HYDROSTATIC TRANSMISSIONS

BY

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PART I

FUNDAMENTALS

Introduction

Hydrostatic transmission is one of the oldest methods of transmitting power. Generally it is a pump, coupled to a prime mover, supplying oil under pressure to a hydraulic motor (actuator) coupled to the load. The Williams Janney transmission has been used successfully since the beginning of the century and is still going strong. The main reasons why hydrostatic transmissions have not been more extensively used are because they have been too bulky and expensive to produce. In the last ten years efforts have been made by hydraulics manufacturers to reduce bulk by using higher operating pressures (up to 6.000 lb/sq in.) and to reduce costs by improved production methods. In this the manufacturers have been only partly successful. In general the increased operating pressures and speeds, coupled with attempts at semi-mass-

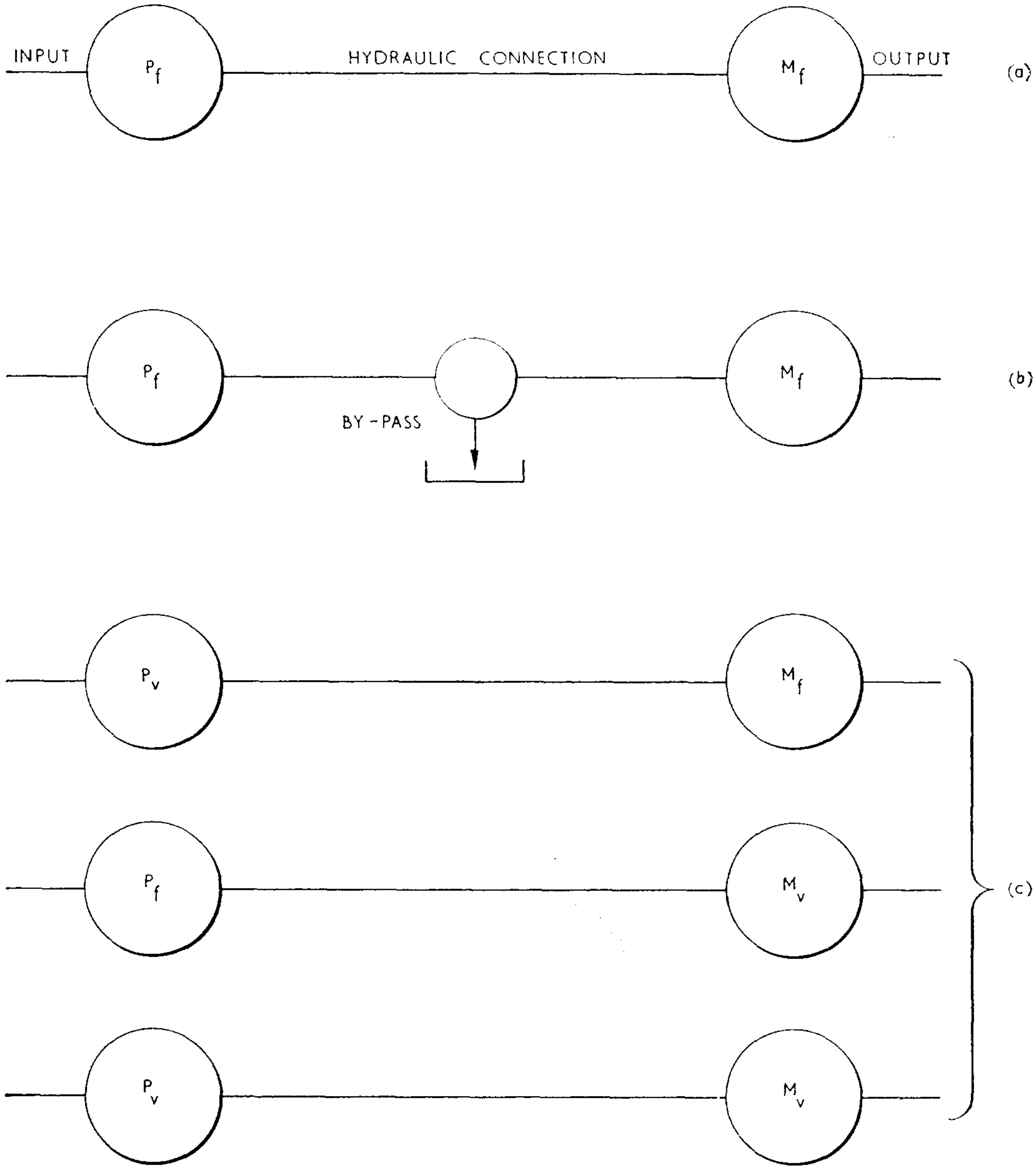
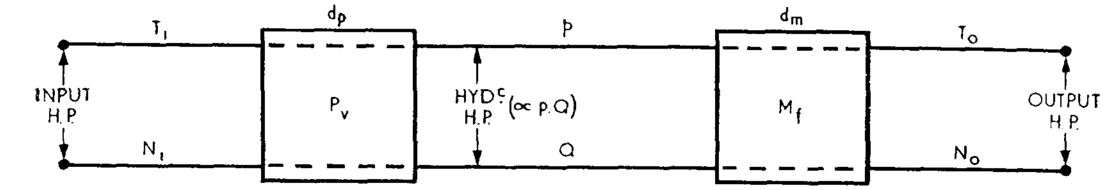


FIG. 1

Key to Symbols

P_f	—Fixed displacement pump	T_i	—Input torque
P_v	—Variable displacement pump	T_o	—Output torque
M_f	—Fixed displacement motor	N_i	—Input speed
M_v	—Variable displacement motor	N_o	—Output speed
P	—System pressure	d_p	—Pump displacement
P_o	—Boost or back pressure	d_m	—Motor displacement
Q	—Flow	N_o/N_i	—Speed ratio

production, have led to quite a number of unreliable units being marketed, and the growing competition in the hydraulics industry has led to partially developed units being prematurely introduced. When these transmissions do come on to the market reasonably priced and with high reliability, we may expect to see them more extensively used, particularly in the automotive (tractors, earth moving equipment, etc.) industry and in the marine and military fields.



Ti does not include the torque to drive auxiliary equipment such as a boost pump

FIG. 2

Briefly, the main advantages of hydrostatic transmission are:—

(*a*) Output speed can be infinitely variable from zero up to some speed ratio, both forward and reverse, with the prime-mover speed constant and in the same direction;

(*b*) Output torque and speed may be varied independent of each other;

(*c*) The capacities of the pump and motor units can be made different so as to have the same effect as a fixed gear train between the motor and the load;

(*d*) The pump and motor units can be placed apart so that the load can be actuated remote from the prime mover;

(*e*) Hydrostatic systems can be made to regenerate when braking or under negative load (runaway load).

The most obvious disadvantages are:

(*a*) Low efficiency over much of the working range;

(*b*) The more highly rated units tend to be noisy.

Basic Principles

Hydrostatic transmission in its simplest form is a positive displacement pump imparting potential energy to some fluid by increasing its pressure. The output torque is proportional to the pressure drop across a motor.

If the pump and motor units are of fixed displacement (FIG. 1(*a*)) then the system would be equivalent to a piece of shafting. If the units were of different displacement the system would then be equivalent to a geared system. The only time such a system would be used is when normal mechanical means is impossible.

The system shown in FIG. 1(*a*) has a fixed speed ratio. FIG. 1(*b*) shows a simple, though inefficient, means of varying the output to input speed ratio. Here we use a by-pass circuit with a throttling valve so that we throw power away and control motor speed by the amount of fluid directed to it. Although inefficient, it can be shown that for certain load characteristics (propeller law) the losses will be quite small. This will be discussed in Part II of this paper. In this part of the paper it is not proposed to discuss the two above transmissions but to concentrate on the more generally used transmission system since this embodies the advantages of hydrostatics.

A hydrostatic transmission as it is generally understood is a pump and motor system with either or both units having variable displacement (FIG. 1(*c*)). To simplify the discussion it is proposed to deal only with the most common combination—a variable displacement pump feeding a fixed displacement motor. The same principles can then be easily applied to any other combination.

If we consider the geometry of a hydraulic pump and motor and refer to FIG. 2, we can quickly deduce the following relationships:—

$$Q \propto N_i \times d_p \quad \text{also } N_o \propto Q/d_m$$

$$\therefore N_o = N_i \cdot d_p/d_m$$

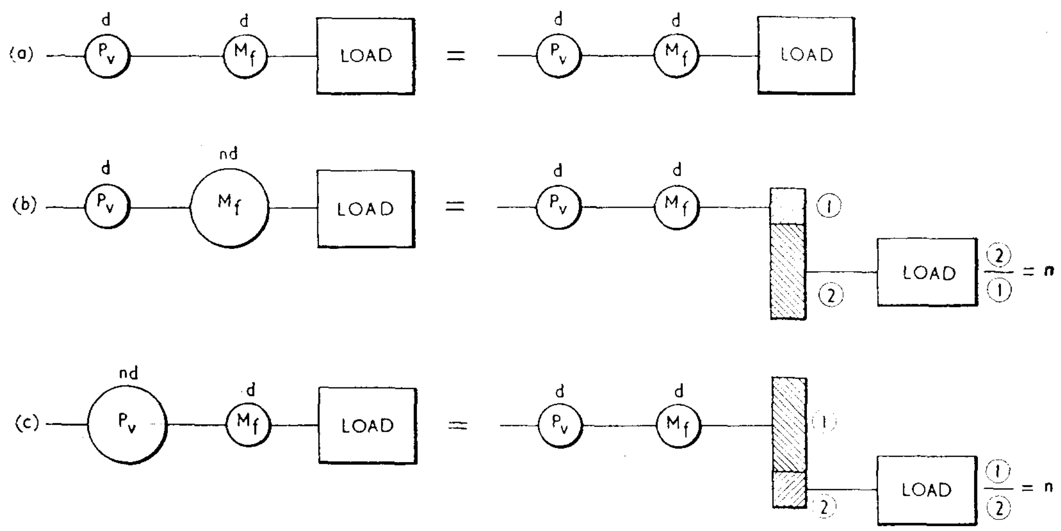


FIG. 3

We can see that the torque loading of the system (ignoring losses) has no effect on the speed ratio N_o/N_i. We can therefore say that in FIG. 2 the N_i—Q—N_o line is independent of the T_i—p—T_o line.

Considering the torque/pressure line in FIG. 2.:

$$p \propto T_i/d_p \quad \text{and} \quad T_o \propto (p-p_o)d_m$$

Since p_o is usually small compared to p, let us for simplicity ignore it.

$$\text{Thus: } T_o = T_i.d_m/d_p$$

We can see clearly that torque and speed in a hydrostatic transmission are independent of each other (ignoring losses); hence one of the main advantages of this type of transmission. If the load varies then the system pressure will vary to accommodate this new load level provided that sufficient torque can be drawn from the prime mover.

Another useful feature of hydrostatic transmissions is that they can be used to gear up or gear down the overall transmission to avoid the use of auxiliary gearing (see FIG. 3). The diagrams should be self explanatory.

Once again to avoid complication in discussion, only units of the same capacity will be considered from now on.

One further advantage of hydrostatic transmissions is that they can be used to regenerate power. If the load becomes negative (i.e. overruns) then the motor can act as a pump and the pump as a motor, and feed power back into the prime mover. Deceleration can be achieved by reducing the pump displacement. The rate at which this is done, however, has to be controlled such that that inertial effects of the load do not develop excessive pressures. This point will be discussed in more detail in Part II when systems are discussed in detail.

Performance

The horsepower capacity of a particular hydrostatic transmission is not a discrete figure; it is dependent upon the application. If as shown in FIG. 4(*a*), we plot torque on speed and draw a constant h.p. curve over a range of speeds we see that we have a hyperbola. For a given transmission we can put a limit on pressure* (torque) for strength and other reasons to do with pump and motor design.

*The pressure for a given unit may of course vary depending upon whether it is a long or a short-life application.

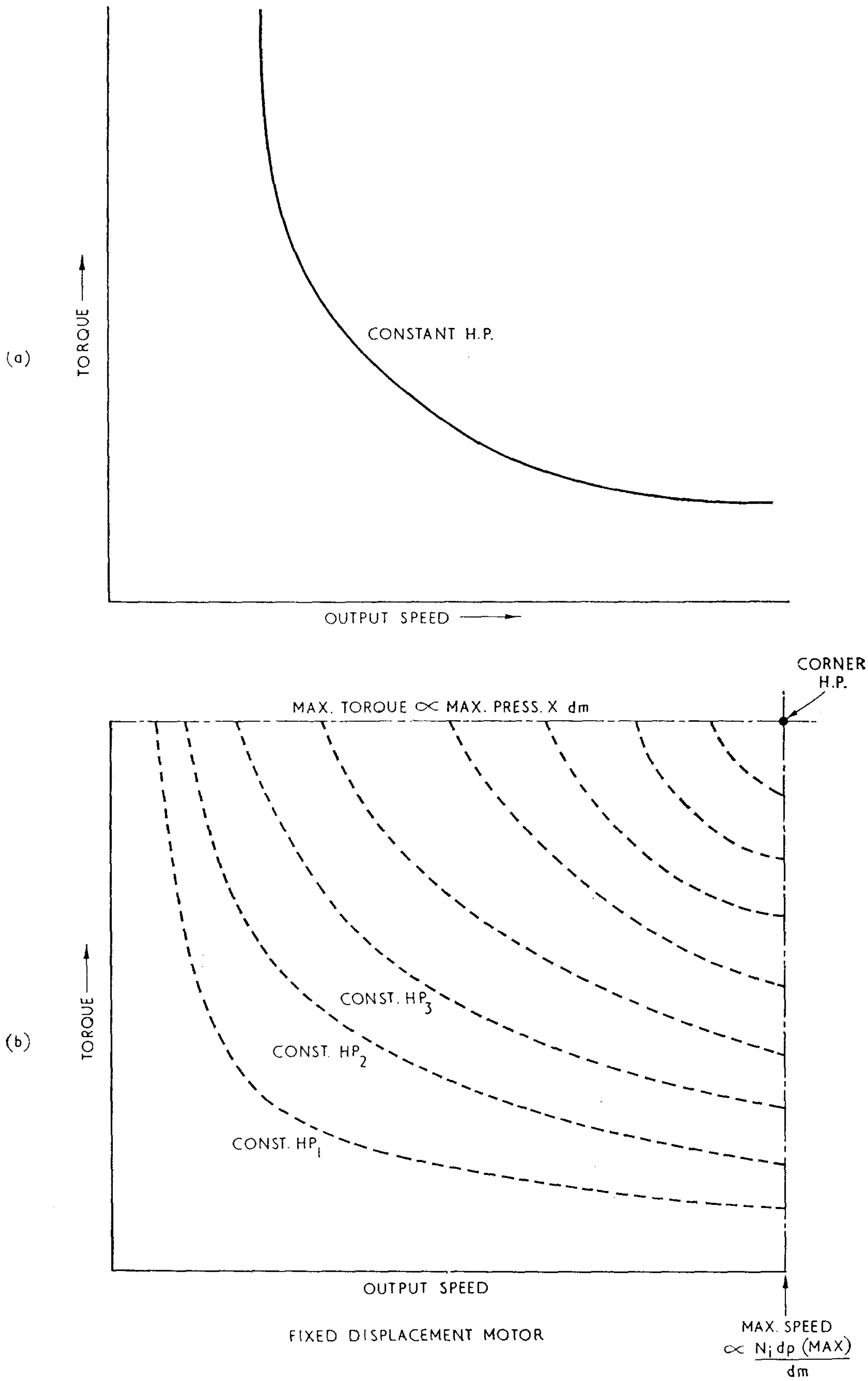
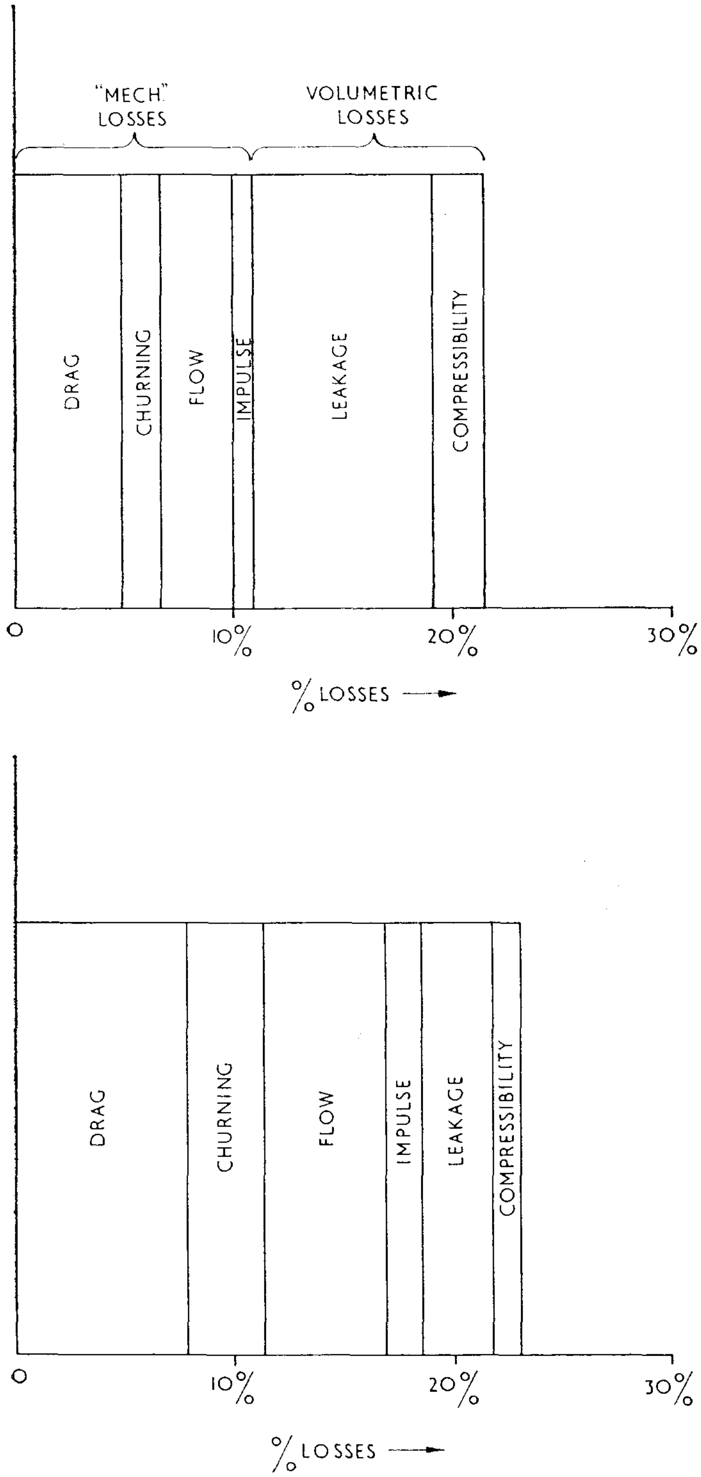

(a)
TORQUE
CONSTANT H.P.
OUTPUT SPEED
(b)
TORQUE
MAX. TORQUE ∝ MAX. PRESS. X dm
CORNER H.P.
CONST. HP3
CONST. HP2
CONST. HP1
OUTPUT SPEED
FIXED DISPLACEMENT MOTOR
MAX. SPEED ∝ Ni dp (MAX) / dm

FIG. 4



(*a*) LOW STROKE

	PRESS	SPEED
PUMP	HIGH	HIGH
MOTOR	HIGH	LOW

(*b*) FULL STROKE

	PRESS	SPEED
PUMP	LOW	HIGH
MOTOR	LOW	HIGH

FIG. 5

Also, since the unit has a maximum speed dictated by mechanical friction and other reasons, we can draw an envelope of operating conditions outside which the transmission must not operate. It will be seen that a family of constant h.p curves have been contained within the power envelope (FIG. 4(*b*)) and as these horsepowers increase it will be observed that the speed range over which we can operate at constant h.p. is reduced until we reach a point where the maximum torque line and the maximum speed line intersect. This point is often

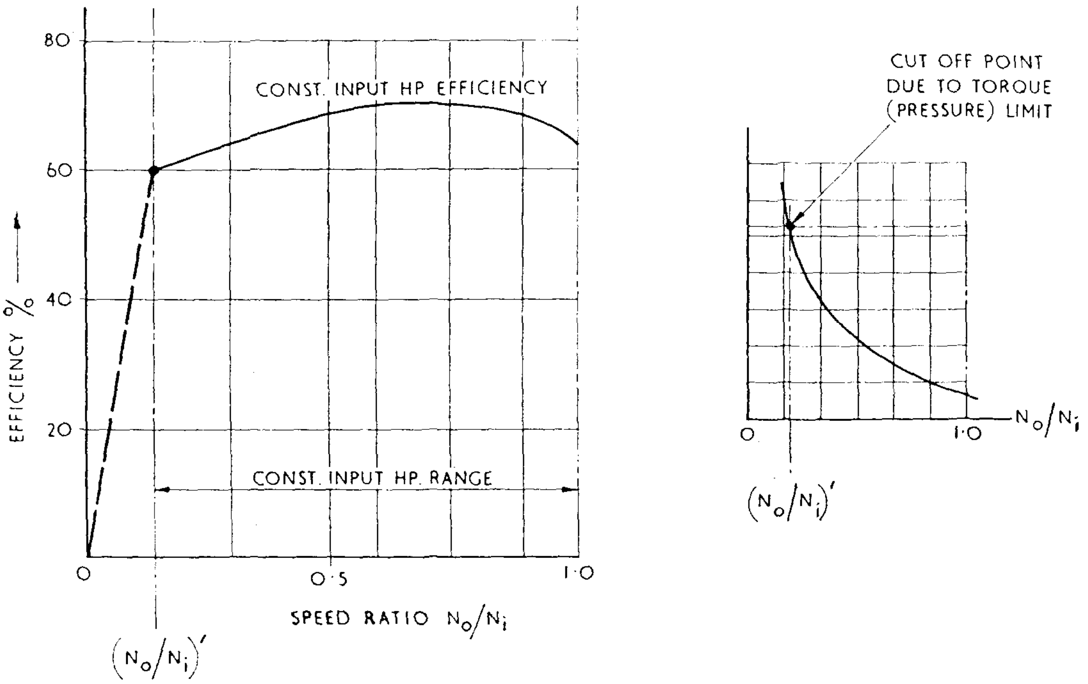


FIG. 6(*a*)

referred to as the 'corner h.p.' (c.h.p.) and this term is sometimes used to denote the capacity of a transmission.

A simple relationship says:—

$$\frac{\text{Corner horsepower}}{\text{Speed ratio required}} = \text{Maximum constant input h.p.}$$

Since in most cases we don't have a close control over the load and we rarely operate under constant h.p. conditions, the above relationship is of little but academic interest. In general we may expect our h.p. output to be anywhere in the envelope, be it constant h.p., constant-torque/varying-speed, or any condition.

Efficiencies

In spite of what manufacturers tell us, hydrostatic transmissions rarely give us peak efficiencies in excess of 80 per cent; most commonly they lie between 70 and 80 per cent.

It is easy to see why the efficiencies are so low when we realize that there is a double conversion of energy involved. The pump has a volumetric efficiency and a mechanical efficiency, as does the motor, and there is also a system efficiency. Even when we attribute high efficiencies to all these components, when multiplied together we shall not get much in excess of 80 per cent.

The breakdown of losses for a typical transmission is shown in FIG. 5. The losses are broken down into their main components for a transmission with constant input h.p. and input speed at (*a*) low pump displacement and (*b*) at full pump displacement. The two charts given alongside the two cases indicate what the pump and motor are experiencing in terms of pressures and speeds. We may make the following observations:—

(*i*) The volumetric losses are the greatest in case (*a*) since the system pressure is highest.

(*ii*) It will be noted that the mechanical losses are also quite high since although the motor is turning at low speed the pump is turning at

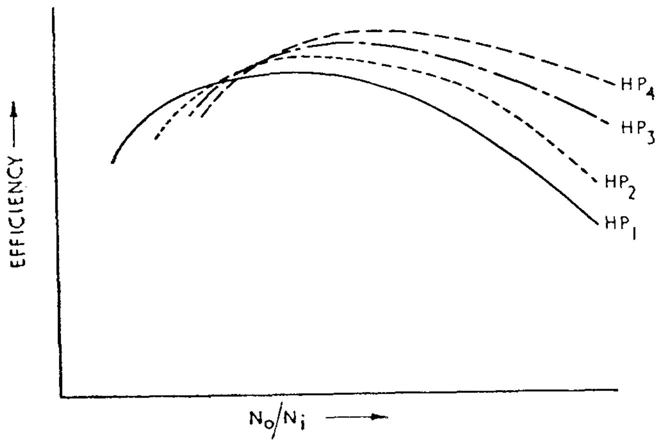


FIG. 6(*b*)—SHOWING HOW A FAMILY OF CURVES COVERING A RANGE OF H.P.S CAN BE CONFUSING

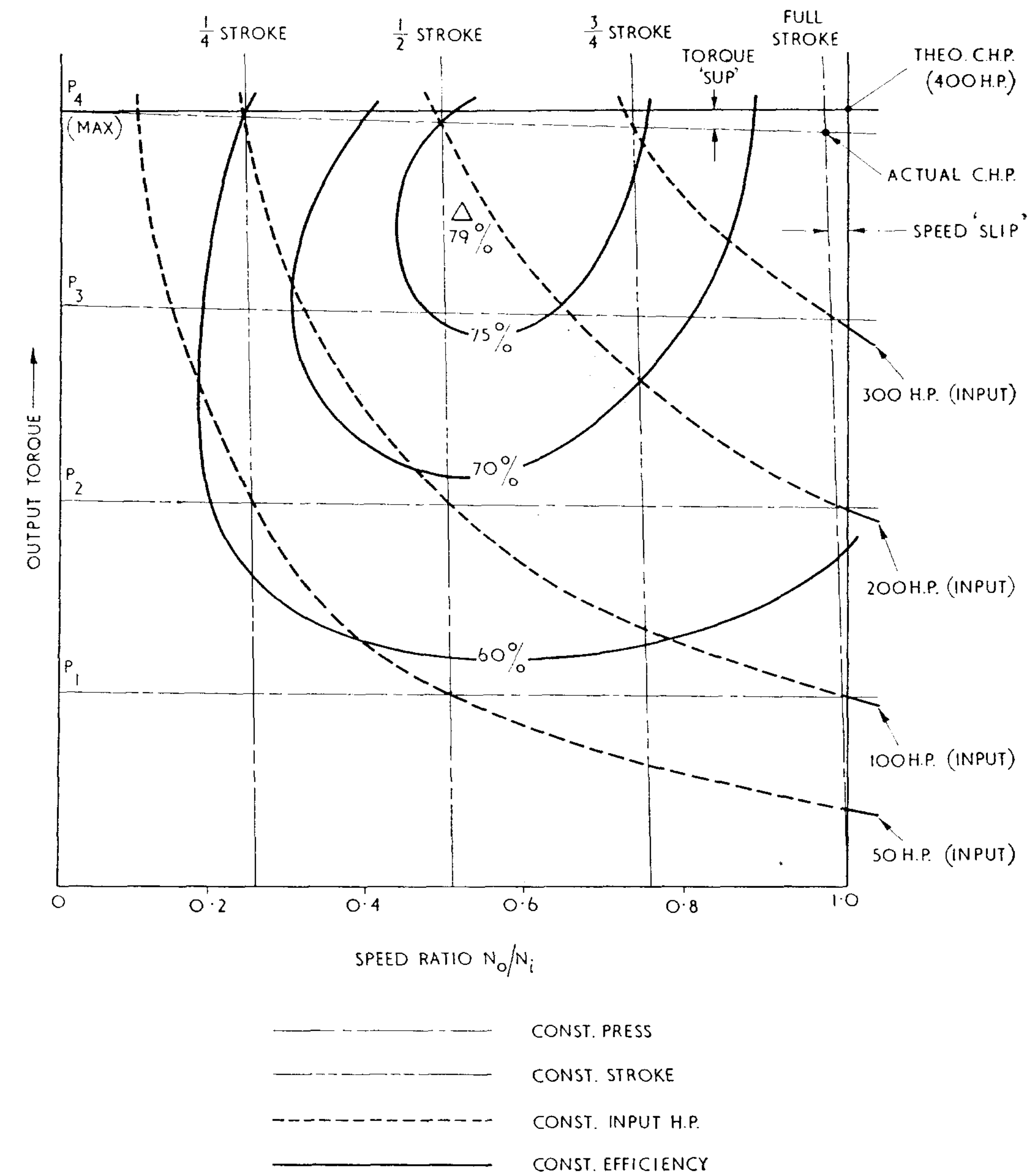


FIG. 7

maximum speed. Therefore most of the mechanical losses at condition (*a*) will be attributable to the pump.

(*iii*) The mechanical losses in case (*b*) become the major loss when the speeds of both the pump and motor are high and the system pressure is low.

FIG. 6 depicts a common way of indicating efficiencies and shows that efficiency is at its maximum at about mid-stroke (usually between one-third and two-thirds maximum displacement). This curve, although a simple method of illustrating the range of efficiencies with varying displacements at constant input h.p., does not prove very useful in practice since we rarely operate a transmission in this condition, as stated earlier. A more useful way of showing efficiencies is shown in FIG. 7. This so-called 'onion curve' is very useful since it gives the whole blanket of performance. There is, unfortunately, no simple means of expressing this information non-dimensionally, which means that a series of these curves is needed for each input speed.

It can be seen that the optimum h.p. for this unit is between 100 and 300 h.p. at speed ratios from 0·3 to 0·8. It would obviously be wrong to use this transmission for powers as low as 50 h.p. since we should need more than 80 h.p. input. Clearly another size of transmission is required.

Part II of this article will appear in the next issue of the *Journal*.