts as a motor to increase the speed of the output carrier. The level of gearloads in the planetary is reduced with respect to these values at the locked reaction point.

As the displacement of the variable input unit is increased, the speed of the fixed planetary reaction unit and the output is also increased, until it reaches its maximum design speed in that particular range.

Since the power is divided upstream of the planetary, no

recirculation takes place, and this mode is, therefore, called the nonrecirculative mode, R_{NR} .

The multirange transmission makes it possible to use the efficient part of the variable planetary/hydraulic device over several ranges by synchronous shifts between the ranges. As a result, the efficiency is maintained throughout a wide torque/speed variation, and the size of the hydraulic units is considerably reduced.

Fig. 4 shows a comparison of sizes of hydraulic units and horsepower transmitted hydraulically for transmissions of

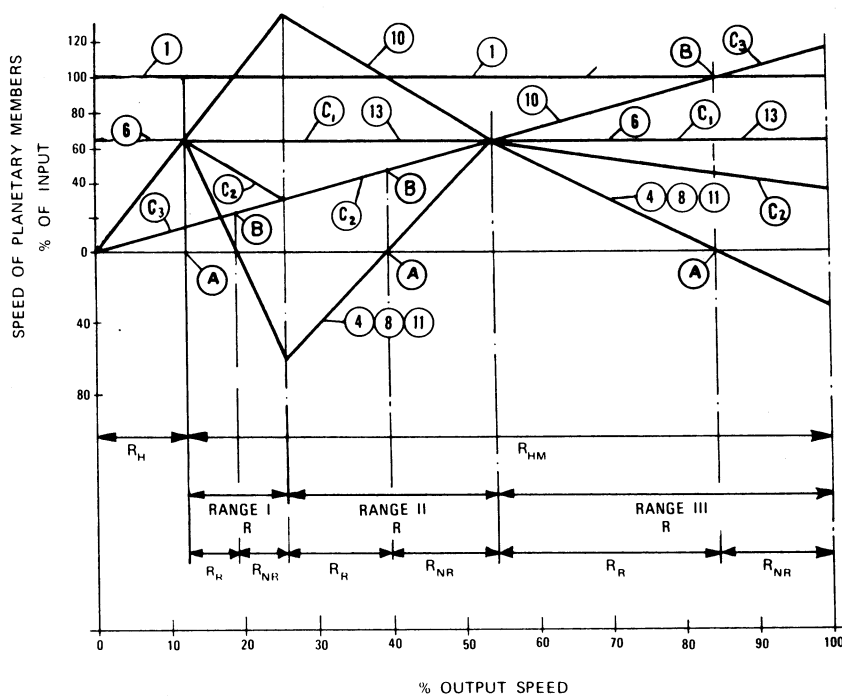


Fig. 3 - Three-range transmission - speeds of planetary elements at maximum input speed

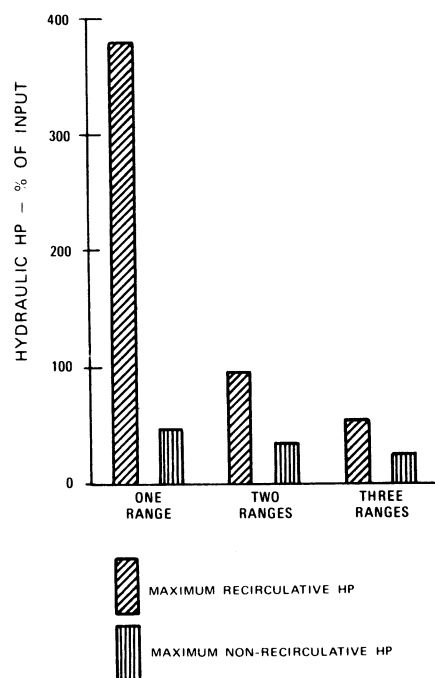


Fig. 4 - Hydraulic horsepower as a percent of input power. Transmission of various ranges, overall ratio 8.6:1

various number of ranges while maintaining an overall ratio of 8.6:1 at full horsepower.

Table 1 gives a description of the specifications of a typical three-range transmission. Transmission consists of hydrostatic range and three hydromechanical ranges. For optimization of performance at maximum speed, the last subrange R_{NR} is shortened so that $R_{HM} = 8.25$ and the last $R_{NR} = 1.18$, as shown in Table 2.

In starting from standstill, or in reverse, the transmission

Table 1 - Specifications

Input, hp	350
Input, rpm	2400
Torque multiplication ratio	8.6:1
Maximum output speed, rpm	2800
Speed variation, rpm	
Forward	0-2800
Low speed, reverse	0-350
High speed, reverse	0-1540

Table 2 - Ratios

R	2.13
R_R	1.565 max recirculated hydraulic hp, 56.5%
R_{NR}	1.36 max nonrecirculated hydraulic hp, 26.5%
R_{NR}	1.18 at top speed, nonrecirculated hydraulic hp, 15.3%

operates in a hydrostatic mode. Two output speeds are available. In the low-speed hydrostatic range, the engine drives gear 1, which drives the variable displacement unit acting as a pump through gear 2. Clutch D is disengaged; thus, no mechanical drive is transmitted to the planetary by gear 6. The fluid under pressure is delivered to the fixed unit, acting as a motor, which drives the planetary assembly through gears 3 and 4. Clutch E is engaged, and locks ring gear 10 to carrier C_1 , thus locking the entire planetary assembly against relative rotation. The planetary rotates as one part, and drives sun gear 14, which drives the output carrier C_3 at reduced speed, because clutch C, which is engaged, holds ring gear 16 against rotation.

The output, therefore, operates at low speed and maximum

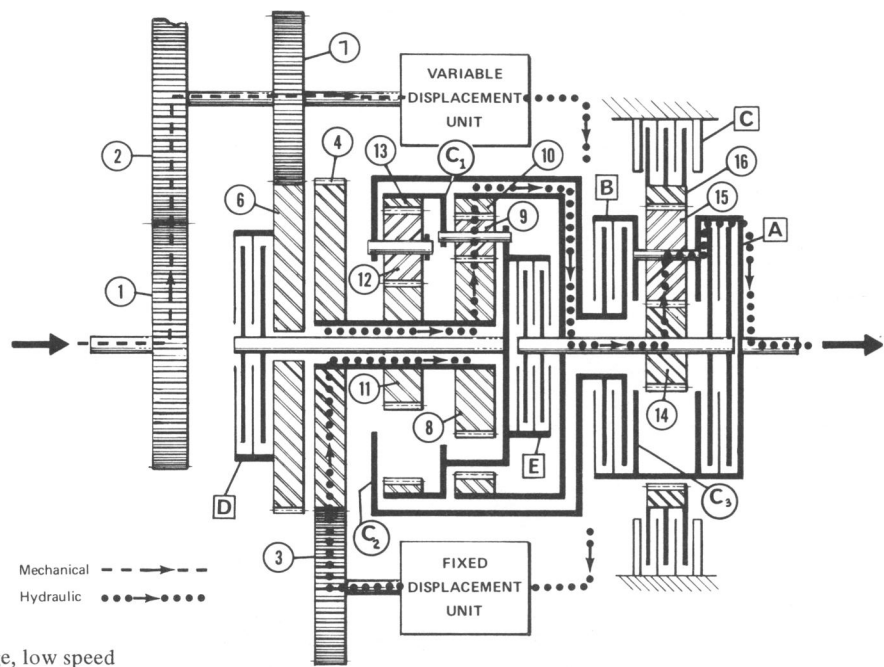


Fig. 5 - Three-range transmission - hydrostatic range, low speed

torque. Starting from zero output speed, the displacement of the variable unit is increased from zero in either direction, and the speed of the output shaft varies proportionately to the displacement in either direction (Table 3 and Fig. 5).

If it is desired to operate at higher reverse speed in hydrostatic range, the drive is not taken through gear 14, but instead through carrier C_2 and clutch B. This eliminates the reduction afforded by the planetary, which consists of gears 14-16, and drives the output shaft directly at reduced torque (Table 4 and Fig. 6).

When the speed in low-hydrostatic range reaches its maximum, and it is desired to go to higher output speed, the transmission shifts from hydrostatic to first hydromechanical range. At the top speed in the hydrostatic range, the locked planetary assembly reaches a speed such that it is possible to engage clutch D at synchronous speed, and immediately thereafter disengage clutch E. The transmission now is in the recirculative portion R_R of the first hydromechanical range R. There is no change in the displacement of the variable unit in either magnitude or direction.

As soon as clutch D is engaged, and E is disengaged, there is a reversal in function of the hydraulic units. The fixed unit, which has been a motor heretofore, now becomes a pump, and the variable unit which has been a pump, now becomes a motor. However, the rotation of the hydraulic units is in the same direction as previously, thus, there is a reversal of pressure in the hydraulic lines, but no change of displacement of the variable unit.

The fixed unit now becomes a pump, and recirculates power to the variable unit, which now is a motor. Thus, the variable planetary system is subject not only to the input power provided by the engine, but also to the added power recirculated by the hydraulic system to the input gear 7. The level of gear loads is raised in the planetary above that

Table 3 - Hydrostatic Range, Low Speed*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	—	—	On	—	On
Hydrostatic, high speed, reverse	—	—	—	—	—
I, hydromechanical	—	—	—	—	—
II, hydromechanical	—	—	—	—	—
III, hydromechanical	—	—	—	—	—

*See Fig. 5.

imposed on it by the engine, and the output torque at gear 14 is, therefore, at a maximum.

To increase the speed of the output shaft, the variable hydraulic unit is reduced in stroke which results in a decrease of speed of the fixed unit and, therefore, an increase of speed of the output shaft. This also reduces the amount of power recirculated by the hydraulic system, as shown in Fig. 2, and, therefore, less power is added to the input by the variable unit. Obviously, as the output speed increases, the output torque decreases.

Finally, the stroke of the variable unit is reduced to zero, which hydraulically locks the fixed unit against rotation. At this point, the locked reaction mode is reached. Now power is recirculated, and the gear loads within the variable planetary are produced solely by the engine input (Table 5 and Fig. 7).

In order to increase the output speed further, the variable displacement unit is stroked in the opposite direction. The pressure remains in the same passages as in the recirculative mode previously described, but the flow changes direction. Thus, the variable unit becomes a pump, and the fixed unit

Table 4 - Hydrostatic Range, High Speed Reverse*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	-	-	-	-	-
Hydrostatic, high speed, reverse	-	On	-	-	On
I, hydromechanical	-	-	-	-	-
II, hydromechanical	-	-	-	-	-
III, hydromechanical	-	-	-	-	-

*See Fig. 6.

Table 5 - First Hydromechanical Range, Subrange R_R : Recirculative*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	-	-	-	-	-
Hydrostatic, high speed, reverse	-	-	-	-	-
I, hydromechanical	-	-	On	On	-
II, hydromechanical	-	-	-	-	-
III, hydromechanical	-	-	-	-	-

*See Fig. 7.

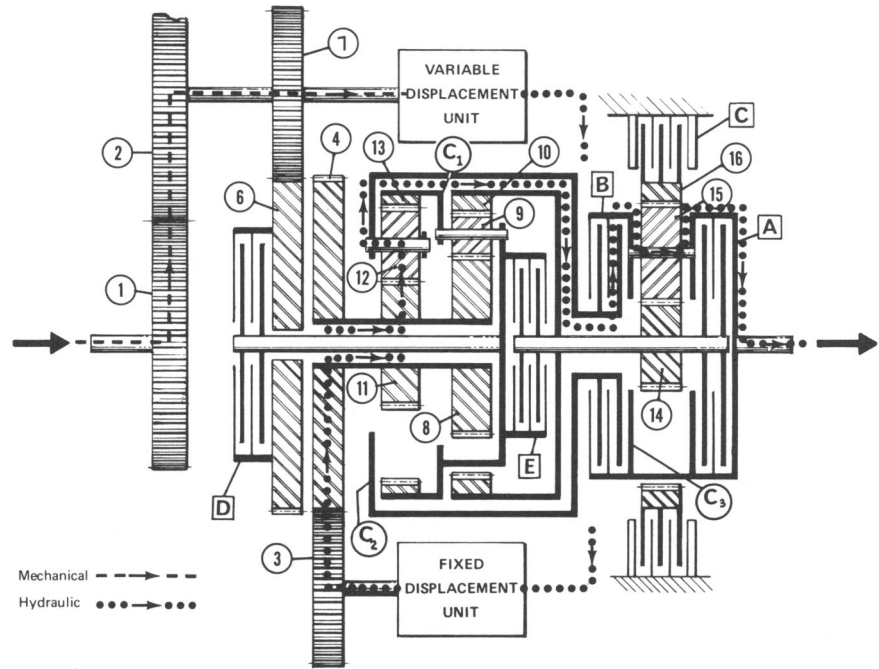


Fig. 6 - Three-range transmission - hydrostatic range, high speed, reverse

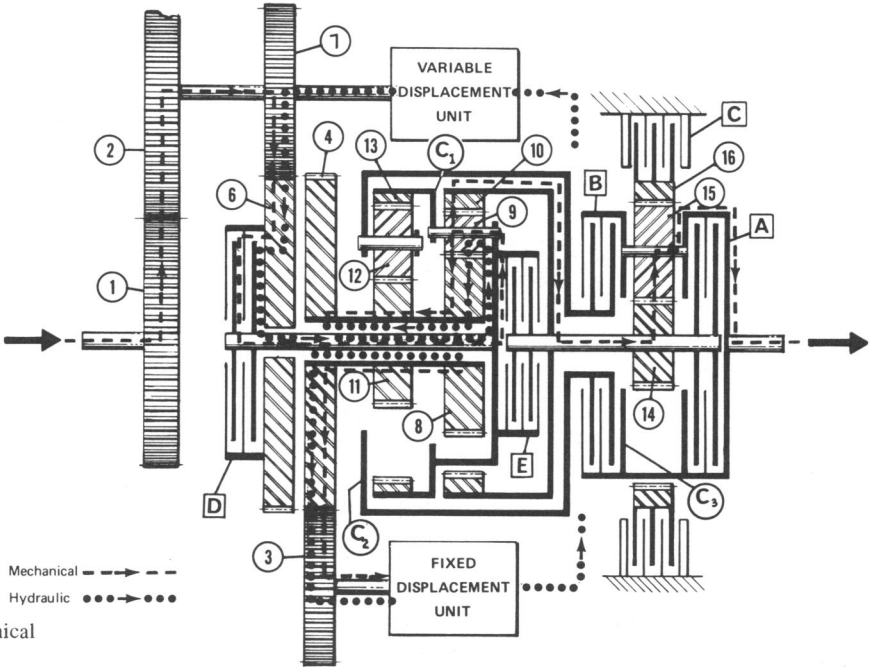


Fig. 7 - Three-range transmission - first hydromechanical range, sub-range R_R : recirculative

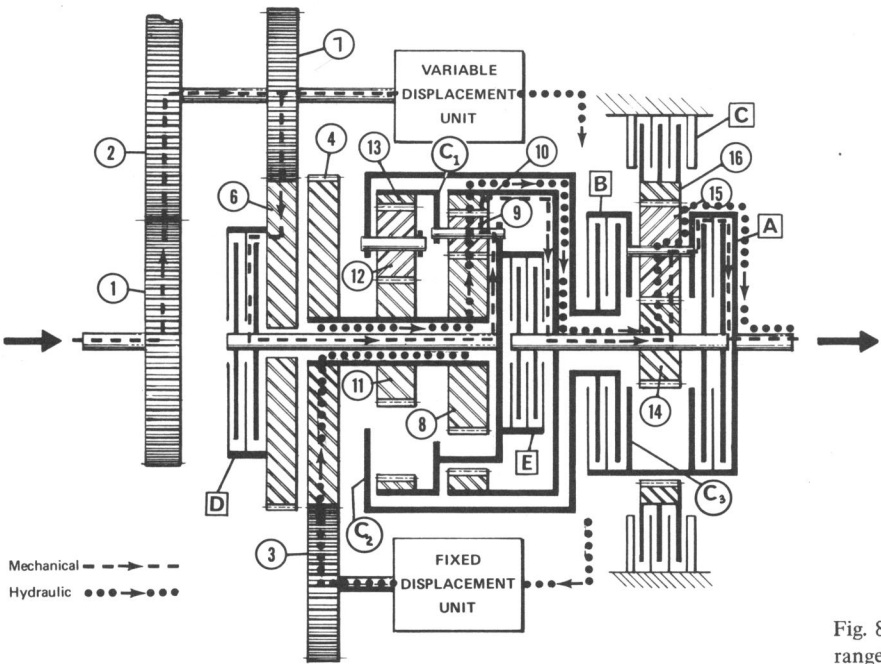


Fig. 8 - Three-range transmission - first hydromechanical range, subrange R_{NR} : nonrecirculative

a motor. Therefore, the power supplied by the engine is divided; some drives the planetary input gear 6, and some drives the variable displacement unit. The power flow is shown in Fig. 8 (Table 6).

For that reason, the gear loads within the variable planetary decrease below the level which can be produced by the engine input, as at the locked reaction point.

The fixed unit begins to rotate in the opposite direction, thereby increasing the speed of the output shaft. This increase in speed is continued by increasing the displacement of the variable unit. As the displacement of the variable unit increases, the amount of power transmitted hydraulically also increases, and the hydraulic unit requires more power to drive it. Therefore, the torque delivered to the planetary by gear 6 decreases, which results in a decrease of output torque as its speed is increased.

When the variable unit displacement is increased to the maximum, the speed of C_2 reaches that of C_3 , and it is possible to engage clutch B at a synchronous speed. The transmission now enters range II, as described further on.

Clutch B is engaged at synchronous speed, and clutch C is subsequently disengaged. The drive is now transmitted through carrier C_1 to ring gear 13, and carrier C_2 is the output of the variable planetary. It is clutched directly to the output carrier C_3 . There is once more a reversal of function of the hydraulic units; the fixed unit becomes a pump, and the variable unit a motor, but there is no change in the direction of rotation of these units and, therefore, a pressure reversal occurs. The transmission is in the recirculative mode, and, since its function is identical to that described for the same mode in range I, it is needless to repeat this description. The power flow is shown in Fig. 9 (Table 7).

As the variable unit is decreased in displacement, the transmission once more reaches the locked reaction mode. Be-

Table 6 - First Hydromechanical Range, Subrange R_{NR} : Nonrecirculative*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	-	-	-	-	-
Hydrostatic, high speed, reverse	-	-	-	-	-
I, hydromechanical	-	-	On	On	-
II, hydromechanical	-	-	-	-	-
III, hydromechanical	-	-	-	-	-

*See Fig. 8.

yond the locked reaction point, the transmission enters the nonregenerative mode and functions as described for a similar mode in the first hydromechanical range, except that the output is through carrier C_2 , clutch B, and output member C_3 , as shown by the powerflow in Fig. 10 (Table 8).

At the end of this range, the speed of ring gear 10 becomes equal to that of carrier C_3 , and a synchronous shift to the third range can be made by engaging clutch A.

In entering the third hydromechanical range, clutch A is engaged, and clutch B is thereafter disengaged. Again, the transmission operates in the same manner as described for the recirculative mode in range I, and the output is through gear 10 and clutch A, as shown in the powerflow in Fig. 11 (Table 9).

The function in this mode is identical to that described for the same mode in the first range. The drive is from gear 10 to the output member C_3 by means of clutch A. The powerflow is shown in Fig. 12 (Table 10).

Table 7 - Second Hydromechanical Range,
Subrange R_R : Recirculative*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	-	-	-	-	-
Hydrostatic, high speed, reverse	-	-	-	-	-
I, hydromechanical	-	On	-	On	-
II, hydromechanical	-	-	-	-	-
III, hydromechanical	-	-	-	-	-

*See Fig. 9.

Table 8 - Second Hydromechanical Range,
Subrange R_{NR} : Nonrecirculative*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	-	-	-	-	-
Hydrostatic, high speed, reverse	-	-	-	-	-
I, hydromechanical	-	-	-	-	-
II, hydromechanical	-	On	-	On	-
III, hydromechanical	-	-	-	-	-

*See Fig. 10.

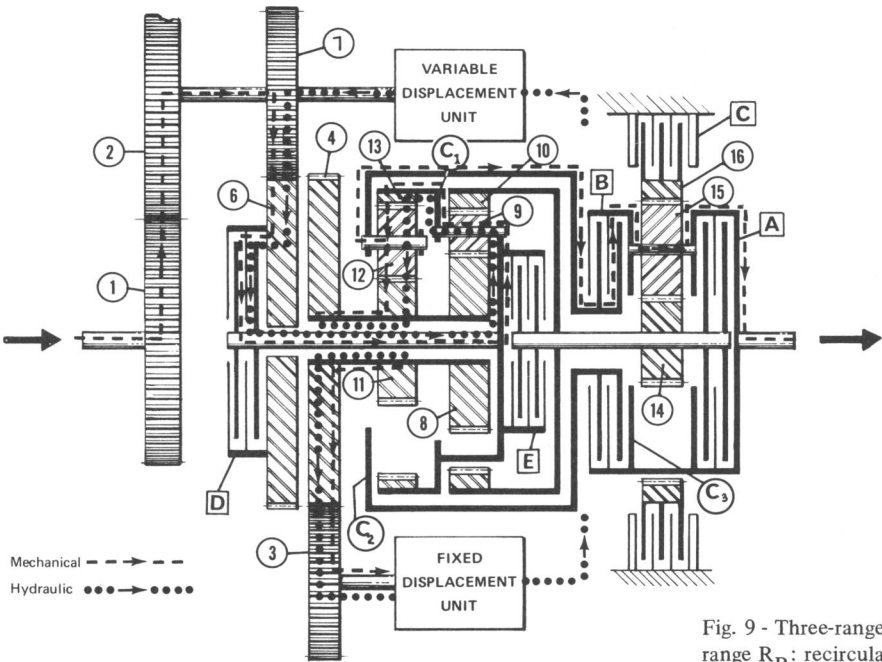


Fig. 9 - Three-range transmission - second hydromechanical range, sub-range R_R : recirculative

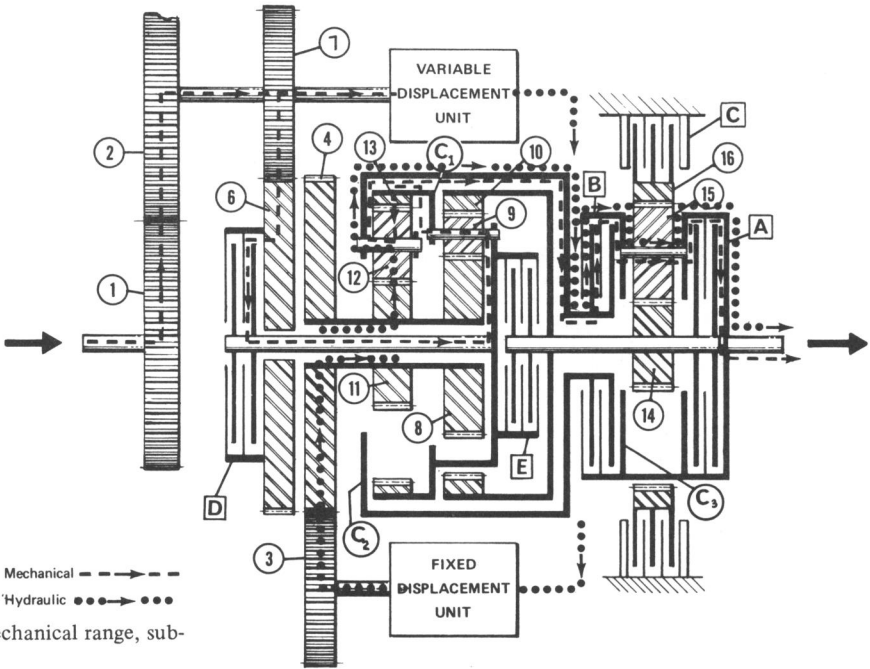


Fig. 10 - Three-range transmission - second hydromechanical range, sub-range R_{NR} : nonrecirculative

Table 9 - Third Hydromechanical Range,
Subrange R_R : Recirculative*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	—	—	—	—	—
Hydrostatic, high speed, reverse	—	—	—	—	—
I, hydromechanical	—	—	—	—	—
II, hydromechanical	—	—	—	—	—
III, hydromechanical	On	—	—	On	—

*See Fig. 11.

Table 10 - Third Hydromechanical Range,
Subrange R_{NR} : Nonrecirculative*

Ranges	Clutches				
	A	B	C	D	E
Hydrostatic, low speed	—	—	—	—	—
Hydrostatic, high speed, reverse	—	—	—	—	—
I, hydromechanical	—	—	—	—	—
II, hydromechanical	—	—	—	—	—
III, hydromechanical	On	—	—	On	—

*See Fig. 12.

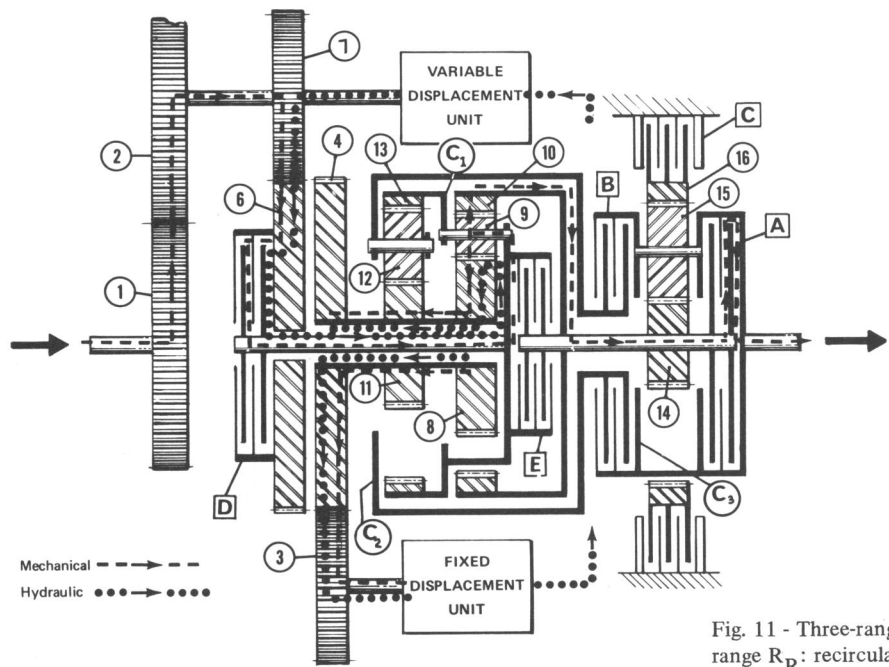


Fig. 11 - Three-range transmission - third hydromechanical range, sub-range R_R : recirculative

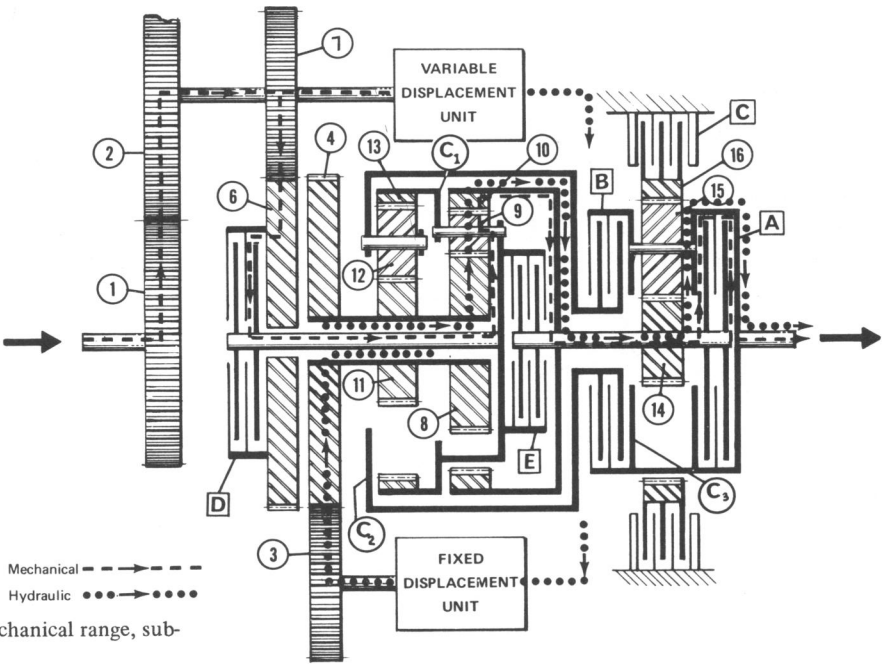


Fig. 12 - Three-range transmission - third hydromechanical range, sub-range R_{NR} : nonrecirculative

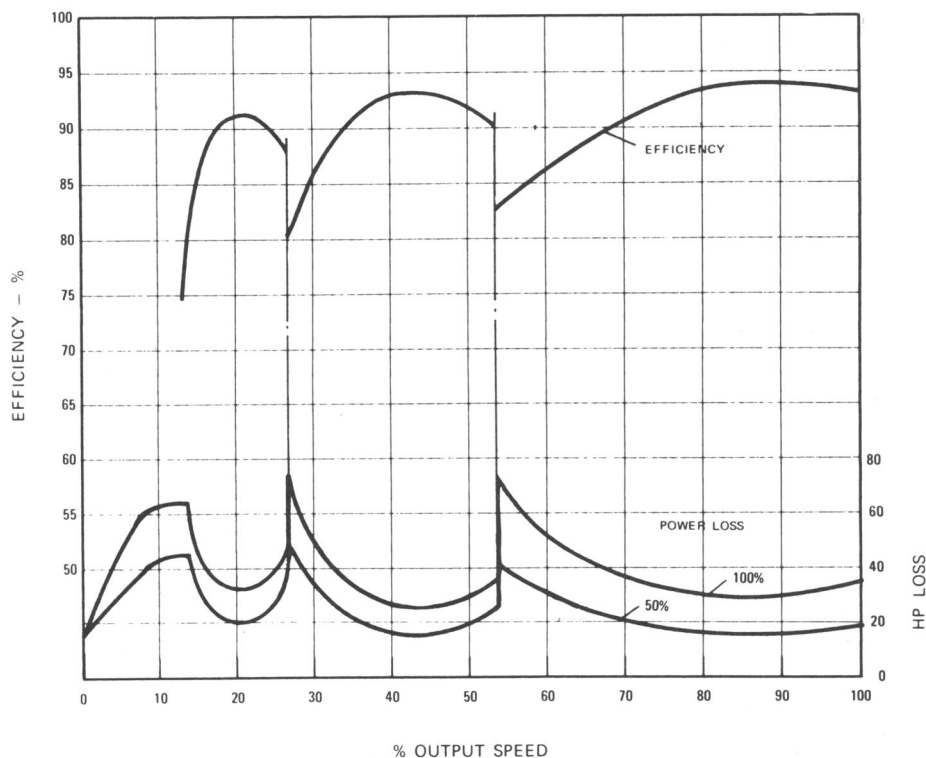


Fig. 13 - Transmission efficiency and power losses, input hp = 350

PERFORMANCE

The efficiency for 100% power is shown in Fig. 13. In addition, the horsepower loss is also shown for 100%, and 50% power level. It is particularly important to note the low loss at static torque and start conditions. This contrasts sharply with the large loss of power produced by torque converters in starting and accelerating conditions.

In a multirange transmission, it is possible to locate the area of highest efficiency in the region where the transmission operates most of the time. This can be done by adjustment of ranges, as shown in Figs. 2 and 3.

SUMMARY

The multirange transmissions have the advantage of high efficiency, ability to transmit large horsepowers with hydraulic units of relatively small size, but they require more gearing and clutch members, and a more complicated control system. For that reason, they are increasingly advantageous

where the conditions require higher horsepower levels, wider torque variation at full horsepower, and/or high efficiency over a wide range.

They reduce significantly the size and/or level of stress within the hydraulic system and, for that reason, they have potential for reduction of noise.

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DISCUSSION

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I AM PRIVILEGED to have the opportunity to discuss this paper because it is a subject with which I have been involved in various degrees during the past several years. It is apparent that the authors, Orshansky and Weseloh, have been extensively involved, because all aspects of the subject are covered so thoroughly in this excellent paper.

The discussion that follows will consist of some of my personal observations and comments, and a few questions, some of which may be controversial, and others which, when answered, should provide further insight into the subject.

The authors mention that the powerplant and the transmission can be considered as one system. This is true as long as we are talking about a powerplant that has very low torque output at low output speed. The conventional reciprocating piston internal combustion engine of the type used in nearly all of today's ground vehicles has this weakness of poor torque at low speeds and is the reason for requiring a torque converter, more often called a transmission—either hydraulic, mechanical, or both. By contrast, the reciprocating piston steam engine needs no transmission at all, since it is a constant torque powerplant.

One of the advantages attributed to the subject transmission is the "ability to program engine speed basically independent of vehicle speed as a benefit afforded by none of the previously existing transmission types." I disagree with this statement in regard to previously existing transmission types, because the potential to program engine speed is inherent in any infinitely adjustable ratio transmission, which includes such existing types as V-Belt, ball type, friction disc, or hydrostatic. Our experience indicates that it takes at least as many man-hours to develop satisfactory controls as it does to develop adequate transmission designs. Perhaps the authors can tell us more about this programmed control hardware?

In this paper the authors stated, "All range changes are stepless and conducted with overlapping clutch sequencing." That is to say, the ongoing clutch engages prior to disengaging the offgoing clutch. This maneuver raises several questions about control—namely:

1. What parameter is monitored to trigger the shift, and how?
2. How much overlap is tolerable?
3. The desirability of a synchronous shift is a noble objective; however, I am certain it is extremely difficult to attain because the hydraulic units used to provide the synchronous speeds have an indeterminate amount of leakage.
4. Are clutches necessarily the friction type?

Finally, in regard to transmission efficiency, Fig. 13 is an excellent display because it shows the variation of accumulated losses in each range and the resultant efficiency as a discontinuous curve. We have found additional discontinuities at the locked reaction mode that are not indicated on these efficiency curves and should show up near the midpoint of each hydromechanical range. This occurs when the hydraulic units change their function from pumping to motoring and vice

versa. Further, the losses look too low for a 350 hp transmission, considering such parasitics as a charge pump, control pump, windage, churning loss, or sump pump, and the frictional drag of three multiple disc clutches.

Some enlightenment in regard to how the losses were determined would be helpful. Were the test results on full-scale hardware, computer results, or guestimates?

AUTHORS' CLOSURE TO DISCUSSION

We would like to thank Mr. Barrett for his thoughts and the stimulating points they bring up.

First, with respect to the reciprocating steam engine, while it is true that such an engine can deliver an extremely wide range of torque, it is more efficient when operating within narrower speed and torque limits. For example, studies of the Rankine Cycle engine operating with a continually variable transmission indicate considerable fuel saving.

Mr. Barrett states that advantages claimed are not limited to the type of transmission which is the subject of our paper "because the potential to program engine speed is inherent in any infinitely adjustable ratio transmission, which includes such existing types as V-Belt, ball type, friction disc, or hydrostatic." In this regard, we agree entirely with Mr. Barrett.

With respect to controls, it is true that they represent considerable difficulty in the design of a transmission. We did not intend to cover more than the basic transmission in this paper and hope to go into the control phase of it at some future time.

Mr. Barrett's question with respect to "the ongoing clutch engages prior to disengaging the offgoing clutch" is very pertinent, and considerable development has been conducted by us to accomplish the desired results. The final design consists of a sequence valve which feels the pressure rise in the ongoing clutch before it releases the pressure of the offgoing clutch, so that until the pressure of the ongoing clutch has reached a value sufficient to transmit torque, the ongoing clutch is not disengaged. The overlap in speed which we found acceptable is preferably not over 50 rpm of relative clutch members. The clutch, theoretically, need not be of the friction type, and experimental work is going on in this direction. However, this further complicates the controls which are necessary for clutch actuation. This is easier to do with electronic controls than with hydraulic controls.

With respect to efficiency, Mr. Barrett is right in that there is a discontinuity at the locked reaction mode as shown in Dr. Looman's book (Ref. 1 of our paper).

The losses were calculated by using the computer program based on actual tests on full-size hydraulic pumps and motors, the charge pumps necessary to supply them, and an estimate of gear losses based on 1% per mesh loss in the planetary gearing and 2% loss in the countershaft gearing. The losses of disconnected clutches were calculated from tests of clutches of similar design but smaller size. The efficiency shown in the paper is, therefore, a combination of actual tests of hydraulic components and what we feel are accurate estimates of other components.