AMERICAN CONSULTING ENGINEERS COUNCIL'S

HEAVY MOVABLE STRUCTURES
MOVABLE BRIDGES AFFILIATE

3RD BIENNIAL SYMPOSIUM

NOVEMBER 12TH - 15TH, 1990

ST. PETERSBURG HILTON & TOWERS
ST. PETERSBURG, FLORIDA

SESSION
WORKSHOP NOTES

Session (1-12)
"Comparison of Two Electrohydraulic Drives." F.Liedhegener presented by P. Stavrou, Rexroth Corp. Pa.

Comparison of two electrohydraulic drives

F. Liedhegener

"Drive technology versus brake technology?"

Every technical drive system offers the user certain advantages. To select properly, the user must, however, also be aware of the disadvantages.

Example: Closing and opening a lock gate or lifting and closing a bridge. (Figs. 1 and 2)

A motion sequence as shown in Fig.3 is to be performed.

Two typical drives will be considered in more detail.

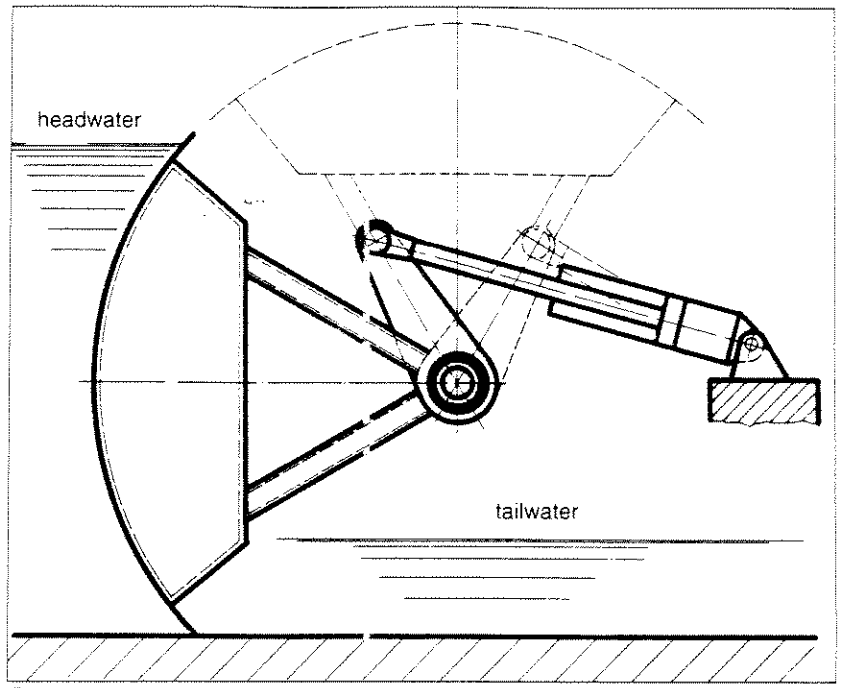

Fig. 1

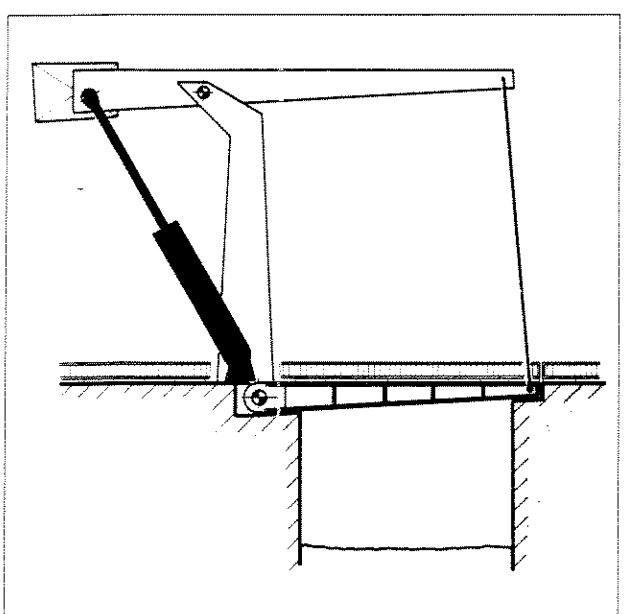

Fig. 2

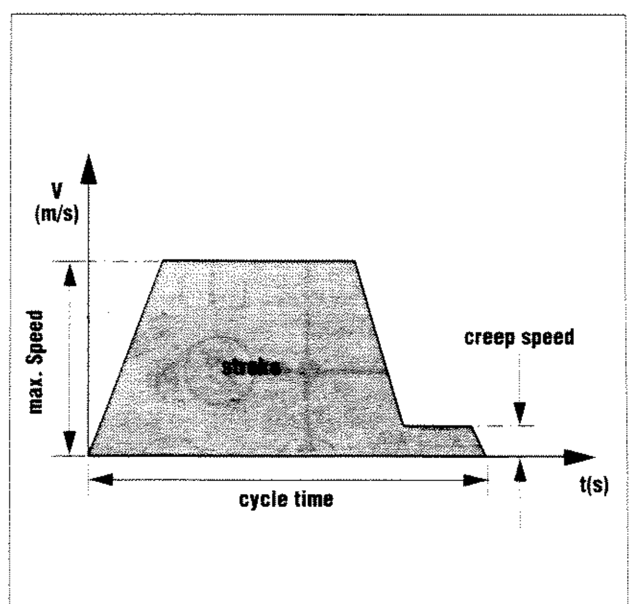

Fig. 3

Drive A and drive **B** each consist of a cylinder which moves a load.

While **drive A** is supplied from an electrohydraulic variable pump (Fig. 4), **drive B** is controlled by an electrohydraulic modulating valve ("proportional valve") (Fig. 5).

The pump *belonging to drive A* is driven under all conditions of load at a certain rotational speed. *The delivery volume can* be infinitely varied via an electrohydraulic mechanism to a value between Q = 0 and Q_{max}. (Fig. 6)

Motion can therefore take place according to the chart in Fig. 3. The load pressure p_L depends on the particular load.

If, for instance, travel is at a particular velocity while the load is increasing, the *pump's drive motor* draws more electrical power (current) from the supply network. *The pump speed remains* unchanged, the load pressure increases as shown in Fig. 7. *The power available at the cylinder* "adjusts to the load". The efficiency depends only on *the quality of the components* (friction, leakage).

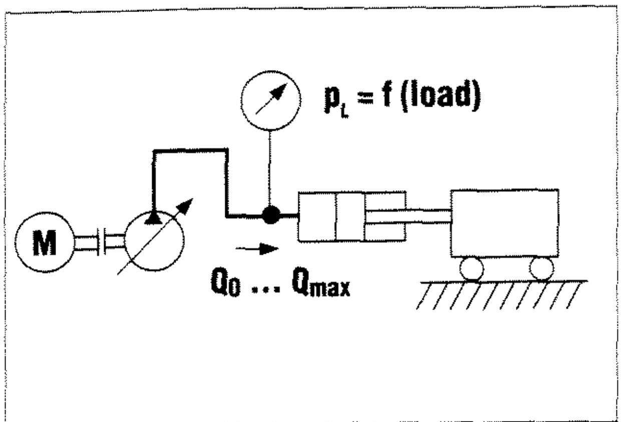

Fig. 4

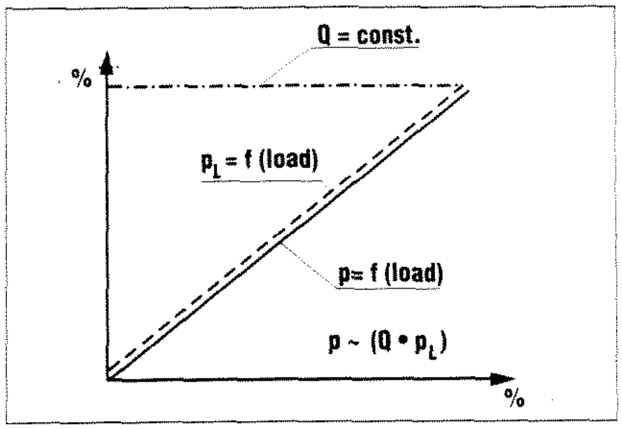

Fig. 7

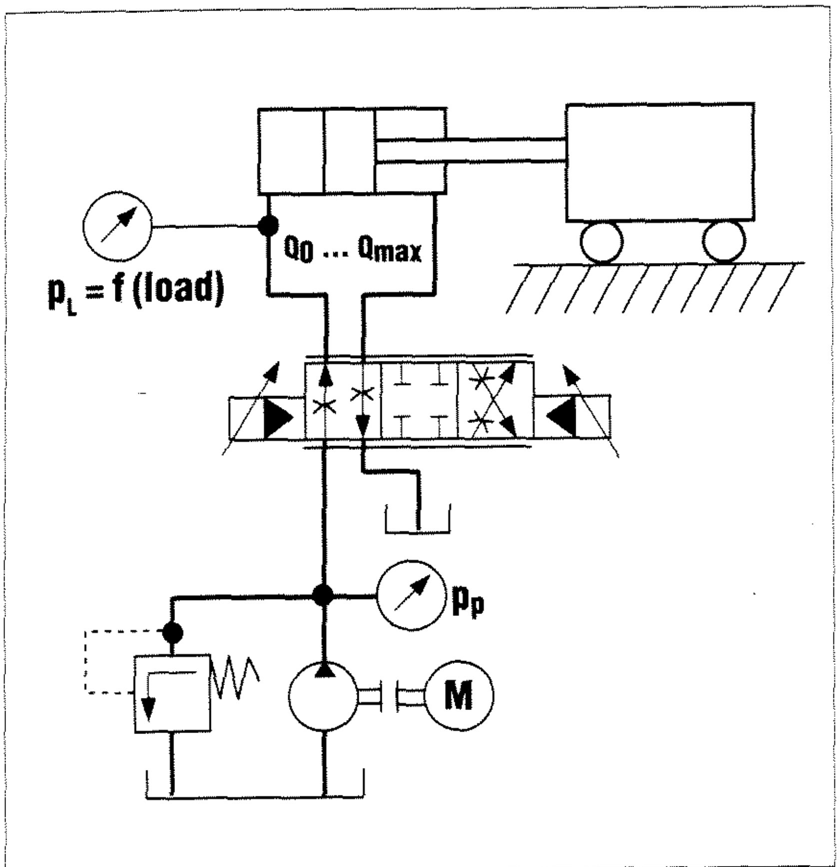

Fig. 5

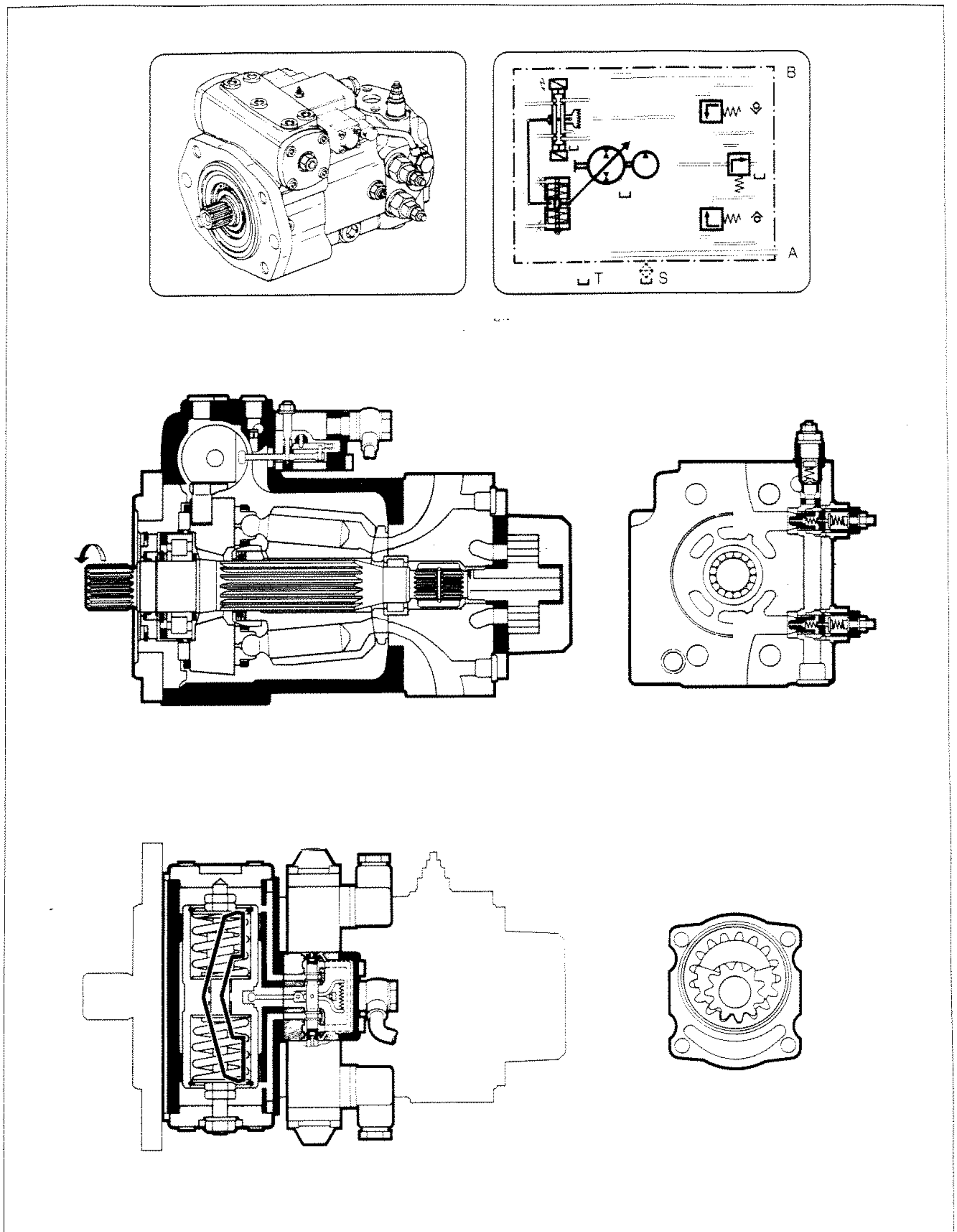

Fig. 6

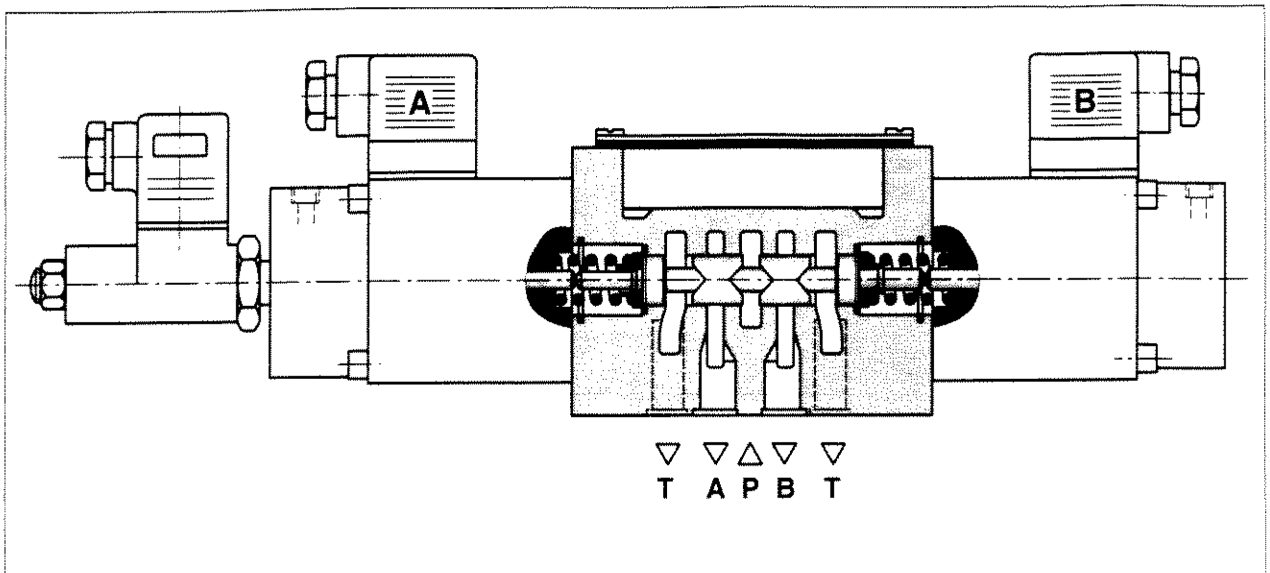

Fig. 8

In drive B, the load pressure p_L **also** assumes a value which depends on the load. In contrast to **drive A**, however, the oil volume Q = 0 ... Q_{max} is provided through an electrohydraulic modulating valve (proportional valve).

Electrohydraulic modulating valves vary the stroke of the main piston in proportion to the electrical input signal. (Figs. 8 + 9)

On taking a closer look, it can be seen that the volumetric flow from the pump to the cylinder and from the cylinder to the tank is "throttled", i.e. the motion sequence shown in Fig. 3 can take place.

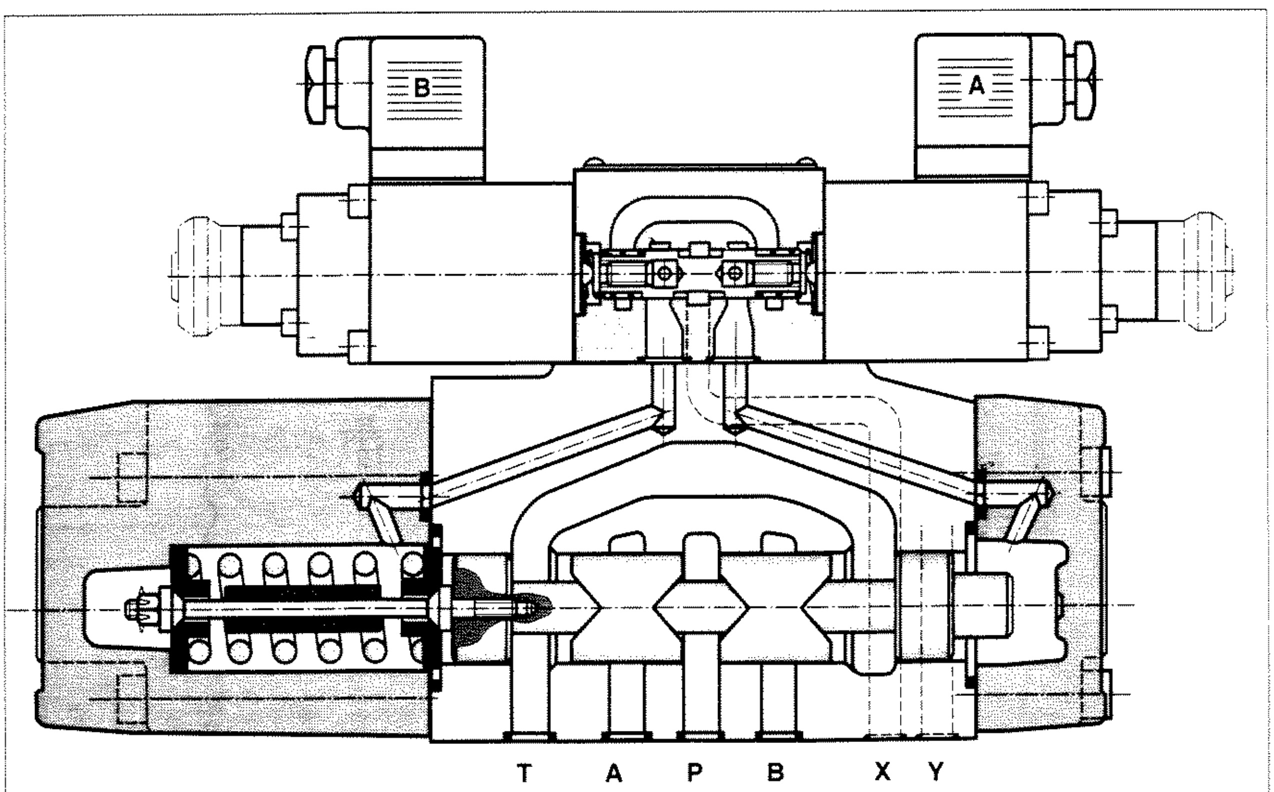

Fig. 9

In contrast to drive A, however, the volumetric flow Q does not remain constant in drive B when the load pressure p_L changes due to the load. (Fig. 10)

While the load pressure rises, for example, the valve pressure drop Δp valve falls to the same extent! As a result, $Q \sim \Delta p$ valve changes. The cylinder becomes slower with a higher load. It can also be seen that the power available at the cylinder $P = f(Q \bullet p_L)$ follows quite a different curve from that shown in Fig. 6.

The available power maximum at the cylinder is obtained when Δp valve is 1/3 of the set pump pressure. There is a clear disadvantage in terms of efficiency compared with drive A.
$[(\Delta \text{ valve} \bullet Q) = \text{Ploss} = \text{heat !}]$

If drive A is more efficient, what advantage does drive+B have?

Oil is a compressible fluid. The oil volume between pump and cylinder piston in drive A must be considered as a spring. The same applies to the oil volume in the two cylinder chambers (and pipes up to the valve) in solution B. **Drive A** can be considered to represent a mass on a spring (Fig. 11) while **drive B** represents a mass **between** two springs (Fig. 12) (a better principle, i.e double spring stiffness). Both systems can therefore resonate if excessive accelerations or sudden load fluctuations occur. The acceleration can, however, be preselected and thus matched electrically for both drives. (Analog setting on the ramp time potentiometer in the contrōl electronics or digital setting with a programmable controller.)

Drive A always requires a "positive" load (a pulling load would create a vacuum and would have to be compensated by counterbalance or brake valves.)

Drive B also allows "negative" forces. (On the land of the modulating valve a higher pressure automatically builds up and supports the load.)

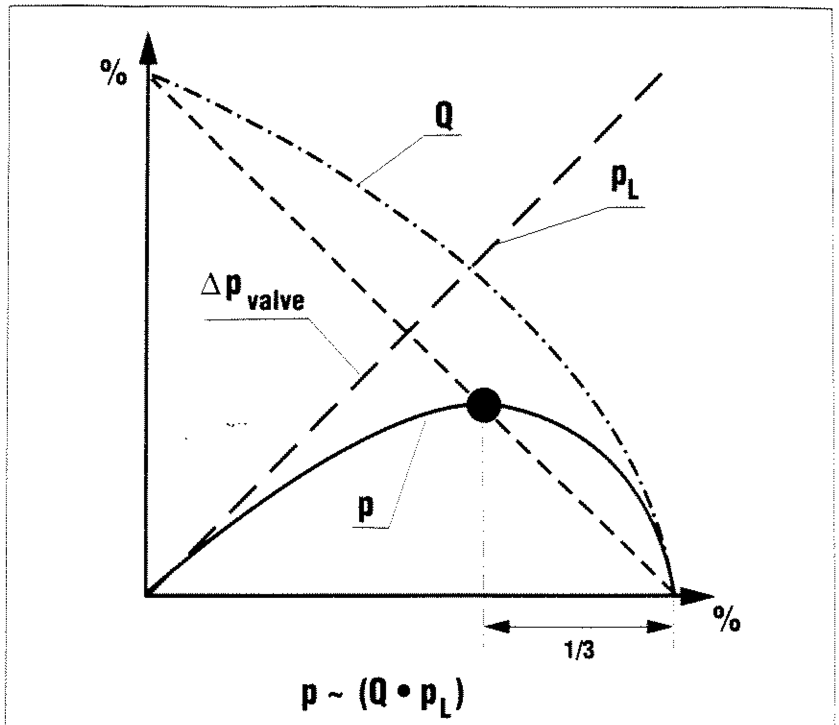

Fig. 10

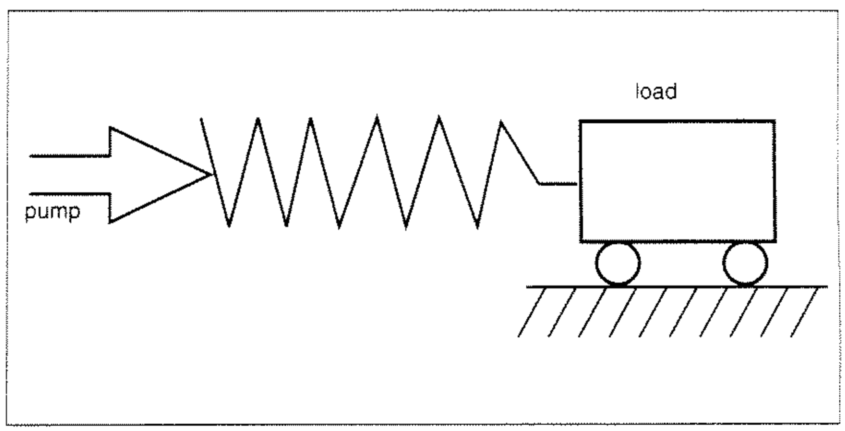

Fig. 11

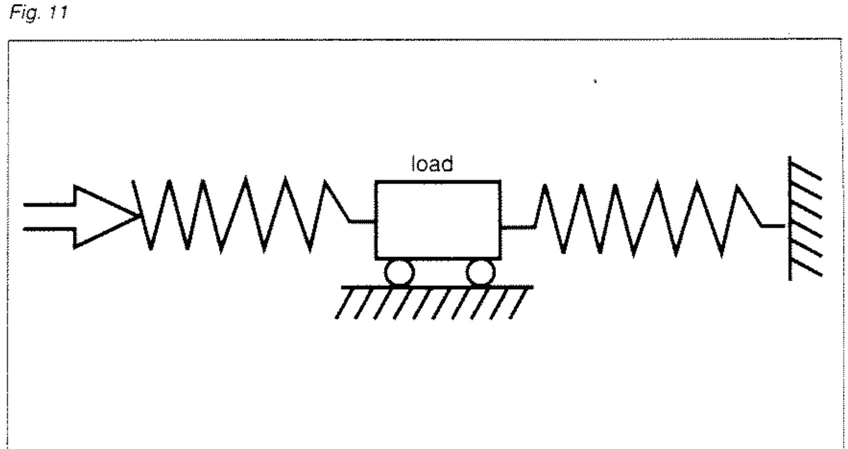

Fig. 12

Both drives can be improved in principle by keeping the distance between pump and cylinder or between modulating valve and cylinder as small as possible. (*Further improvement of load stiffness.*) This measure can frequently be carried out more easily on **drive B**.

A larger cylinder diameter is always the best method of improving the load stiffness of a hydraulic system, but this is often not possible in practice for reasons of cost and design limitations.

The fundamental disadvantage resulting from velocity variation as a function of load in the case of drive B can be offset by, for example, using a pressure compensator (*Figs. 13 and 14*).

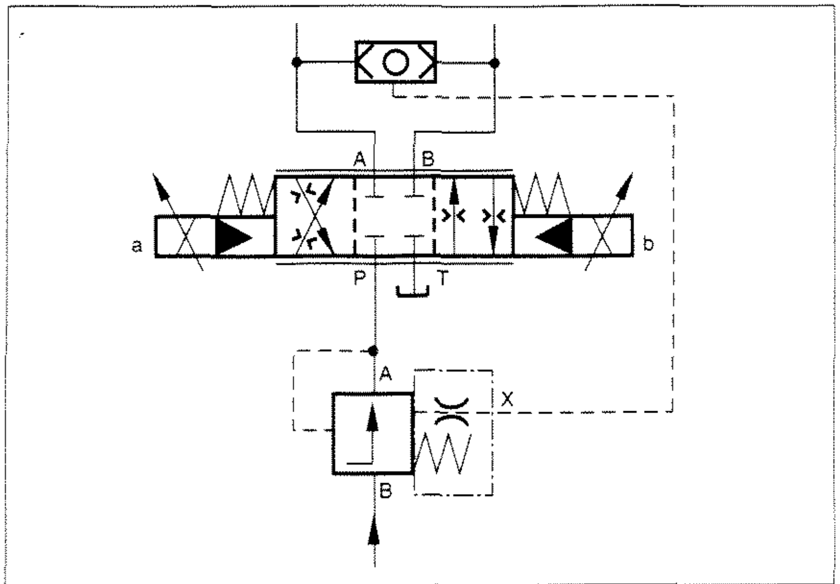

Fig. 13

Naturally, drive B can also be built with an electrically adjustable flow control valve on the inlet side. (Fig. 15)

Compared with a control having modulating valve and pressure compensator, one can see that there is now no land. One of the two "hydraulic springs" is therefore also omitted.
The load stiffness is less.

Negative forces must be absorbed by counterbalance or brake valves. This solution is favoured on cylinders with large piston area differences, i.e. on thick, buckle-proof piston rods. The high pressure which arises from the pressure ratio in the annular side of the cylinder could cause destructive damage.

While solution A represents a drive in the classical sense, solution B involves "braking technology". Acceleration of the load in drive B occurs by "releasing the brake", so to speak, i.e. the modulating valve releases the throttle cross-sectional areas.

Whereas drive concepts like solution A are generally measured by their efficiency, "braking systems" like solution B, which are not capable of being so efficient, are used to optimally control dynamic masses.

In the end, it is the application which decides as to which concept is used.

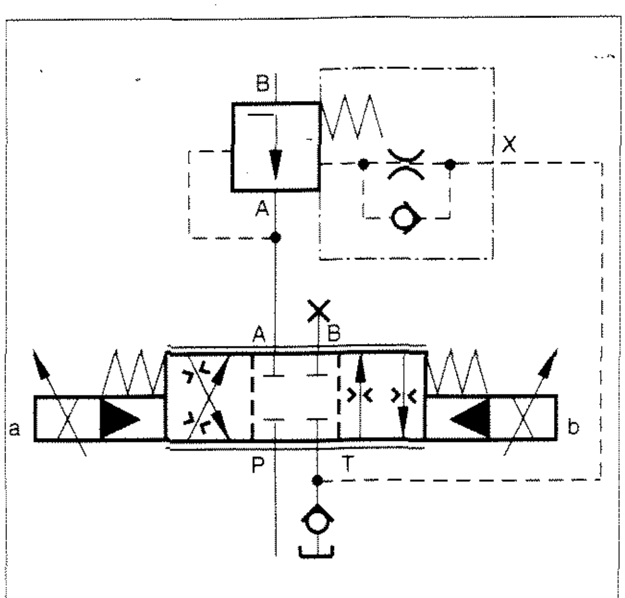

Fig. 14

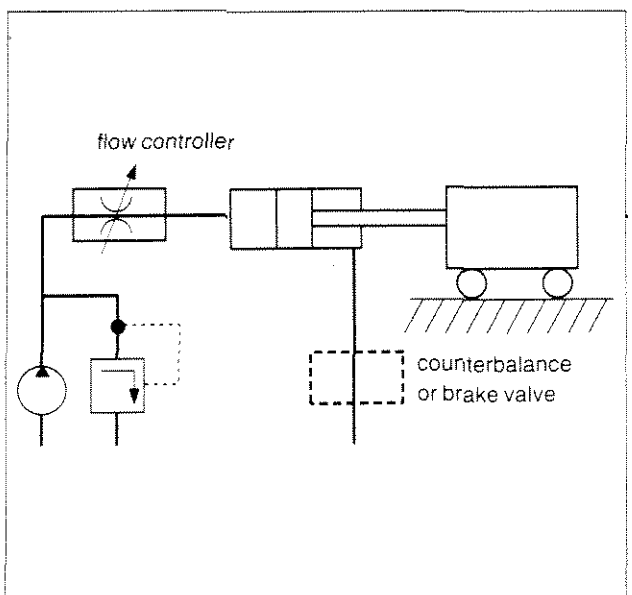

Fig. 15