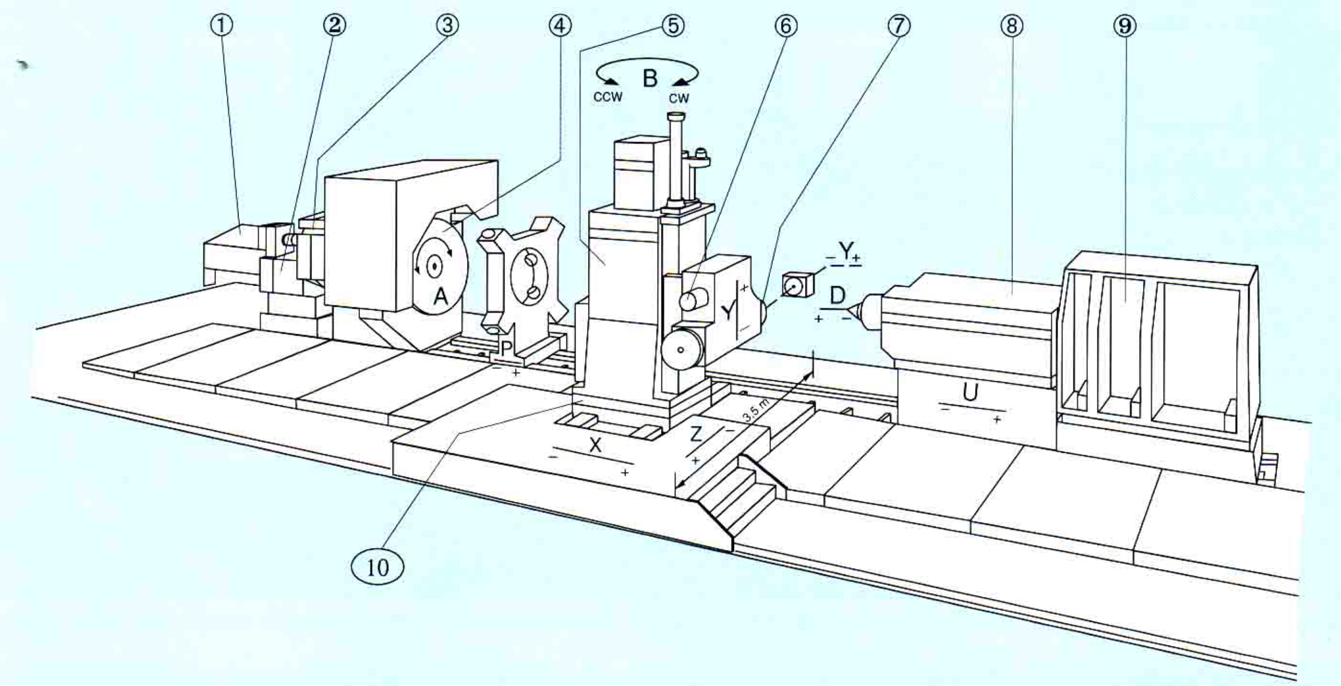
# 5 Application examples of drives with control of the secondary unit

## 5.1 The milling head drive on a machine tool

In a CNC machining centre with a total of 35 hydraulic functions, the main milling spindle drive was equipped with a hydraulic drive having a power of 72 kW under secondary control *(Fig. 17)*.




1 Hydraulic power unit
2 Motor for main spindle drive
3 Headstock
4 Face plate
5 Multi-tower
6 Hydraulic motor
7 Milling head
8 Tail stock
9 Magazine for tool-pallet systems
10 Cross slide

Figure 17: *Overall view of the CNC machining centre.*

During the various operations of this machine, a number of unusual machining processes may take place. Instead of a conventional turning tool, a high powered milling head may be used which means that in addition to the turning of cylindrical, eccentric and elliptical returned parts, other shapes such as polygons, flat areas and grooves can be milled in the stationary workpiece without the need for re-clamping.

The drive motor for the milling head is installed in the so-called multi tower which is mounted on the cross-slide *(Fig. 18)*. It may be slewed separately about its vertical axis. This permits the tool to be positioned at any angle and corner position with regard to the workpiece access.

For physical reasons, the tacho-generator for speed control could not be mounted on the though shaft of the hydraulic motor. It was therefore driven by means of a toothed belt *(Fig. 18)*. The torque of 275 Nm necessary to drive the head was produced by an A4V axial swashplate unit of 90cm$^3$ displacement. For a machine tool, an unusually high operating pressure of 255 bar was used. Maximum speed of the motor was 2850 rpm. The machine specification stated that any speed variation due to load change on the milling head, e.g. due to interrupted cutting must be kept to within 5 rpm.

This was achieved in spite of the great distance between the primary unit and the secondary unit (approximately 80 m) due to the hydraulic spring being eliminated by virtue of the imposed operating pressure. An additional 10 litre accumulator was installed immediately in the vicinity of the secondary unit in order to cater for short term energy peaks required at this point.

One of the main advantages of this hydrostatic drive is the extremely low weight and compact size of the unit employed. A DC motor of certain similar power could not have been installed within the same available space.

A further important point is the high cutting power which could be achieved even at low speeds, thus considerably reducing production times.

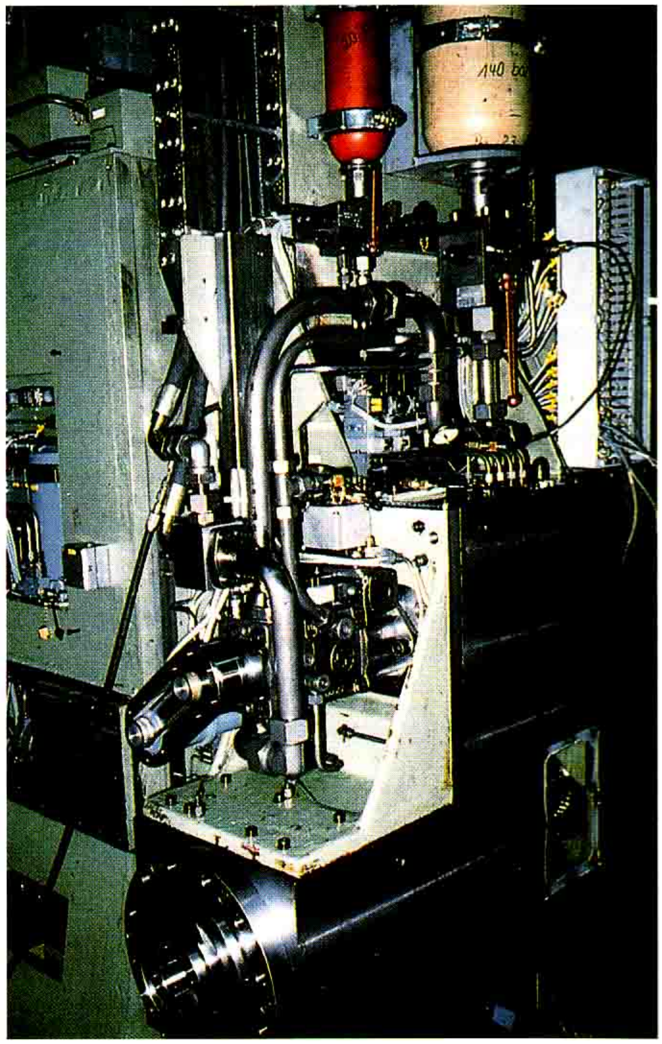


Fig. 18: *Drive motor on the multi-tower*

## 5.2 A drive for a roughing mill

It has also proved possible to utilize the technical advantages of secondary control on the drive of a roughing mill. In such a rolling mill, a number of mill stands are installed in series to reduce the thickness of bars by progressive rolling operations. In this operation, the speed of the rolls increases from stand to stand and the torque reduces as the bar section is reduced. However, the same power is required at each stand.

When rolling, the bar may be subject neither to compression nor tension as otherwise the quality would be reduced. This therefore places high requirements on the dynamics of such drives. *Fig. 19* shows a comparison between the physical sizes of an electrical drive and a hydraulic drive for such a mill stand. In each case, the drive power is between 350 and 400 kW.

Even from the first glance at the comparison, it becomes apparent that this was a test case in which it was being attempted to use a hydraulic drive in place of the existing DC machine. This meant that the interface for the hydraulic motor at the back of the planetary drive already existed. Such a process is not unusual as there is then an underlying assurance that, should the hydraulics not fulfil the requirements, the electric drive can be reinstated.

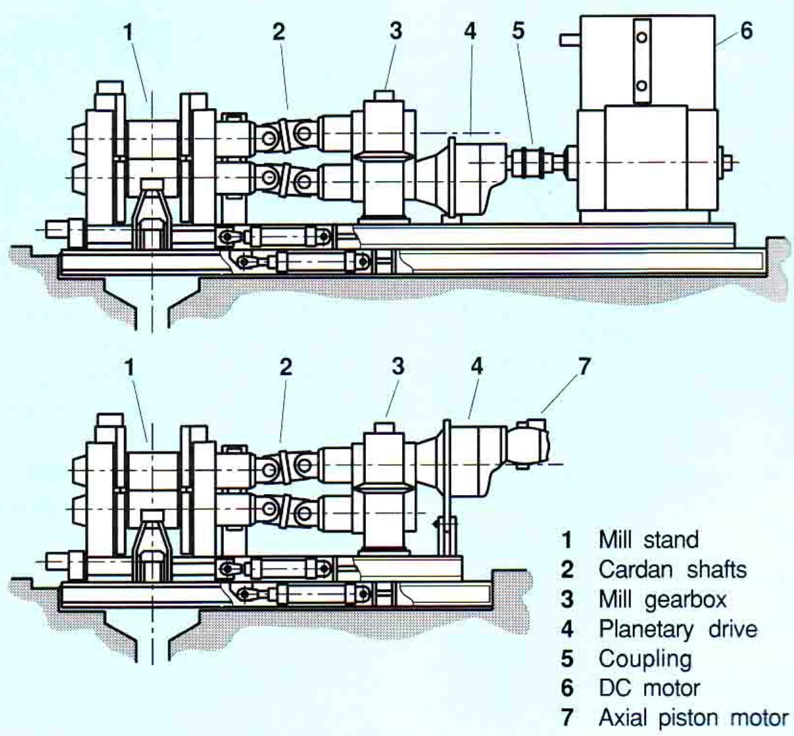



Fig.19: *Roughing mill stand with electrical drive (top) and hydraulic drive (bottom).*

In such a case, it is not surprising that the hydraulic drive was hardly any cheaper, if at all, than its electrical counterpart.

If, however, one utilizes all the possibilities of a hydrostatic drive, the twin output mill gearbox (with a ratio normally of 1:1) together with the cardan shafts can be totally eliminated *(Fig. 20)*.

The axial piston units together with their associated reduction gearboxes can then be built directly onto each roll. As the minimum roll centre distance is pre-defined and cannot be increased, a simple spur gear output can be used should the diameter of the planetary gearboxes prove too large.

As the majority of rolling mill drives have been electrically driven up to the present time, the designer of the hydrostatic drives must utilize the accepted definitions of dynamic characteristics for speed following a stepped load change to VDI/VDE guide-lines 2185.

A measure of the error influences and the quality of rolling stems from the overriding change of speed when the torque suddenly changes is the control envelope $A_1 \cdot t_{settle}$ of the speed control loop *(diagram 7)*.

The load settling time is the time which starts when, after a step in torque, the control value leaves the preset tolerance band and ends when this band is re-entered for the final time on settling.

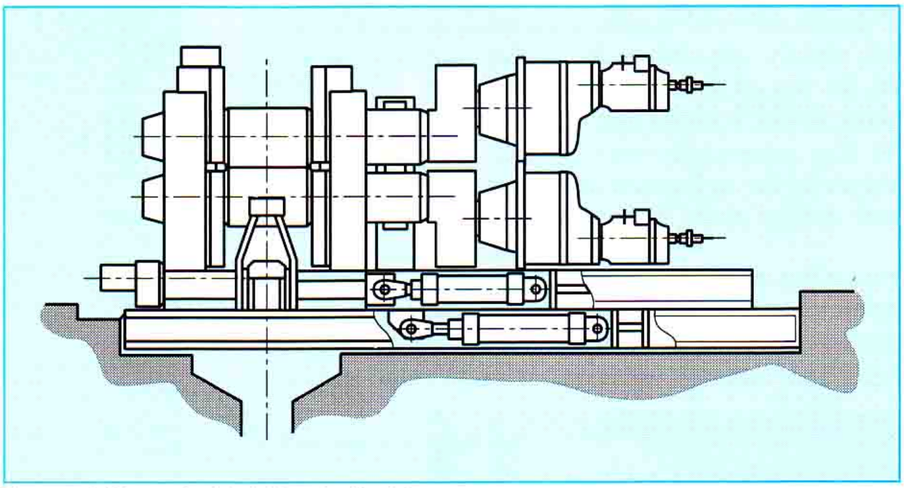
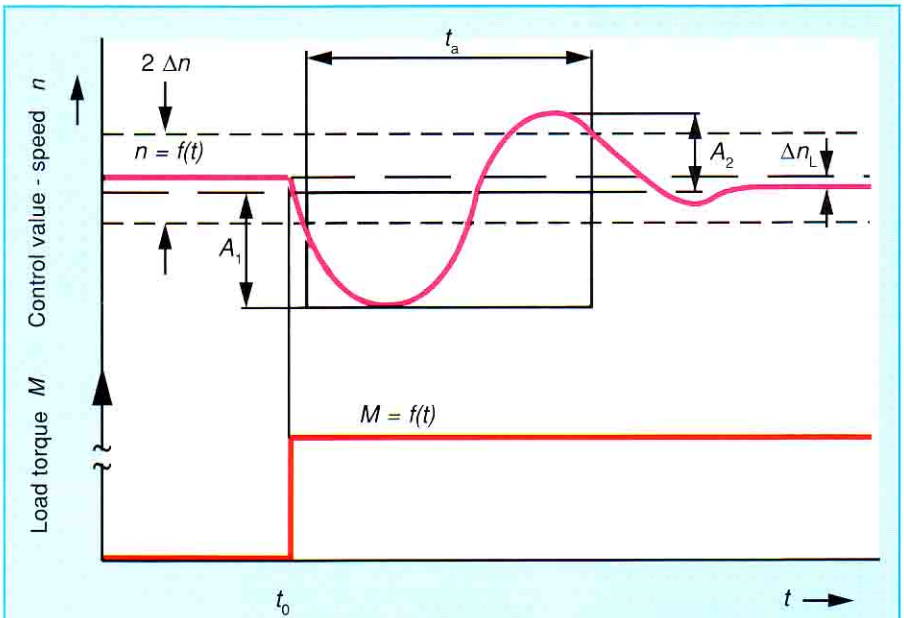


Fig. 20: *Mill stand with full hydraulic drive*




$n$ = Nominal speed
$\Delta n$ = permisible speed variation
$2 \cdot \Delta n$ = permissible tolerance band
$\Delta n_L$ = Speed variations with load
$t_a$ = Load setting time
$A_1$ = Max. speed variation
$A_1 \cdot t_a$ = Control envelope

Diagram 7: *Control variation with a stepped change in torque.*

The control envelope is a product of the load settling time and the greatest variation of the control value from its steady state. The damped oscillation process on a change in load is characterized by the ratio between the amplitude of the first and second control values A2/A1 measured against the final value.

The torque shock in rolling mills occurs when the billet runs into a pair of rolls which are running at a preset speed. As the settling time must be less than 200 ms, and the possible smaller, a drive under secondary control has considerable advantages over the disadvantages of conventional hydrostatic drives as will the made in clear in the following study *(Fig. 21)*.

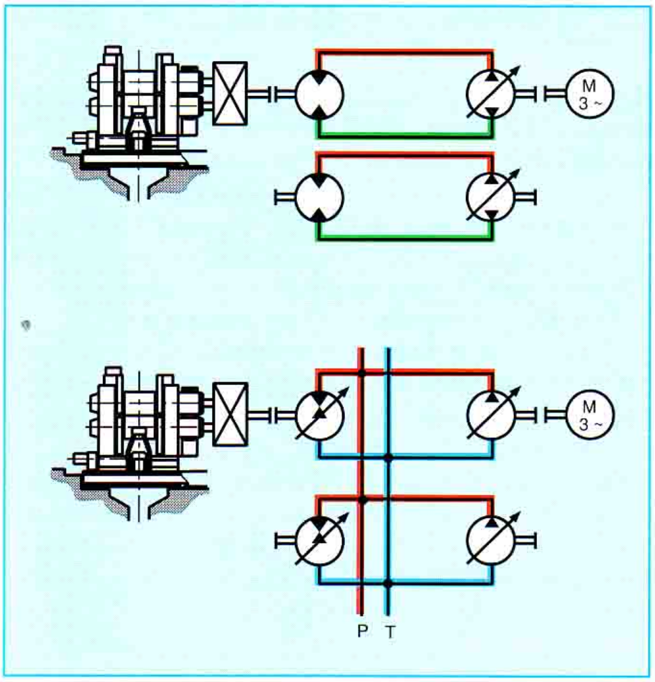



Fig.21: *Mill stand with conventional drive(top) and a drive with secondary control (bottom)*

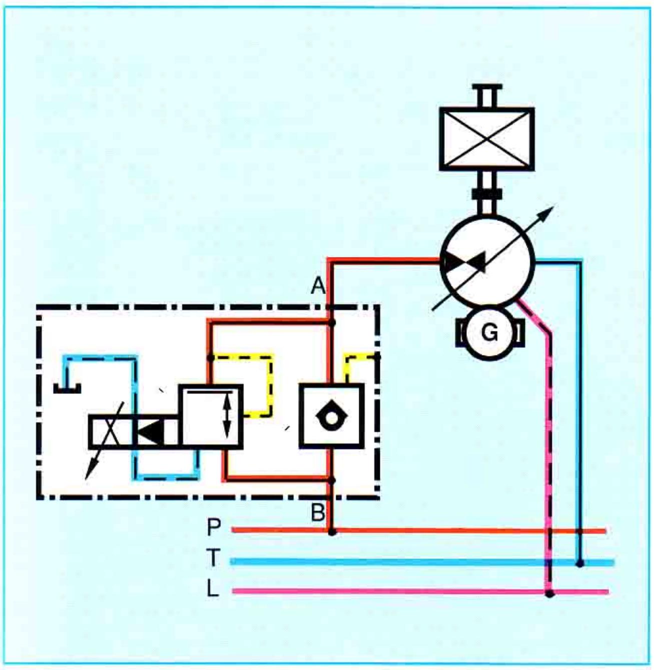



Fig.22: *Auxiliary circuit for a rolling mill drive*

In a conventional hydraulic drive to a mill stand, the speed is determined by the pump which drives a fixed displacement motor on the mill stand *(Fig. 21 top)*. Under light running conditions the pressure difference at the motor is small and the rolls run at the preset speed.

Example for light running operation:

| | |
|---|---|
| Pressure difference | $\Delta p$ = 30 bar |
| Swivel angle of pump | 12 ° |
| Displacement of the pump | 0,8 • $Q_{max}$ |

When the billet enters the rolls the torque rises almost instantaneously. The pressure in the hydraulic system also rises at the same rate to, for example, 280 bar.

As the speed during this process must remain constant, the pump requires to deliver the compression volume in addition to external leakage. This alters the swivel angle from 12° to 13°.

This process is very short term and is over in 15 to 20 ms. The system can then start to oscillate due to the effect of the hydraulic spring and in practice must be damped to prevent this, i.e. the control operation at the pump must be damped so that it has no affect on the settling time of the speed control at the rolls.

Under conditions of secondary control *(Fig. 21 bottom)*, the entry of the billet has quite a different effect. Speed control is performed on the secondary unit at the mill stand itself. An imposed operating pressure is available which is substantially constant and the hydraulic spring is therefore under constant tension. The swivel angle of the secondary unit has adjusted itself to suit light running conditions, for example, at 2 °.

When the torque shock occurs the unit swivels to maximum swivel angle of 15 ° in the minimum possible time and then swivels back to the required steady state value of, for example, 12°.

For this operation, an A4VSO 500 DS1 axial piston unit under secondary control requires approximately 80 ms with a corresponding effect on the rolling technology.

This apparent disadvantage is easy to overcome, however, by means of a trick circuit *(Fig. 22)*. A logic element (3) is built into a manifold (1) directly on the pressure flange of the axial piston unit. An electrically adjustable pressure reducing valve (2) is also installed. The logic element acts as "a hydraulic isolator" in order to be able to isolate the flow of energy during an emergency situation. This same assembly also allows the unit to operate as a generator and return energy to the ring main.

On safety grounds, this logic valve is always installed.

Under light running conditions, the logic element is closed and the operating pressure at pressure flange A is reduced by means of the pressure reducing valve. As the torque requirement under these conditions is unchanged, the swivel angle must therefore be greater. It is in fact increased to a value close to the steady state conditions which occur when a billet is being rolled - this angle is known beforehand - so that the time loss for the change in swivel angle when a torque shock occurs is compensated for. In this case, a valve (3) simply requires to be operated and the displacement of the axial piston unit need only be changed to a small degree.

Any variations from the preset value can be determined and in an adapted control, can be included in the control process.

A tailor made electronic control loop is of paramount importance in drives having secondary control using an electrical tacho and electro hydraulic servo valve in the control loop. This is no doubt clear from the above example. Often, accuracy of control and precision of the drive is exclusively a question of signal processing.

## 5.3 A drive for a large plate rolling mill

The advantage of high speed accuracies and high response in the speed control loop has also been proved in a roller conveyor drive of a large plate rolling mill.

The requirement was to operate 12 motors of varying sizes in synchronism under secondary control. The maximum speed variation —which must be achieved 500 ms after the occurrence of a disturbing torque— was ± 1,5 rpm for all 12 motors. This meant, that all 12 motors had to operate within a tolerance band of 3 rpm absolute. The motors, without any mechanical inter-connections, drive of a total 391 rollers which are grouped together by means of chains in groups of varying numbers. As gear drives with ratio 125:1, 140:1 and 200:1 were interposed between the motors and various groups of rollers, and as these gears were not backlash free, the problem was not made any easier.

Due to the particular requirements, the drives were first of all simulated on a computer with particular regard to the torque shocks to be expected. The computer results were good and the technical risks could be foreseen. It was therefore decided on the mechanical side to install standard motors type A4VSO. In this case, the analogue DC tacho generators had to be replaced by incremental generators.

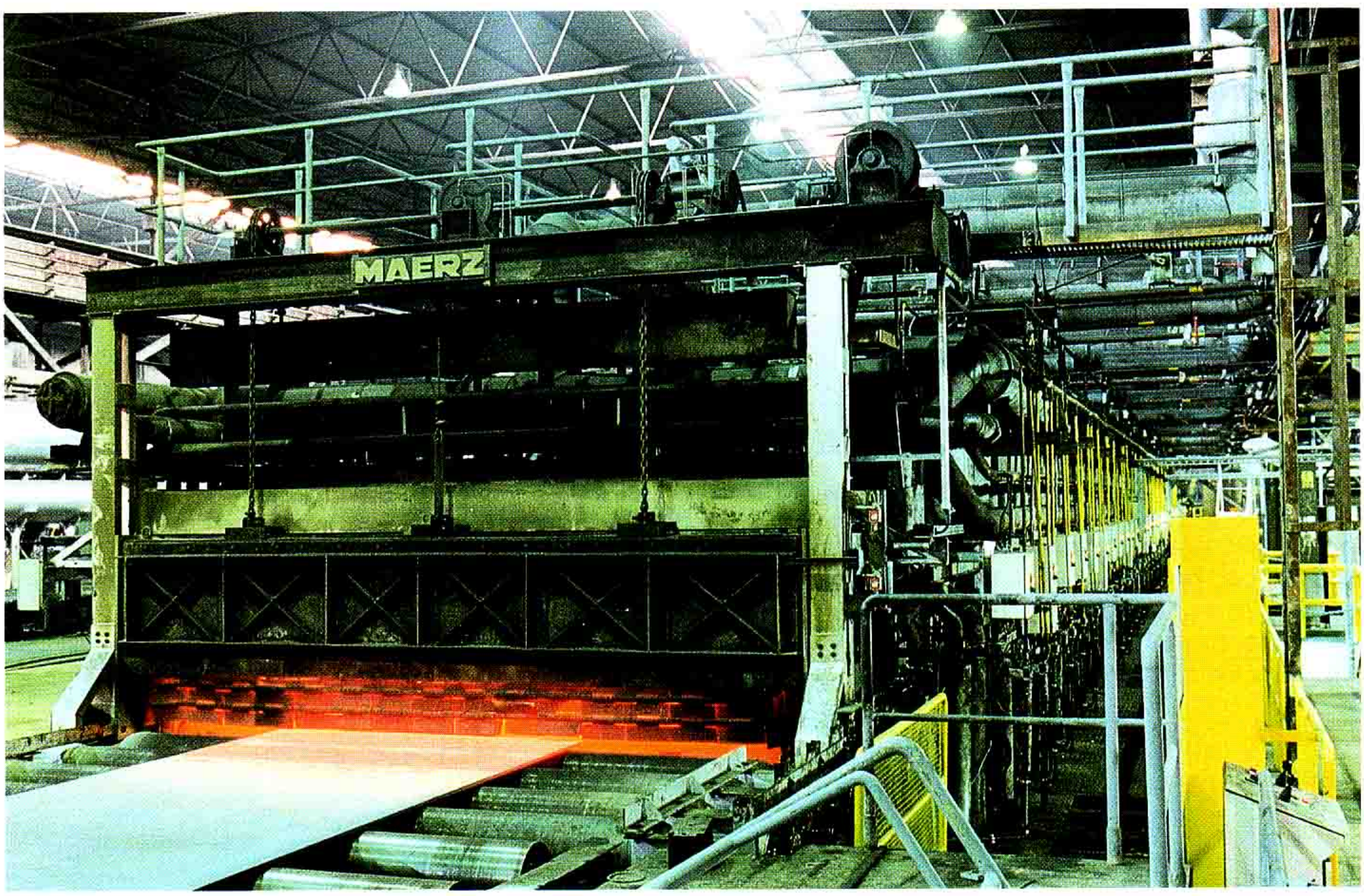



Fig.23: *Heating oven line with roller conveyors, total length 114 m*

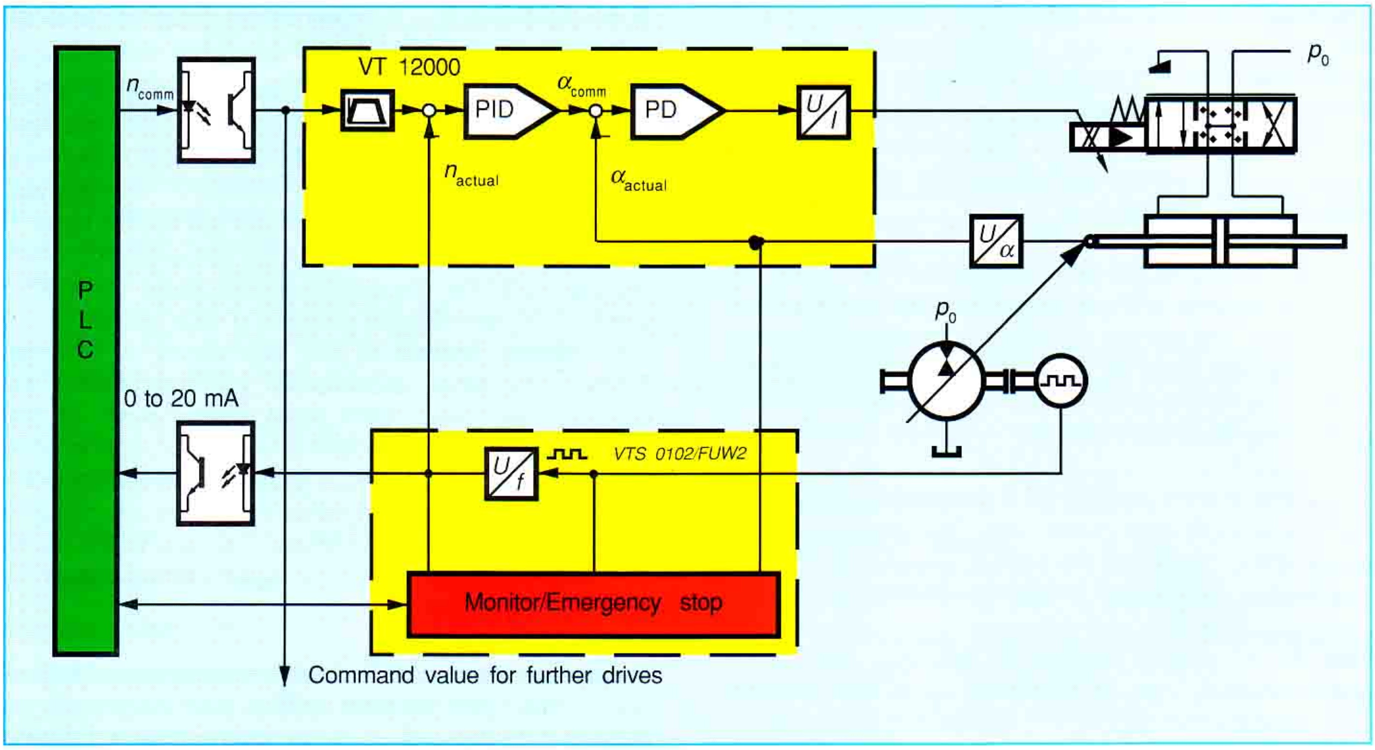



Fig.24: *Combination of control card VT 12 000 and VTS 0102/FUW*

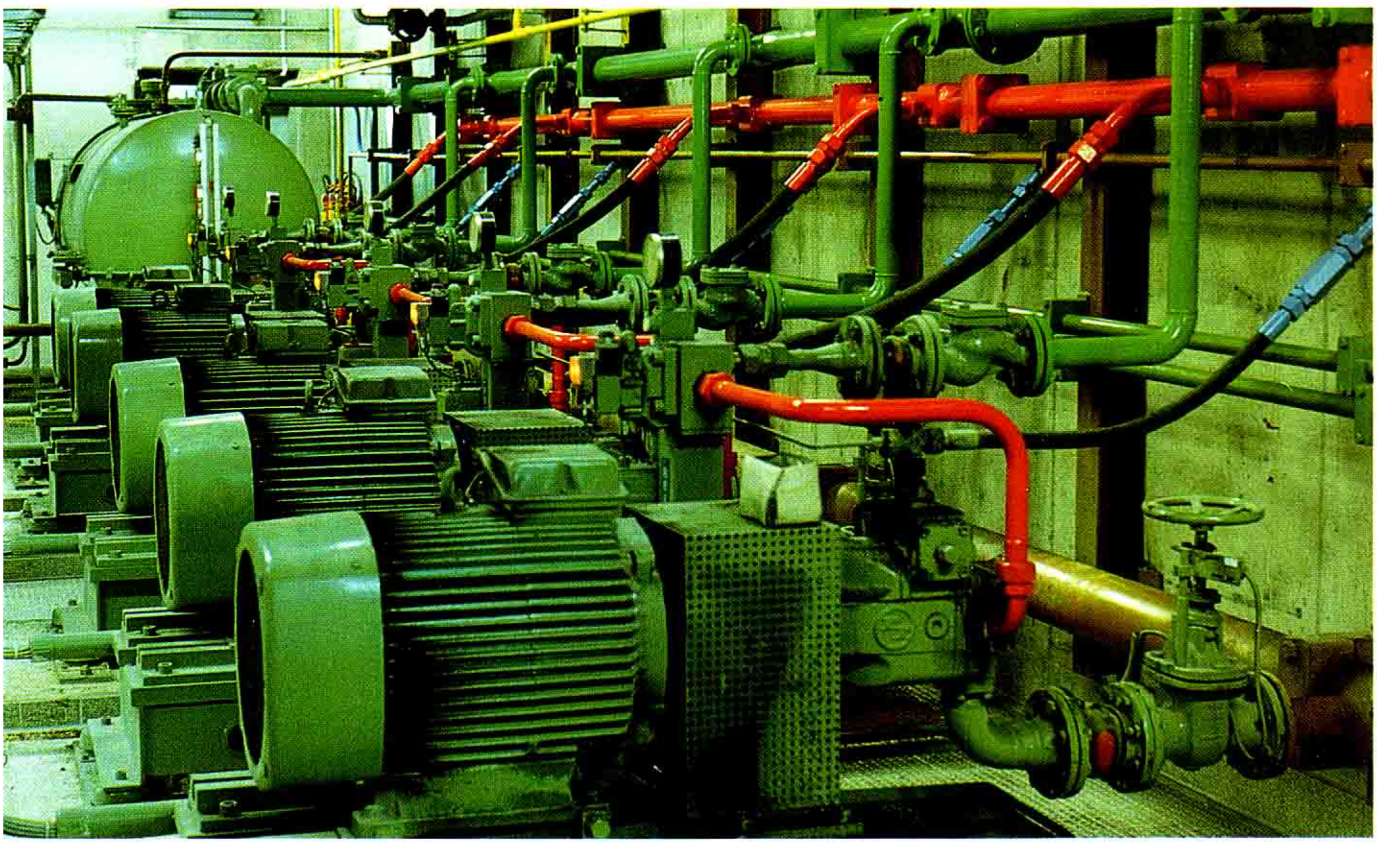


Fig.25: *Central oil supply*

However, there was no requirement to change from the standard control card type VT12000. On the other hand, due to the digital tacho signals the previously described frequency/voltage converter and monitoring electronic card FUW1/VTS 0102 was required *(Fig. 24)*. All the drives were operated by means of a single common command speed value. As the speed variations did not place any special requirement for speed regulation on the secondary control, this was ideally suited for this type of operation, expensive synchronization closed loop controls were not required.

This roller operation was originally equipped with electrical drives which could not fulfil the synchronization requirements. This in turn led to marking of the surface of the stainless steel plates and a reduction in quality. Commissioning of the drives with secondary control by the customer went without a hitch and the tolerance limits were never exceeded.

With an overall length of the installation of 114m, the distance between primary and secondary units could be up to 80m. The central oil supply is installed in a cellar *(Fig. 25).* The distance between the actuators and the supply has no influence on the dynamic operation of a drive using secondary control.

The pump station consists of 5 pressure compensated A4VSO 125 DR axial piston units. The drive power per power unit is 55kW and the working pressure is 160 bar. The system is operated in open circuit.

With the installation installed as described here, good quality products can be produced up to the following dimensions:

| | |
|---|---|
| Plate width | 800 to 3800 mm |
| Plate thickness | 3 to 300 mm |
| Plate length | 6000 to 16000 mm |
| Max. weight per plate | 15 Tonnes |

The speed of the units is a djustable between 0,25 to 15m/min corresponding to total speeds between 34 and 2050 rpm. The maximum temperature is 1200°C.

## 5.4 A drive for a coke oven feed machine

Exact positioning of coke oven feed machines is a prerequisite for automization as the operating crew no longer travels with the machine but sits in a central station and merely perform a monitoring function. The operator therefore required that the maximum permissible value of position from "the oven centre" did not exceed ± 5mm. Having stated this, the affects of heat cause the relative positions of the individual ovens and also position of the oven battery relative to the foundations to change. Furthermore, when accelerating and decelerating the service machines, mechanical deformation of the machine frame can occur and cause the chassis to skew with regard to the tracks. A parallelity correction operation must therefore take place every time the machine stops to ensure that the machine is square to the tracks so that the push rod is not damaged when it enters the opening of the oven.

It is thus quite obvious, that these high technical requirements mean that the most modern technology must be applied and that this must be guaranteed to operate reliably around the clock in spite of the most difficult ambient conditions.

At the project stage, three variations for the transmission drive were considered:

- DC motors.
- Asynchronous motors with a superimposed voltage a
- Hydraulic motors.

At first, hydraulic motors were only considered in a conventional drive circuit. In addition to the transmission, there are number of auxiliary cylinder motions which must be operated when the machine is not travelling.

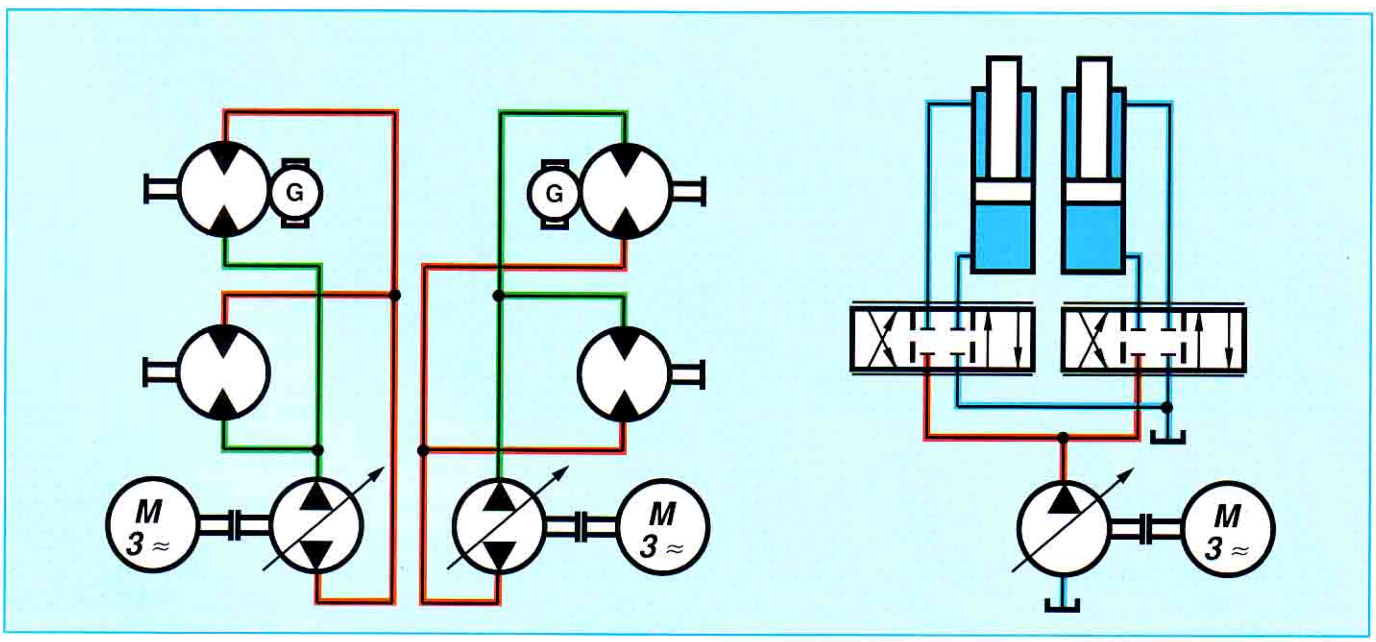



Fig. 26: *Conventional drives in closed (left) and open (right) circuits.*

At first, the drive problem was solved as shown in simplified form in *Fig. 26*. In this solution, the transmission drive was split in two parts. The transmission speed was "flow coupled" via the displacement of the two pumps which also catered for the synchronization. Speed feedback was achieved by means of a tachometer on each side. These were built onto the hydraulic motors or, to avoid slip onto a non-driven wheel.

A further open circuit was included with a pressure compensated pump for operating the cylinders. The control of these was via proportional valves.

The disadvantages of this design were:

- Three hydraulic circuits, each with its own pump.
- Three electric motors (double shaft)
- Expensive piping
- Poor redundancy
- Due to the hydraulic spring in the transmission circuit, this could tend to oscillate due to the high response of the closed loop control and could effect the positional accuracy.
- In the partial load range, at low pressures and high oil velocities, the efficiency was poor.

The situation could be improved by introducing a constant pressure with pressure compensated pumps operating in open loop as shown in *Fig. 27*.

The pumps deliver only the flow required and demanded by the actuators in order to keep the operating pressure constant. However, in order to obey the laws of flow coupling, proportional valves or similar must be incorporated into the energy carrying lines.

The advantages of this system compared to *Fig. 26* are:

- Only two pumps with electric motors are required.
- The system can be operated by a single pump (good redundancy).
- The flow requirements of the cylinders is not required as an extra, as the machine is either operated or manipulated.
- Reduced piping.
- The hydraulic spring is shortened (now only between the motors and the proportional or servo valves).
- Improved oscillation characteristics.

Disadvantages:

- Poor energy balance.
- Braking power is converted into heat.
- In the partial load range the pressure difference at the valves is converted into heat.
- All valves arranged in the energy flow path.
- In addition, the pressure drop at the valves must be generated at the pumps (in case of servo valves up to 70 bar).
- A high cooling power is required.

After weighing up all the advantages and disadvantages the results of the preliminary survey leaned towards a control electric drive.

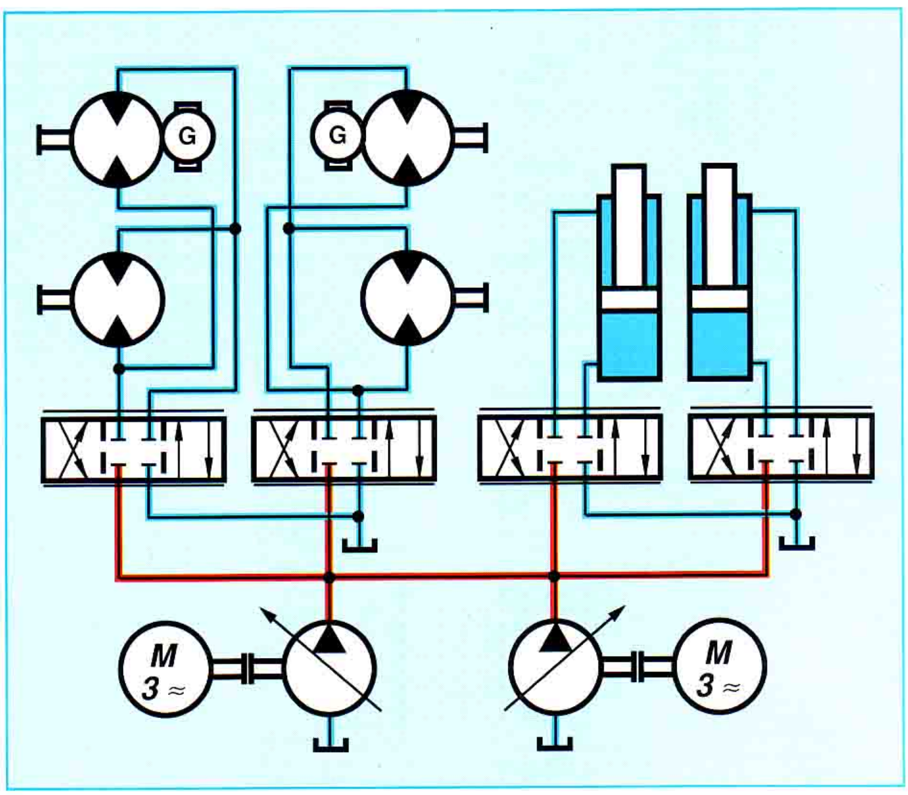



Fig. 27: *Conventional drive in open circuit*

However, at this point in time systems with secondary control made their appearance although they then formed a new drive concept with few references to their credit.

After critical analysis of the system with secondary control a decision was made to use this system as shown in *Fig. 28.* The decisive factors were not so much the possibility of energy recovery during the deceleration phase, but more the advantages of control and the fulfilment of requirements for high positional accuracy.

The advantages of secondary control in this instance were:

- One or two pumps/electric motors (redundancy)
- The flow requirement of the cylinders did not need to be considered.
- Low cost piping.
- The hydraulic spring had no effect on the dynamics and oscillation characteristics of the system.
- Good energy balance as no control devices interfere with the flow of energy.
- Minimum cooling power required.
- The deceleration energy can be stored and re-used for the next acceleration phase.
- The electrical drive power can be reduced.
- The control range is extremely wide and
- The braking energy can be returned to the electrical power lines if required.

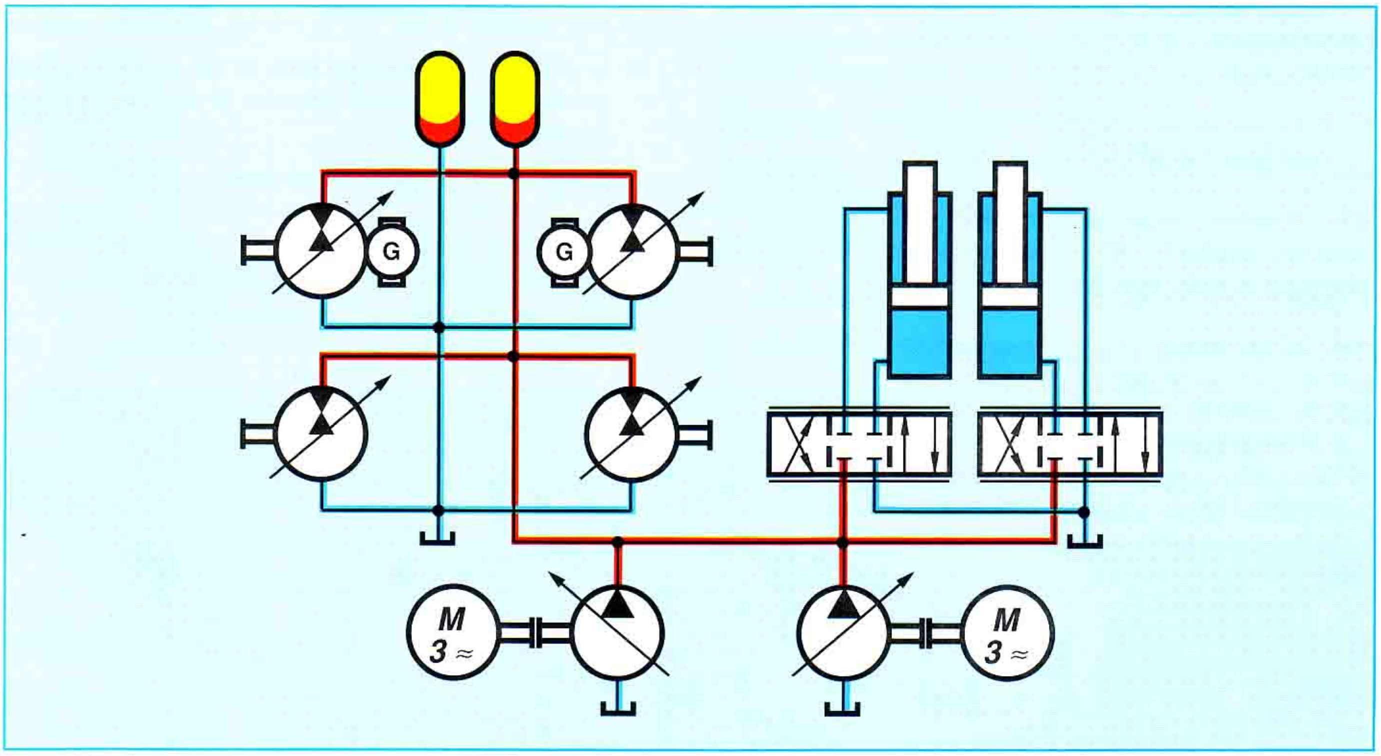



Fig. 28: *Drive with secondary control with pressure coupling in open circuit.*

The transmission drives for the coke injection machine *(Fig. 29)* and the coke carry over machine *(Fig. 30)* were therefore equipped with motors under secondary control. The dynamic response of this drive system showed that it could achieve a positional accuracy of ± 1mm with a machine weight of 6600kN for the ejection machine. The highest speed of the machine was 1,5m/s. The transmission motors are coupled to the wheels via gearboxes *(Fig. 31)*.

As these gearboxes are not backlash free, this has a negative effect on the dynamics of the control and on the positional accuracy. The influence of this backlash must therefore be eliminated from the system characteristics. This is achieved by means of hydraulically tensioning the two motors with respect to one another. In this way, the flank contact of one set of gears is opposed to that of the other set. As the diameter of the drive wheels is 1250 mm, a positional accuracy of 1mm is equivalent to an angular rotation of the wheel of 0,09°.

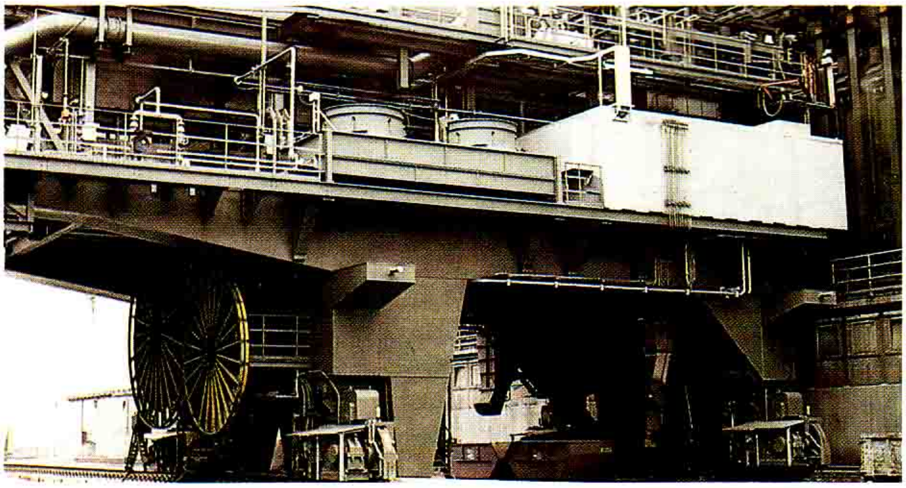


Fig. 29: *Coke ejection machine*

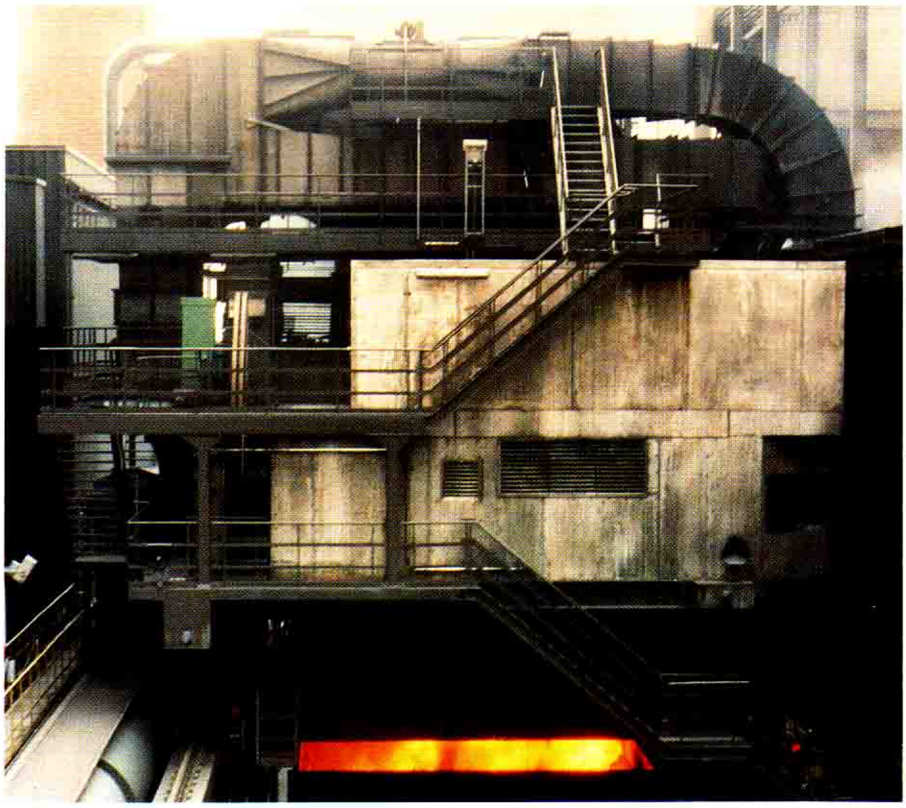


Fig. 30: *Coke carry-over machine*

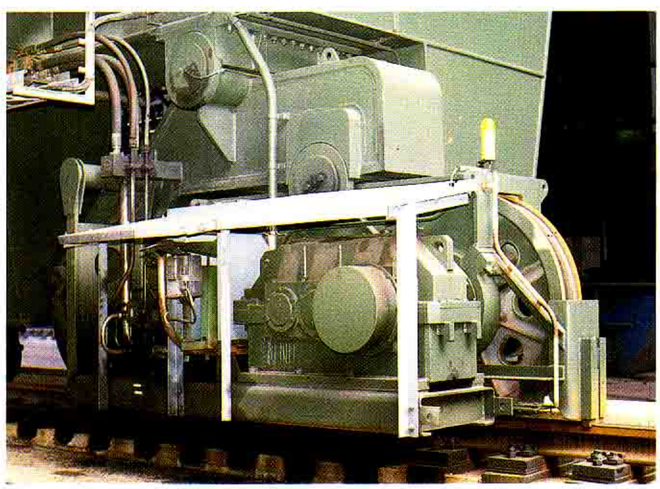


Fig. 31: *Drive for the coke ejection machine*

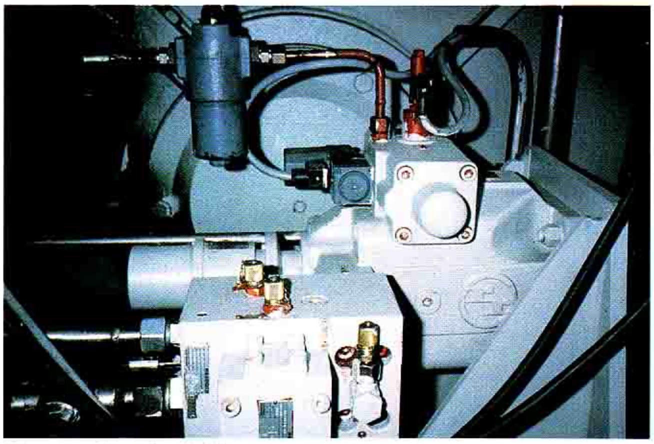


Fig. 32: *Standard drive motor*

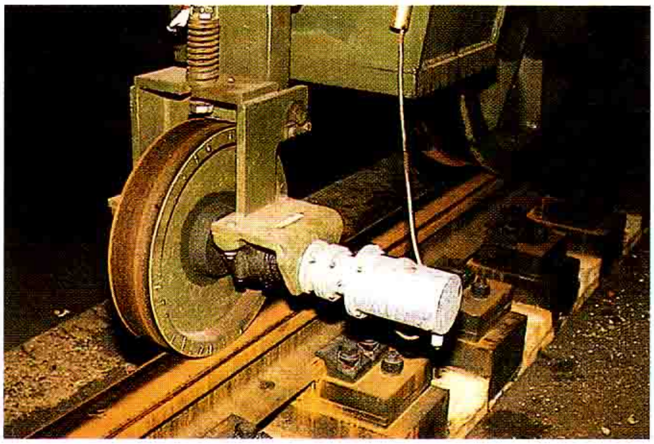


Fig. 33: *Arrangement of the absolute positional transducer*

*Fig. 32* shows a transmission motor of standard design with built on servo valve, swivel angle feedback and a built on hydraulic isolator. The motor is equipped with an analogue tacho generator with integral mechanical centrifugal switch. The speed and synchronization control are both performed on an analogue basis. The distance traveled is determined by an incremental impulse generator mounted on a non-driven wheel *(Fig. 33)*. The approach and final positioning is achieved under digital control.

The excess energy present during the deceleration phase is stored for use in the next acceleration phase or, should the accumulator be full, returned to the electrical power line. Should this be necessary, the pressure controlled pumps act as motors and drive the three phase electric motor at above synchrous speed thus feeding power into the power lines.

Operational experience gathered since the end of 1986 show that the drive concept selected has fulfilled all expectations under the given drive conditions. In particular, the control characteristics in automatic operation, for which the internal positional control system is fed through an overriding operating system in the positioning phase allows the positioning accuracy to be achieved with ease.

## 5.5 A drive for a mobile manipulator

In this exercise, a mobile manipulator *(Fig. 34)* was required to move a ring weighing 100kN from a press and to place this in a ring rolling machine some 40 m away within the shortest possible time. The positional accuracy was preset at ± 2mm. In addition to the lifting and tipping cylinders, the transmission and slew drives also had to be operated. Once more, it was decided to use a system with secondary control as simulation calculation showed that this could maintain the preset tolerances. The overall weight of these vehicles with load is 530 kN. A maximum speed of 2,7 m/s is required with an acceleration of 0,85m/s². Although the installed corner power through the actuators taken together amount to approx. 600 kW, the energy recovery and storage achieved in the transmission and through the use of secondary control made it necessary only to install a diesel engine having a power of 143 kW at 1600 rpm.


Fig. 34: *Mobile manipulator in operation*

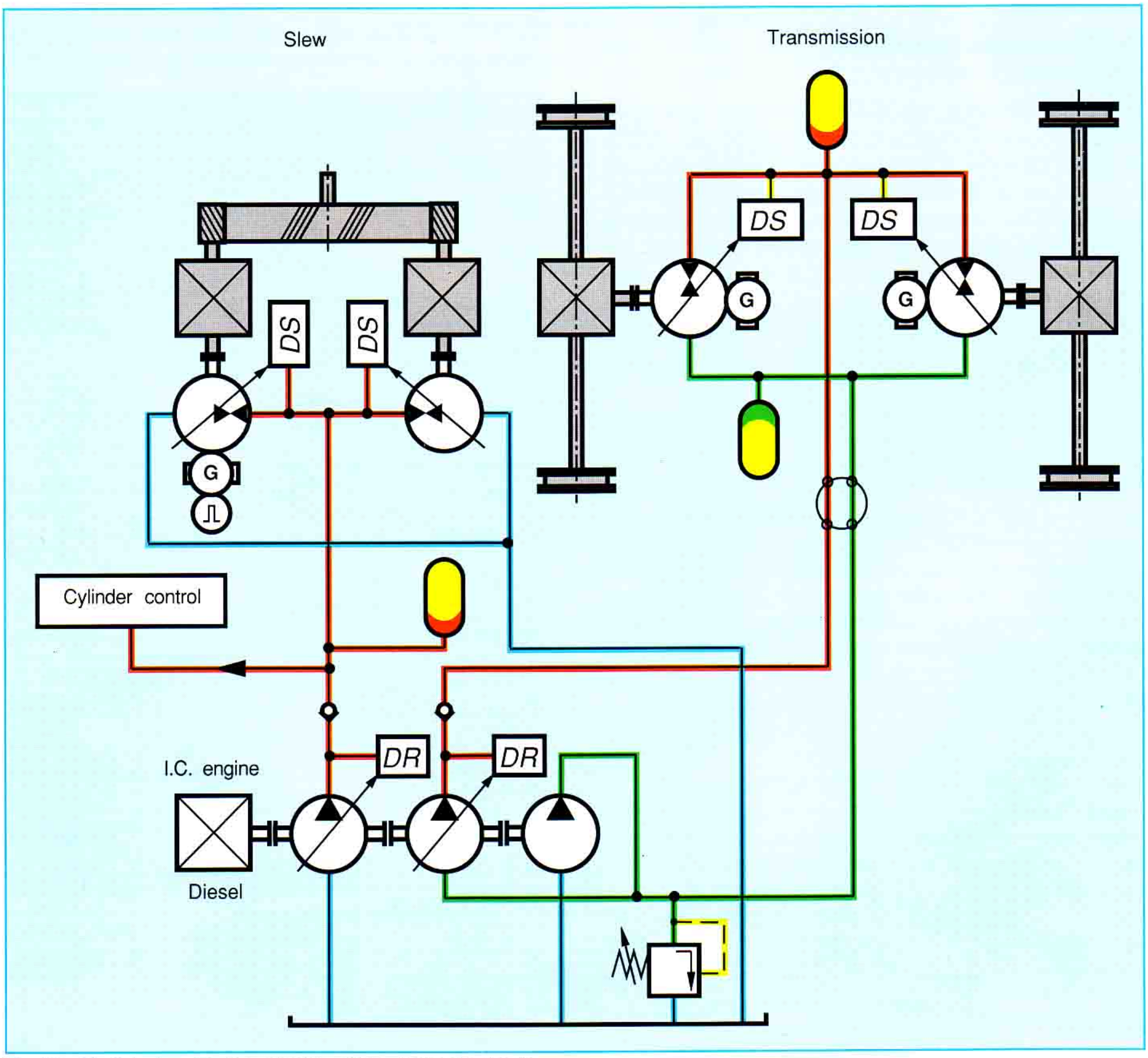



Fig. 35: *Sketch showing the principle of the hydraulic circuit*

*Fig. 35* shows the principle of the circuit. The diesel engine drives a tandem pump A4VO 250 DR + A4VSO 125 DR onto which the boost pump for the closed transmission circuit is mounted *(Fig. 36)*. The slew drive and the cylinders are operated in an open circuit. All motors under secondary control are of the same type —A4VSO 125 DS. On the slew drive, an incremental generator is installed in addition to the analogue tacho purely for the purposes of position. Once more, a motor drive was applied so that gear play could be eliminated in the subsidiary swivel angle position controlled loop. *Fig. 37* shows one of the motors for the slew drive.

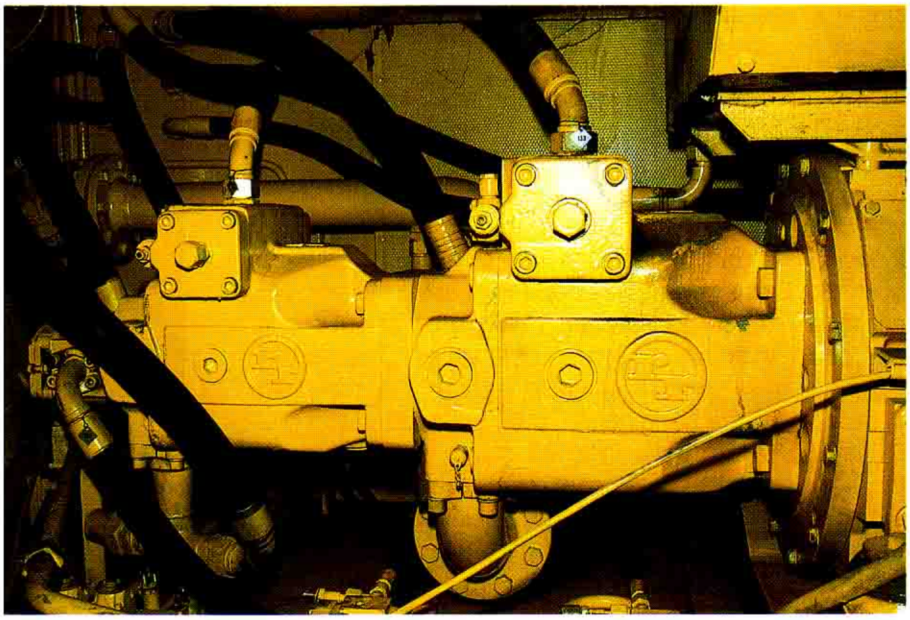


Fig. 36: *Diesel engine with tandem pump*

The transmission operation can be described as follows:

During the previous hydrostatic deceleration operation, the accumulator system was loaded to 260 bar. At this operating pressure a very high rate of acceleration can be achieved as both drive motors are simultaneously swivelled to the maximum swivel angle of 15°. During this operation, the accumulator pressure falls to 200 bar. Up to this point in time, the pump is still at zero stroke. Below 200 bar, the pump swivels out and if the acceleration process is not complete, the operating pressure falls further to 195 bar.

When the travel speed has been reached the motors swivel down to approx. 12% displacement in order to overcome rolling resistance. The flow requirement therefore falls so that the pump also swivels back to approx. 30% of its displacement. During the deceleration phase the motors swivels over zero in the opposite direction and act as generators thus recharging accumulator system up to 260 bar. During the deceleration period, the pressure controlled pumps swivel back to zero stroke. Energy recovery is also achieved in the slewing operation and when the cylinders are operated. In this case, the deceleration energy of the slew is either stored or fed to the cylinders.

This continual transfer of energy without the need to convert the energy into another form reduces the primary power required from the diesel engine and reduces the amount of unnecessary heat produced which would otherwise have to be eliminated by the use of a heat exchanger. As a result, the heat exchanger could be kept reasonably small.

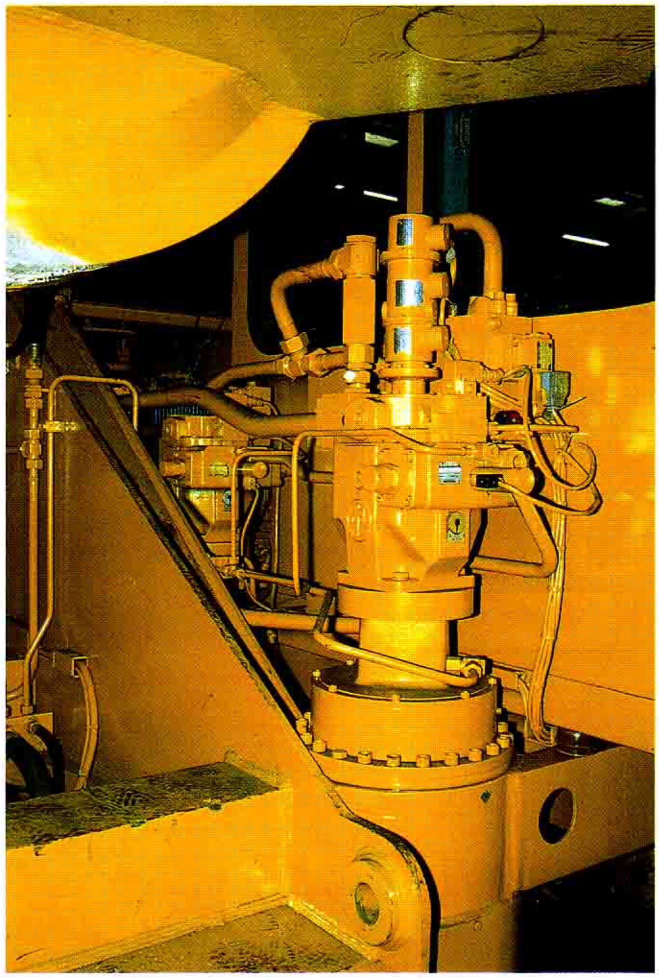


Fig. 37: *Drive motor for slew*

## 5.6 A drive for a bucket wheel excavator

Right at the start of development, a system with secondary control was installed in the in a bucket wheel excavator as shown in *Fig. 38*. The device, with an operational weight of approx. 4500kN has a capacity of 1390m$^3$/h and a drive power of 2 x 300kW.

Although conventional hydrostatic drives with control of the primary unit have been employed in bucket wheel excavators since 1972 high powered devices have preferably been equipped with DC motors.

The disadvantage of this type of drive is the high weight of DC motors and the necessary gearboxes. In this respect, it should be noted, that any additional weight at the bucket wheel head leads to an increase in the ballast weight and an associated strengthening in the design which can have a three or four fold factor on the overall weight of the excavator. This in turn leads to additional costs in addition to the high cost for the DC drive itself.

In order to avoid the disadvantages mentioned above, a hydrostatic drive with four hydraulic motors and individual gearboxes was selected for the bucket wheel drive.

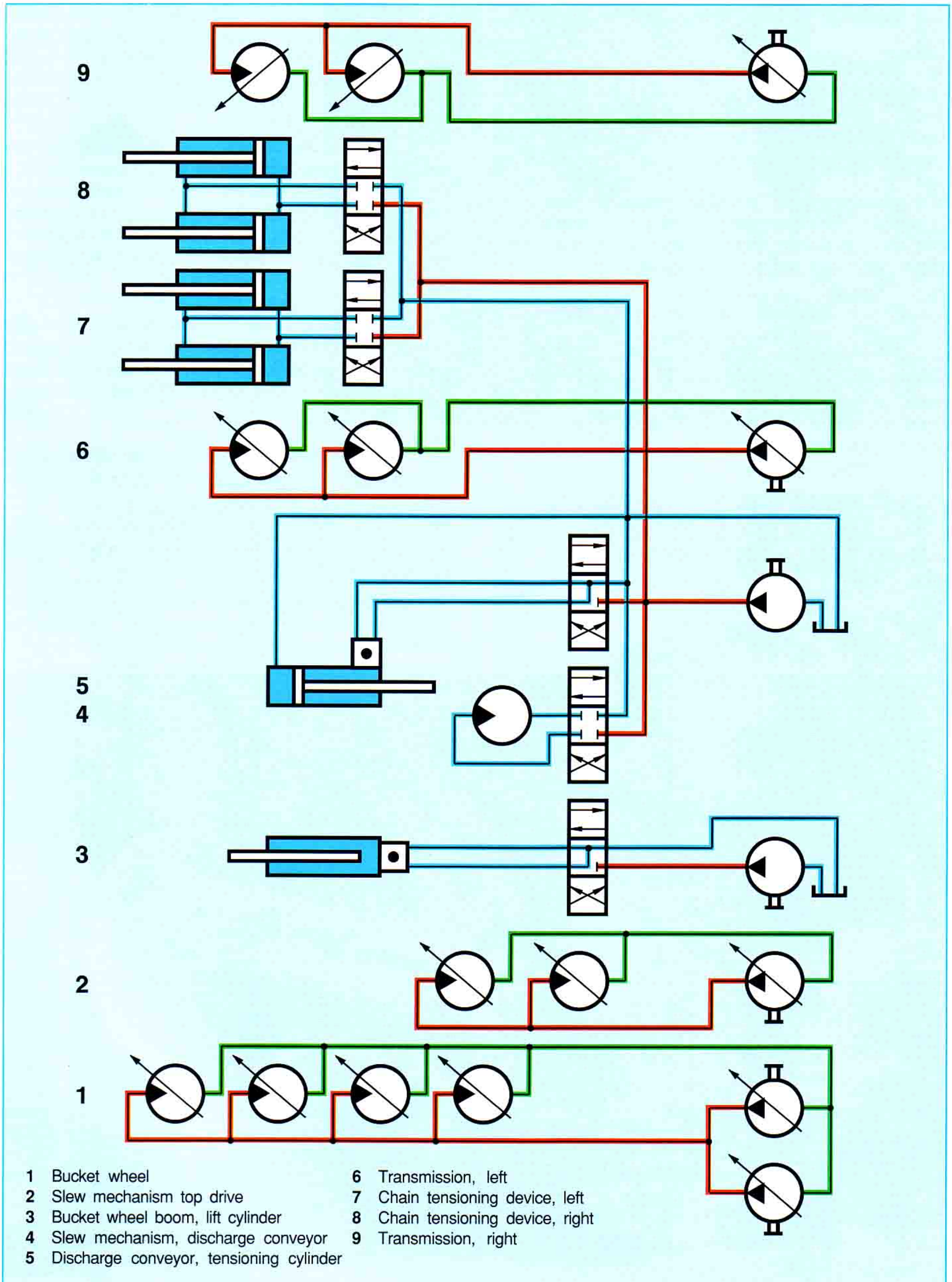
Due to the stepless controllability of speed over the whole speed range, optimum excavation and emptying of the bucket can be achieved under the most widely varied ground conditions. Stones and rocks can be carefully excavated.

Due to the elimination of the bucket wheel shaft and the replacement of the large bucket wheel gearbox by four individual gearboxes and the electric motor by four hydraulic motors, a considerable cost saving could be made compared with a comparable electrical drive system. In addition, the weight at the bucket wheel head was produced by 45kN.

In total, the operational weight of the bucket wheel excavator could be reduced by approximately 1350 kN due to the use of secondary control. In this, the reduced amount of piping due to the common oil supply played made no mean part.

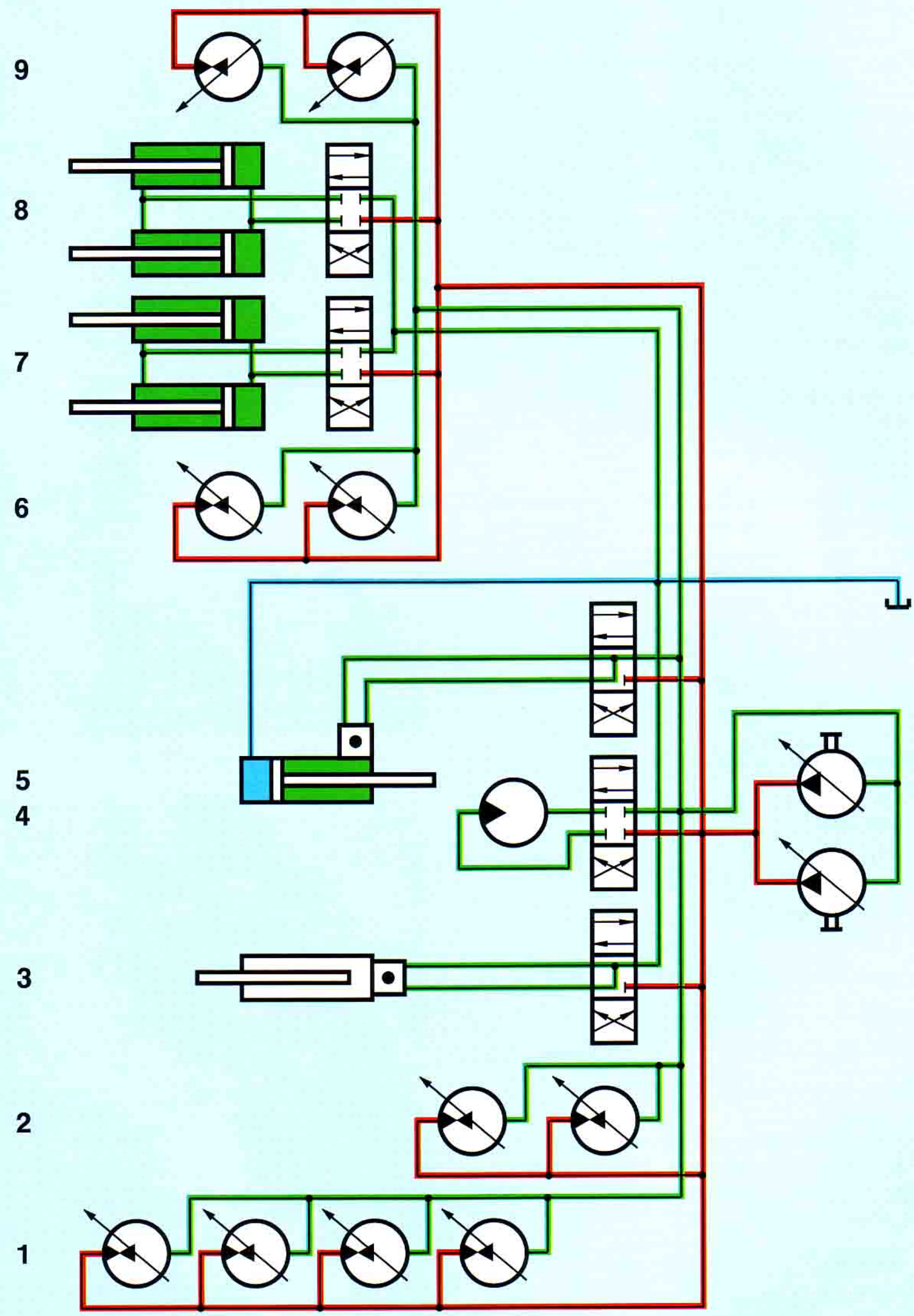
*Figs. 39 and 40* show a comparison of the two hydrostatic drive concepts - primary control and secondary control. In the system with secondary control, the primary end is limited to 2 pumps. In a conventional drive system a total of 7 pumps would be required. The auxiliary pumps for the boost and pilot systems are not shown.


Fig.38: *Bucket wheel excavator*




1 Bucket wheel
2 Slew mechanism top drive
3 Bucket wheel boom, lift cylinder
4 Slew mechanism, discharge conveyor
5 Discharge conveyor, tensioning cylinder
6 Transmission, left
7 Chain tensioning device, left
8 Chain tensioning device, right
9 Transmission, right

Fig. 39: *Drive for bucket wheel excavator, simplified illustration of a conventional circuit.*




1 Bucket wheel
2 Slew mechanism, top drive
3 Bucket wheel boom, lift cylinder
4 Slew mechanism, discharge conveyor
5 Abwurfband, Zugzylinder
6 Transmission, left
7 Chain tensioning device, left
8 Chain tensioning device, tight
9 Transmission, right

Fig. 40: *Drive for bucket wheel excavator, simplified illustration of a drive with secondary control.*

In the case of the system with secondary control, redundancy is higher as, should one pump fail, the machine can still be operated at half power. Alternatively, redundancy can be further extended by the use of more pumps. As the oil column operates under conditions of nominally constant pressure the position of the primary unit with regard to the actuators is freely selectable, e.g. it can be used as the counter weight for the bucket wheel.

The material costs at the secondary end in a secondary control system are greater as axial piston units controllable over zero must be installed. However, the advantages in a system under primary control are not always what they may seem as variable motors are often employed in conventional systems.

In mobile and construction machines it makes little sense to equip an existing machine, already using a conventional system, with drives with secondary control.

The advantages shown here are only fully realizable if the design requirements can be met more fully and if the device can be re-designed accordingly. This becomes more relevant, when a greater number of actuators are attached to the hydraulic power lines. Also, dependent upon the duration of operation and the degree of parallel operation, the investment costs at the primary end can be reduced.

Fig. 41: *3000 kN tower crane on a pipe layer*

## 5.7 A drive for an offshore crane

The installation and operation of devices underwater require modern lifting equipment. This must be capable of the following duties under the rough working conditions found at sea:

- Laying of pipelines.
- The milling of trenches on the sea bed.
- The installation of underwater manifold assemblies.
- Underwater piling and
- The support of submersible vehicles and divers.

A Dutch company have developed a new generation of offshore cranes for these duties with the help of which work can be carried on in water depths of up to 400 m even under adverse weather conditions.

In order to fulfil the high requirements which are placed upon offshore cranes, hydraulic drives with secondary are once more installed.


Fig. 42: *700 kW crane installation on a semi-submersible.*

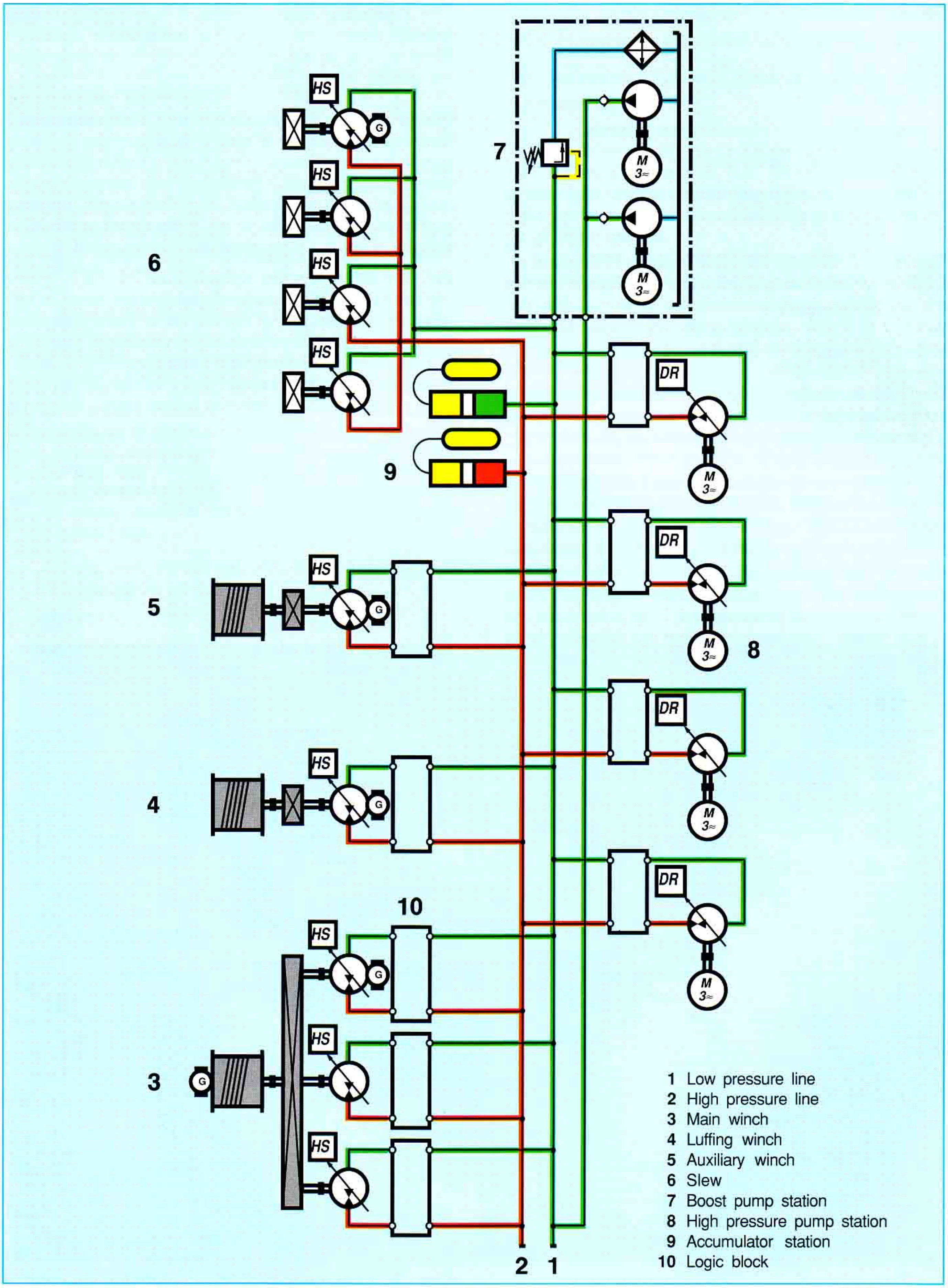



Fig. 43: *Hydraulic system for the offshore crane.*

The decisive factors for this decision were:

- The lower level of installed power required.
- The very short response times of the closed loop control.
- The high dynamic characteristics achieved.
- Energy recovery when lowering loads.
- The possibility of storing energy without the need to convert this to another form.

*Figs. 41 and 42* show two 700kW crane installations on a pipe layer and a semi-submersible. How these have not only fulfilled the requirements placed upon them but also demonstrated their reliability under continuous operation is explained below.

The hydraulic system in *Fig. 43* once more consists of a central oil supply and a common ring main system for all actuators. It is operated as a closed circuit.

The question as to whether an open or closed circuit should be chosen is solely dependent upon the position of the oil tank with regard to the low pressure connection of the secondary units and if these must act as generators when the load is being lowered. The reason for this is that a secondary unit must receive a flow of oil under these conditions on the low pressure side. If, as in this case, the tank is set at a much lower level than the secondary units a closed circuit or a prefilled circuit must be used.

The following actuators were driven:

- The main winch.
- The auxiliary winch.
- The topping winch.
- The slew mechanism.
- The positioning winches.

The hydraulic system remains fully functional but at reduced power even if only one pump station is operational as long as the operating pressure remains within the correct range.

Energy recovery when lowering loads or decelerating the slew mechanism leads to energy storage in the hydraulic accumulator system or a feedback of energy into the electrical power system on board. The requirements for a constant pull on the line, active stroke compensation and high position accuracy mean that the short reaction times and dynamic response of secondary control are definitely necessary. The movement of the ship is measured by means of an accelerometer and converted to suitable movements of the derrick via a micro-processor. The energy recovered from wave movement makes a possible saving in primary energy of up to 50% possible. In this way, in spite of large power variations, the electrical power lines on board are evenly loaded. This sea following installation permits the crane to be operated in winds up to force 5. Due to the high powers required, axial piston units with secondary control series A2P1000HS are installed with a built on tacho unit on the auxiliary drive.

## 5.8 Drives for Dynamic Test Stands

The trend in automobile manufacture is unmistakable, it is to reduce the testing of full production models on the roads from its present level of 80%. For this to become possible, testing and optimization must be possible in the laboratory. This in turn places high demands on simulation technology and in particular on the dynamics employed. The simulation of driving conditions on suitable test stands brings with it the advantages both of reduced testing times and an increase in the number of tests possible.

The requirement to test under real conditions (which occur in the vehicle on the road) in the laboratory refers to components, assemblies and structures right through to complete vehicles. In this and dependent upon the flow of energy, three testing principle can be defined:

- Pure braking systems
- Systems with energy feedback in which only the energy losses need be used as the "power input" as the energy output of the test specimen is re-used as a second input
- Systems with energy recovery into the hydraulic or electrical power lines.

If in pure braking systems, water turbulence or eddy currents brakes are used, hydrostatic drives in either a conventional form or under secondary control can be considered in competition to DC machines or frequency controlled AC machines.

Table 3 shows a comparison of speed drive systems such as are used on test stands. It will be easily seen, that due to the dynamic response in four quadrant drive, that the axial piston machine under secondary control is superior to all other systems.

From the point of energy recovery advantages are to be found.

When feeding energy back into the electrical power line, the AC motor is driven by the hydrostatic unit at above synchronous speed. It therefore acts as a generator, generating pure sinusoidal current. Dynamic operating conditions are covered by the hydraulic accumulator. The electrical power line is thus not subject to overload peaks. The advantage of hydraulic energy recovery and storage without converting the energy into another from can be utilized if all the test stands of a testing centre are connected to a common hydraulic system.

| | Eddy current power brake | DC machine | Axial piston machine with secondary control |
|---|---|---|---|
| Internal moment of inertia | 30 | 80 | 1 |
| Space required on test stand | 55 % | 210 % | 100 % |
| Space required in total installation | 15 % | 120 % | 100 % |
| Direction of rotation | Bi-directional without loss of power. | | |
| Direction of torque | only load torque | Four quadrant operation | |
| Dynamic response | suitable | medium | high |
| Energy recovery | heat energy | electrical energy | electrical energy, hydraulic accumulator, energy return to central hydraulic installation |
| Quality of recovered electrical energy | not possible | not pure sine wave due to chopper | pure sine wave due to three phase machine |
| Loading of the electrical power lines for dynamic tests | possible | high due to load peaks | no load peaks, due to built-on hydraulic accumulator |
| Price comparison | 30 % | 120 % | 100 % |

Table 3: *Comparison of brake/ motor/ generator units*

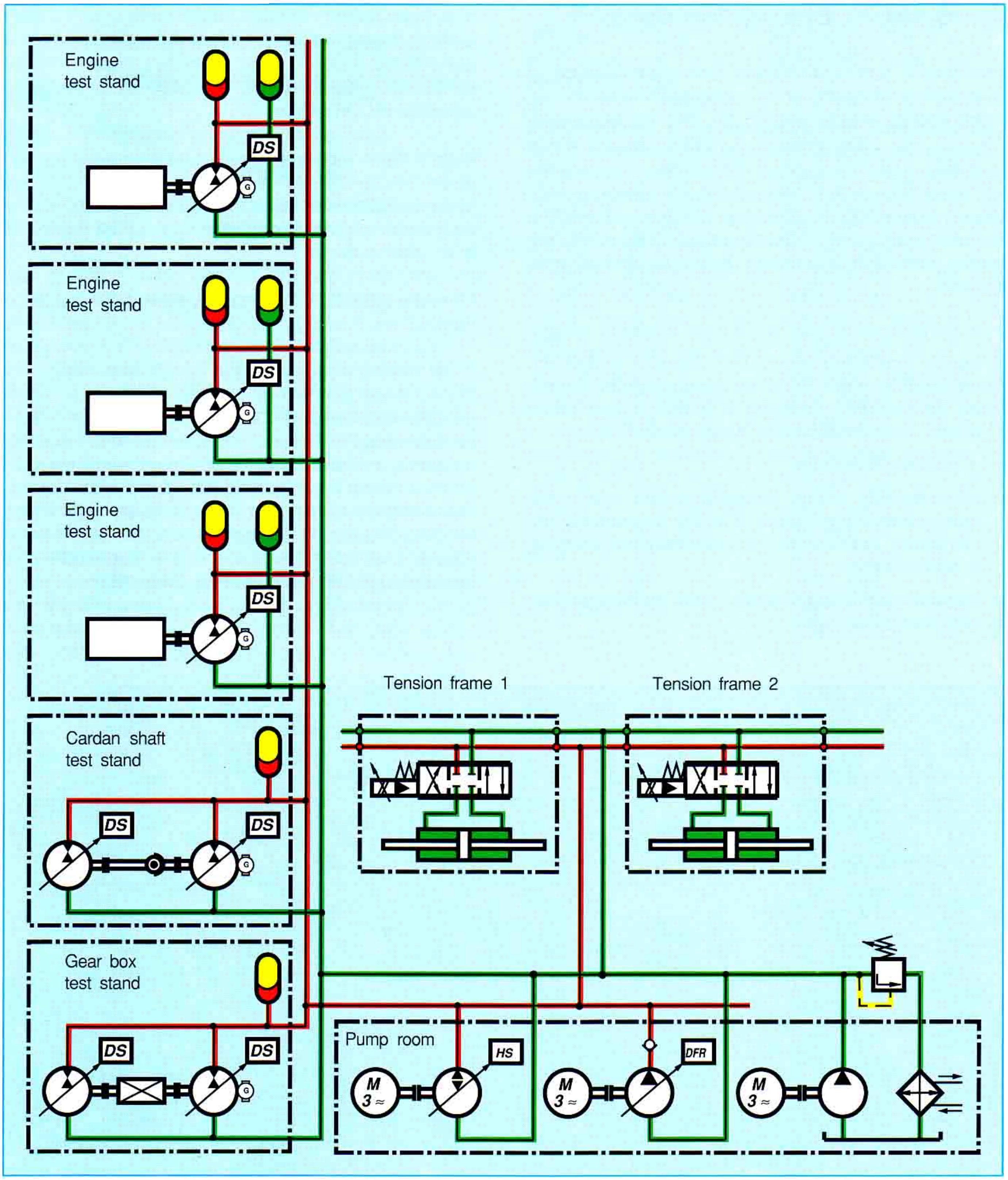



Fig.44: *Ring main system for a test bed installation with energy recovery.*

Such a system which has been carried out for a well known German automobile manufacturer is shown schematically in *Fig. 44.*

The starting point for this system was an existing test area consisting of two tensile testing units with hydraulic pulsing test stands, i.e. cylinders with servo valves in the energy feed lines. Power supply to this system was via a common pipeline system with two pressure compensated piston pumps designed for 280 bar.

The piping system was extended and in the first extension stage, engine test stands, drive shaft test stands and gearbox test stands were connected. All of these units operate under secondary control with imposed operating pressure. The braking energy of the engine test stands is now returned to the system and is available without throttling for use by the other actuators. As the engines can operated either on drive or on overrun, dynamic changes can be covered by hydraulic accumulators. These are mounted directly in the test cells.

The energy flow within the drive shaft and gearbox test stands is hydraulic/mechanical/hydraulic. The necessary acceleration power and also the losses of the units under test and the hydrostatic units are therefore all that need to be fed into these systems.

As all test stands work completely independently from each other the question of energy balance is of particular importance. The drive to the ring main consists of pressure compensated axial piston units mounted in a common pumping room. The axial piston units may be swivelled over centre and can act as either pumps or motors. In this way, any excess energy produced within the ring main causes the pumps to act as motors driving the AC motors and returning energy to the electrical power lines and also any power deficit is covered by the units acting as pumps. The number of electric motors and thus usage of primary energy is considerably reduced within this combined system. Energy recovery also reduces the amount of heat produced. By monitoring the swivel angle of the units the energy requirement of energy overflow of the system can be easily determined. This in turn means that a logic circuit can be fitted permitting the pump units to be switched in and out as required thus further reducing power losses.

Two test stands are described which are connected to this combined system.

### 5.8.1 Test Stand for Dynamic Rotary Group Testing

For the dynamic testing of a drive unit consisting of an internal combustion engine, a manual or automatic gearbox, and through to the drive wheels, the loading must simulate the vehicle mass and the resistance to movement. The energy taken from the loading units is then returned to the central hydraulic power lines.

As such test installations represent a considerable investment a great deal of flexibility is normally required with regard to the construction and adaption possibilities regarding the various items to be tested and testing programmes.

*Fig. 45* shows the test stand adapted to test a complete drive chain. The loading units simulate the travel conditions from the smallest unloaded vehicle right through to the largest loaded vehicle. During acceleration or deceleration of the vehicle the vehicle mass is also simulated in order to be able to make any required measurement on the stand without the need for re-building or re-setting.

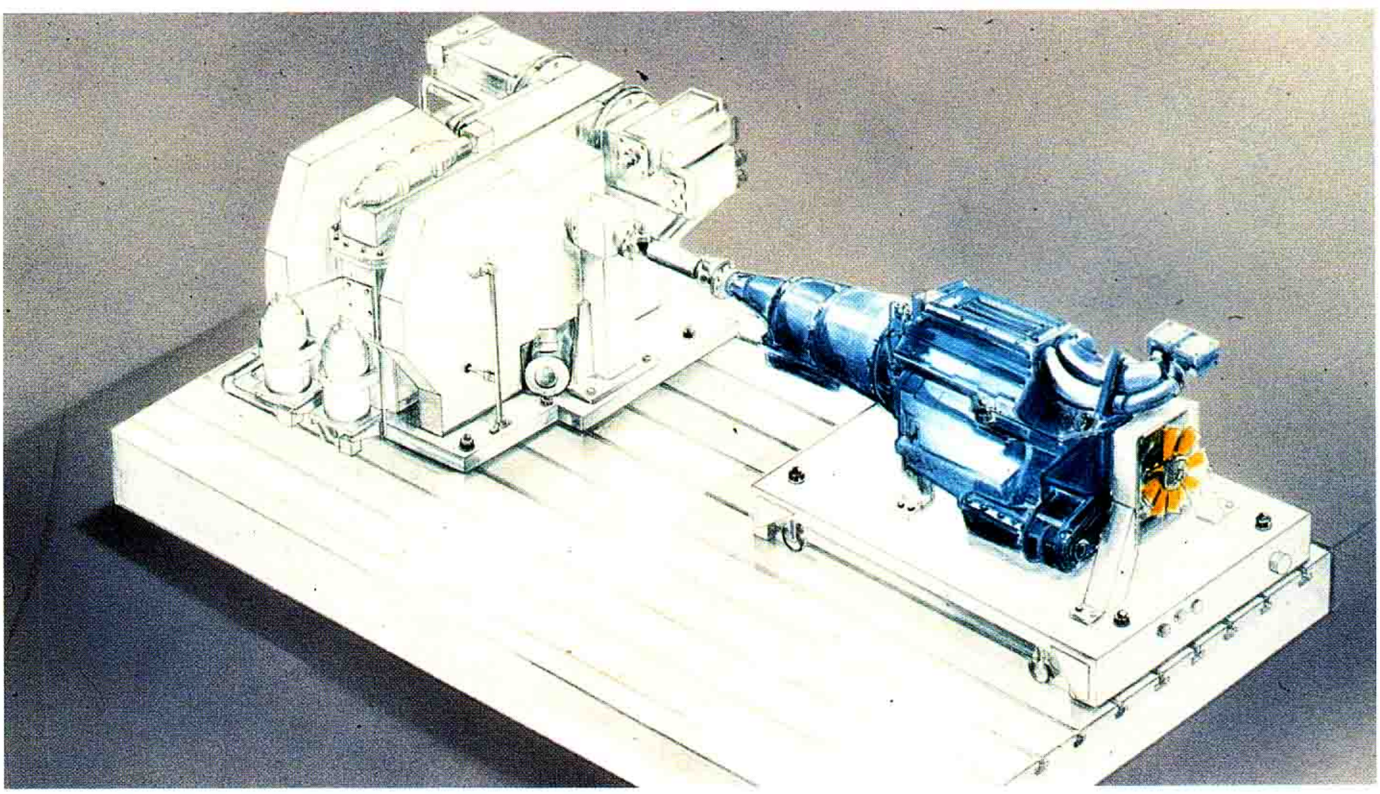


Fig. 45: *Test stand for drive components*

*Fig. 46* shows the arrangement of the rotating masses and the axial piston unit A4VSO... DS under secondary control.

For the simulation of the mass, the torques are inserted on the load side. These are calculated from the preset vehicle mass, the maximum travel speed and the momentary gradient of the drive speed.

The measured rotary acceleration is then used as an input value for the closed loop control. If the masses on the loading side are considerably too small or too large, severe changes in torque (e.g. gear changing) cause all loading systems to generate simulation errors and often also cause oscillation in the control. In order to eliminate these errors the system is equipped with a number of rotary inertia units which can be brought in when the unit is stopped. In this way, very good control accuracy is ensured and at the same time the cost of the braking machines and control technology is kept within bounds as only a small part of the 20-30% of the presently selected rotating mass needs to be added or subtracted in the simulation.

Between the engine, gearbox and brake unit is fitted a drive shaft. The speed and torque are measured immediately before the braking unit. A spur gear reduction is utilized to match the speed range required to that of the hydrostatic units. Dependent upon the power and speed range required, two or four hydrostatic machines are installed. In order to simulate resistance of the vehicle to travel, the gradient, rolling and air resistance values and vehicle frontage area, axle ratio and the dynamic rolling radius of the tyres are all preset. The air resistance is calculated dependent upon the vehicle speed. The gradient value can be varied as required during the run. All other values are taken as constant for the test. Hydraulic accumulators are necessary to achieve the required dynamic response in the system shown, as the system is connected to a central hydraulic power line which includes a time delayed reaction time. By means of careful matching of the transfer ratios, a suitable loading unit can be built up even for complete drive chains with two or four driven wheels.

The design data of the test stand *Fig. 46* is:

- Speed range (in both directions of rotation).
- 0 to 7000 rpm.
- Load torque (referred to $n_{max}$ = 7000 rpm) ± 2000Nm
- Vehicle weight (range) 700 to 4500 kg.
- Mass simulation at loading unit with a control constant in the range of ± 0,2 kgm².

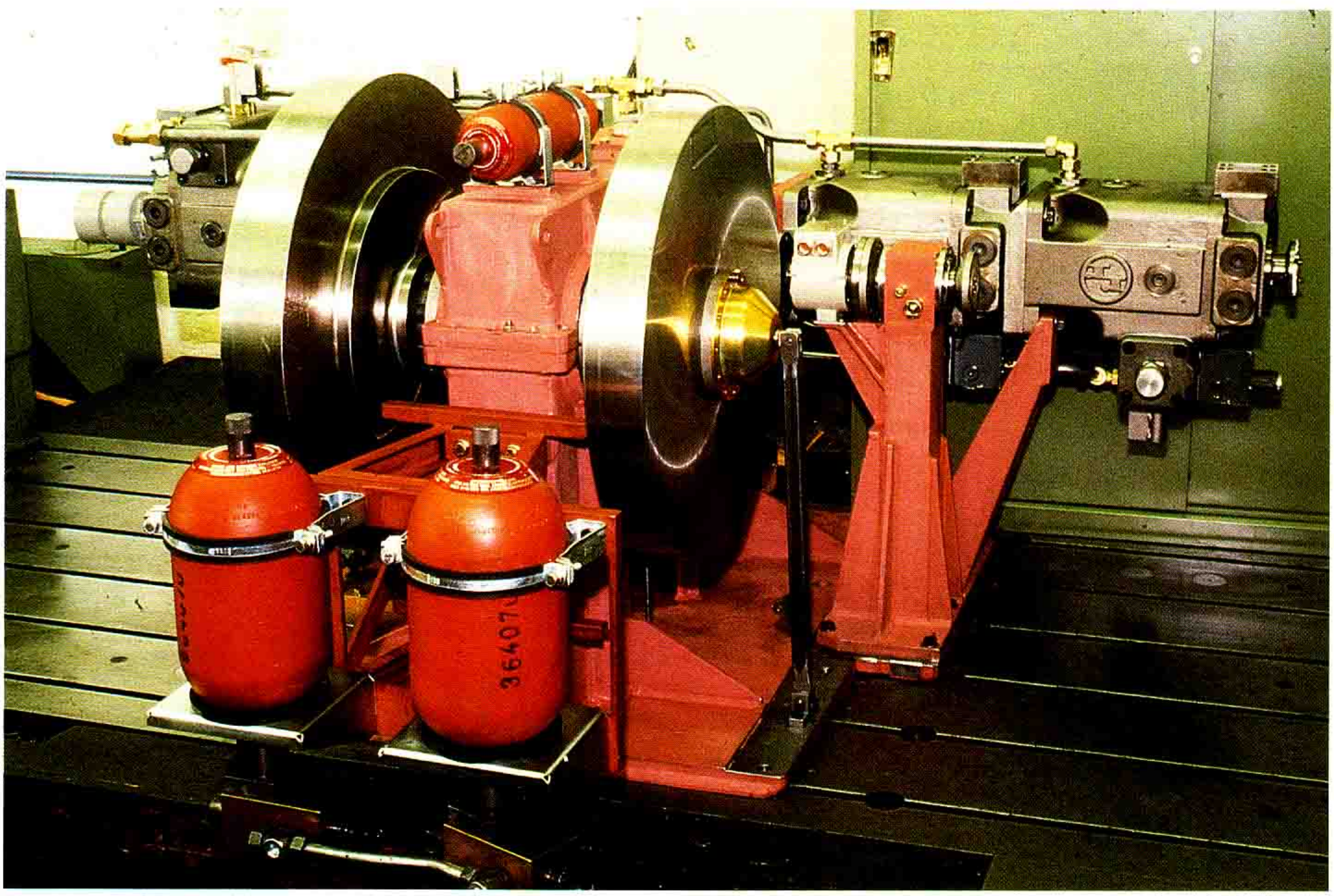


Fig. 46: *Loading device for test stand*

### 5.8.2 A Drive Shaft Test Stand

The drive shaft test stand shown in *figure 47* is built on the principle of back tensioning the unit. This means that the power delivered at the output end is fed back into the input end. In this way, only the power losses need to be taken from the power supply unit. The braking energy from the rotary group test stand *(5.8.1)* can be most sensibly used here without the need to convert this into any other energy form.

By means of gear coupling, four units can be coupled together and tested simultaneously. The intermediate bearing is set into a slide unit so that the vertical movement of the wheel can be simulated. The low rotating mass of this system is necessary in order to achieve the high torque and speed response required.

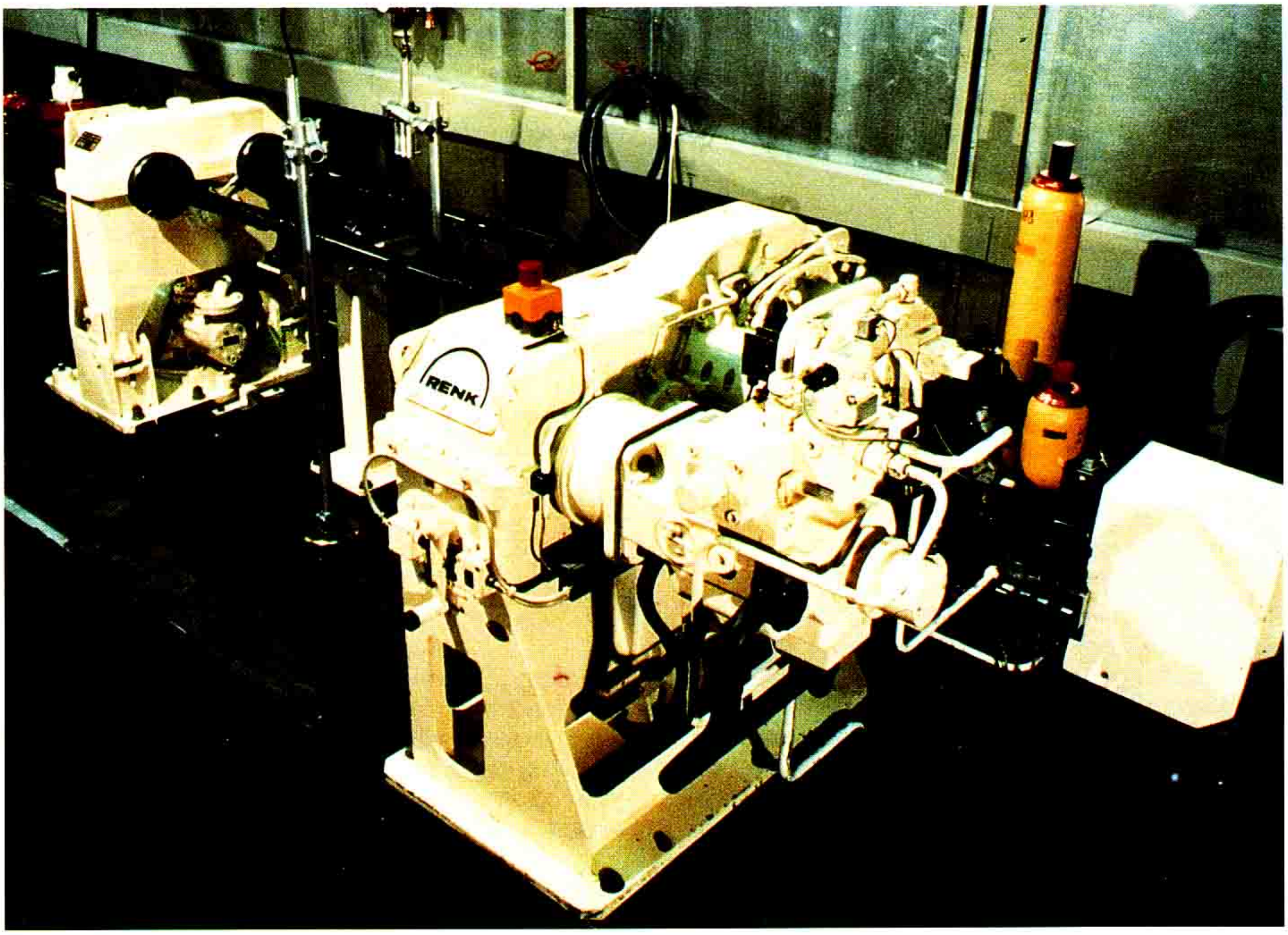



Fig. 47: *Test stand for drive shafts and cardan shafts.*

### 5.8.3 Test stand for axles and axle drives

When testing of multi-speed gearboxes with a wide gear ratios or units such as axles with a variety of gear ratios, it is often no longer economic to meet the demands of the mechanical loading side.

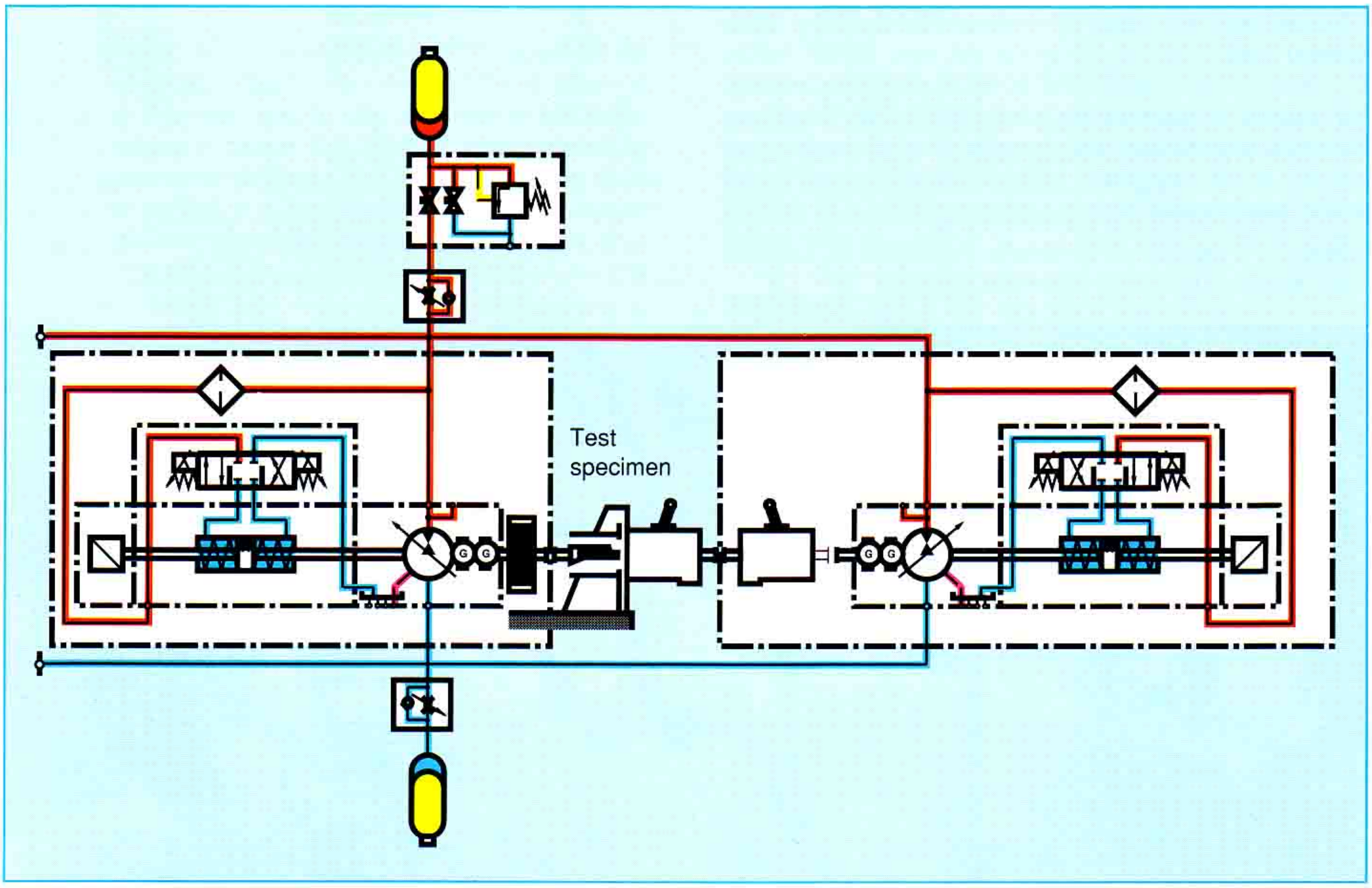



Fig. 48: *Hydraulic circuit for gearbox test stand*

Thus, in a test stand for a multi-speed gearbox as shown in *figure 48*, the braking unit requires to have as wide an operational range as the unit under test. This leads to unavoidable problems regarding the speed of the brake unit. In 1st. gear, the unit must produce maximum torque. Similarly, in top gear, a high speed is required, but both conditions do not arise at the same time. The problem can be solved in a very simple manner by mounting a second test gear box " back to back" with the test unit. This cancels out the gear ratio.

As the technical specifications of the two units are identical, the two hydrostatic units can have the same displacement. A further possibility when testing axles and gearboxes is to match the output speed range to that of the hydrostatic unit by means of a planetary gearbox. The planetary gearbox can be bridged by means of a gear type coupling when the unit is stationary. Thus a wide speed range is available for testing the vehicle gearbox.

*Figure 49* show an overall view of a test stand for built on this principle for testing axles and axle gear boxes. The torque and speed measuring shafts are built directly into the planetary gearboxes, so the additional inertia and space can be achieved. Speed imbalance problems are also avoided in this way.

As may be seen from the view on the drive side *(fig.50)*, the simulated internal combustion engine is extremely compact, has a high energy density and a low moment of inertia and a fast response. Two axial piston units type A4VSO250 DS are employed as a tandem unit with direct coupled tachometer are used on the drive side. The braking unit on the wheel end of the rig is of the same type, so that only 30 % of the power input need be accounted for as lost power for cooling.

All three units under secondary control in the drive and braking systems can be either speed or torque regulated. A further control function permits the speed difference between the two sides of the unit to be preset. In order to be able to cover the whole of the speed and torque rangewhen testing of manual and automatic gearboxes, and in addition to the two stage planetary gearboxes, the pressure on the high pressure side of the hydrostatic drive can be set as required.

The design data of the test stand is as follows:

- Four quadrant drive including stationary operation
- A maximum speed gradient of 3000 rpm/sec. (referred to the test unit, which means that the gear reduction in the planetary gear units must be taken into consideration.)
- The differential speed between the output drives must be variable up to 2000 rpm/sec.
- The torque of the two drives variable up to 50 000 Nm/sec.

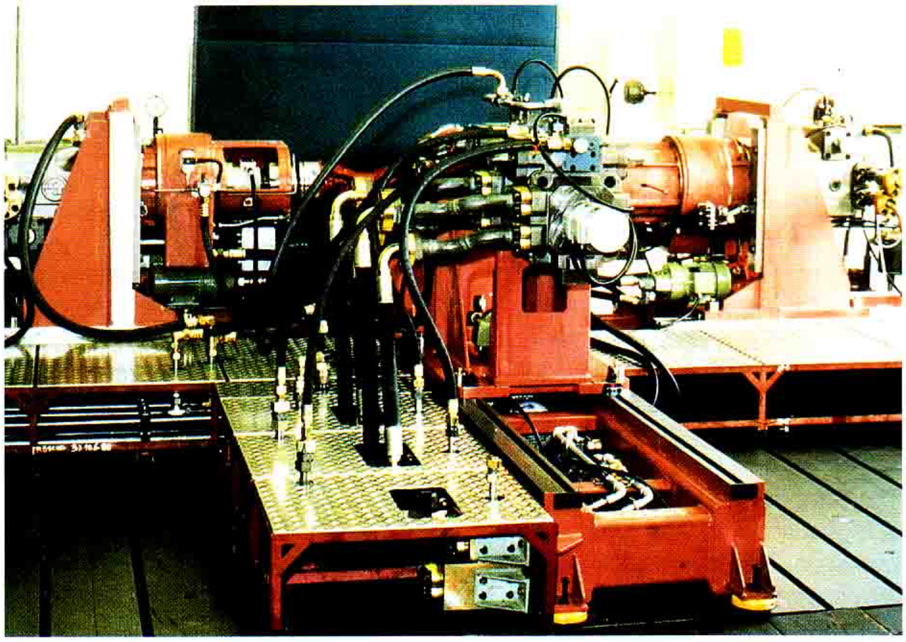


Fig. 49: *Overall view of the axle test stand*

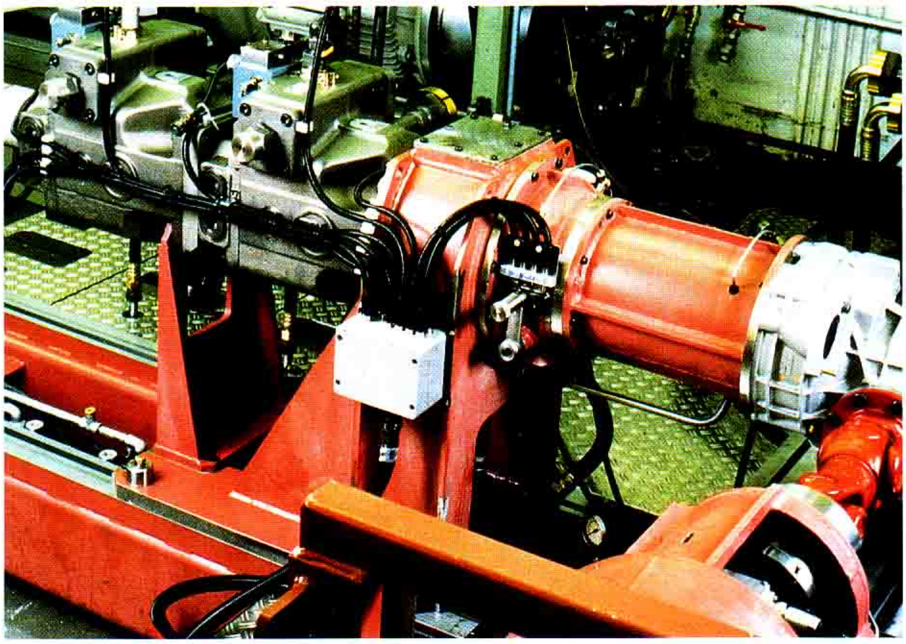


Fig. 50: *The drive side of the axle test stand*

### 5.8.4 Flat Bed Driving Simulator

Flat bed test units have been well known for some time in the automobile industry.

The installation shown in *figure 51* can be used as a tyre test rig. Such units are the basic elements for an axle test rig —with two such units— and a test stand for road simulation —four such units, onto which a complete vehicle, securely fixed about its centre on gravity, can be built. With centre-of-gravity mounting, the vehicle is fixed in position on the test bed such that its movement in the vertical, sideways and longitudinal direction is not affected.

The complete road testing of a car can then be carried out at 'speeds' up to 250 km/h with the vehicle still in the workshop. This has the advantage that the vehicle can be driven without regard to environmental and traffic conditions. In addition, the test conditions can be accurately reproduced and the need to mount instruments in the vehicle is overcome. Thus, the standard of driving comfort and the dynamic characteristics of the vehicle can both be tested under optimum conditions.

In order to fulfil this duty, the conveyors are equipped with contact-free bearings so that vertical movement can be introduced by means of cylinders. Due to the better damping of hydrostatic bearings as opposed to air bearings, the carrier bearings are manufactured on a water basis. In order to be able reproduce driving round curves, two of the conveyors must be able to be swivelled.

Due to the need to swivel the tracks, the choice of suitable drive motors was not solely dependent upon good dynamic characteristics and smooth running over the whole speed range. In addition, the motors must be light and compact. The choice was thus inevitably made for a system with secondary control. *Fig. 51* shows a standard A4VSO250DS unit. The power can be doubled without influencing the dynamic characteristics by the use of a tandem unit.

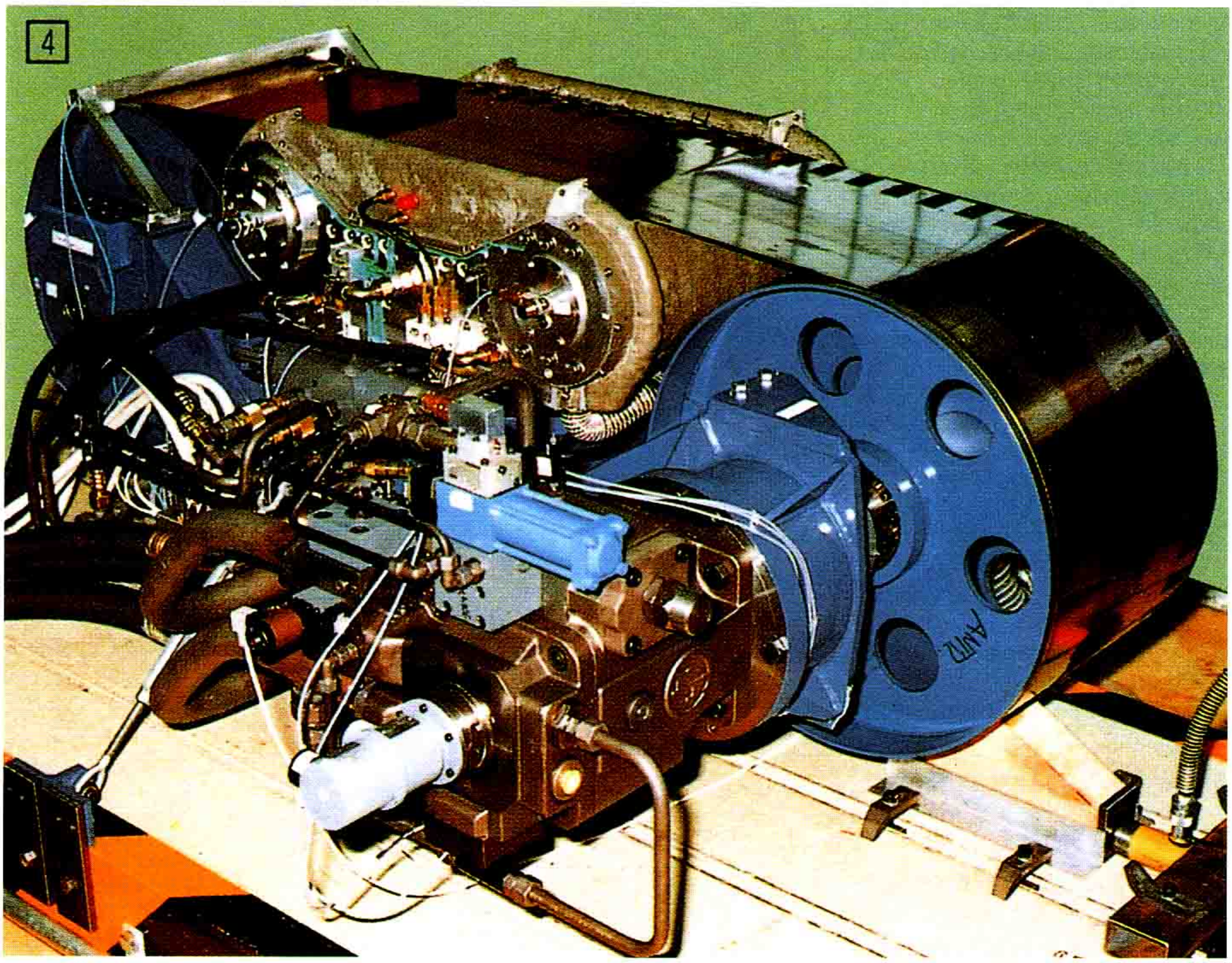



Fig. 51: *Flat bed test stand*

### 5.8.5 Fast Response Test Stand for Internal Combustion Engines

In developments in the automobile industry, the following tendencies are apparent:

- weight reduction
- the choice of models is being extended
- driving comfort is being improved
- driving noise is being reduced
- the time interval between the introduction of new models is being reduced
- reliability is being improved.

These requirements have had an unavoidable influence on the design of test rigs and on the requirements placed on simulation technology. As a general rule, the capabilities required exceed those of existing test stands.

An engine test stand will be described here the specification of which was laid down in 1983, and which was the first unit of its kind to demonstrate the possibilities of dynamic response under secondary control.

When the specification was laid down, it not only stated the technical parameters such as pressure, flow, temperature, velocity, acceleration and gas relationships must be measured and evaluated, but also that it must also be capable of dynamic endurance tests in which the loading and responses change automatically as would be found in a road test. The characteristic of this stand is the dynamic control of speed and torque and the measurement of these values.

As a rule, machines of this type in the automobile industry are equipped with semi-conductor controlled DC shunt wound motors. In this way, the energy produced by the internal combustion engine can be accepted by the DC machine and fed into the three phase power lines via suitable circuitry. This calls for extensive and expensive control circuitry. In many cases, due to the large rotating inertias, the dynamic control of speed and torque is inadequate.

The recently proposed introduction of field excited rectifier machines which have similar speed/torque characteristics to DC shunt wound machines are really of no help here, even when one takes into consideration that the rotating masses can be reduced.

Due to the high rates of dynamic response for speed and torque required in this instance, the company decided to use a system with secondary control.

The important points directing this decision were:

- that on acceleration and deceleration, an energy interchange with the hydraulic accumulator could take place without the need for dynamic feedback into the electrical power lines.
- the electric supply could be arranged for the average usage and not for the peak powers required.

The following requirements were placed upon the test stand:

- Max. power : $P = \pm 290$ kW
- Max. torque: $M = \pm 550$ Nm
- Speed range: $n =$ 600 to 7000 rpm.
- Dynamic response of speed: The loading machine must, when coupled to the test unit, be able to be driven under a triangular function from 1000 rpm to 7000 rpm and back again in **1 s** !

*Fig. 52* shows the test cell with the load unit consisting of a splitter box with two, speed controlled, axial piston units in parallel.

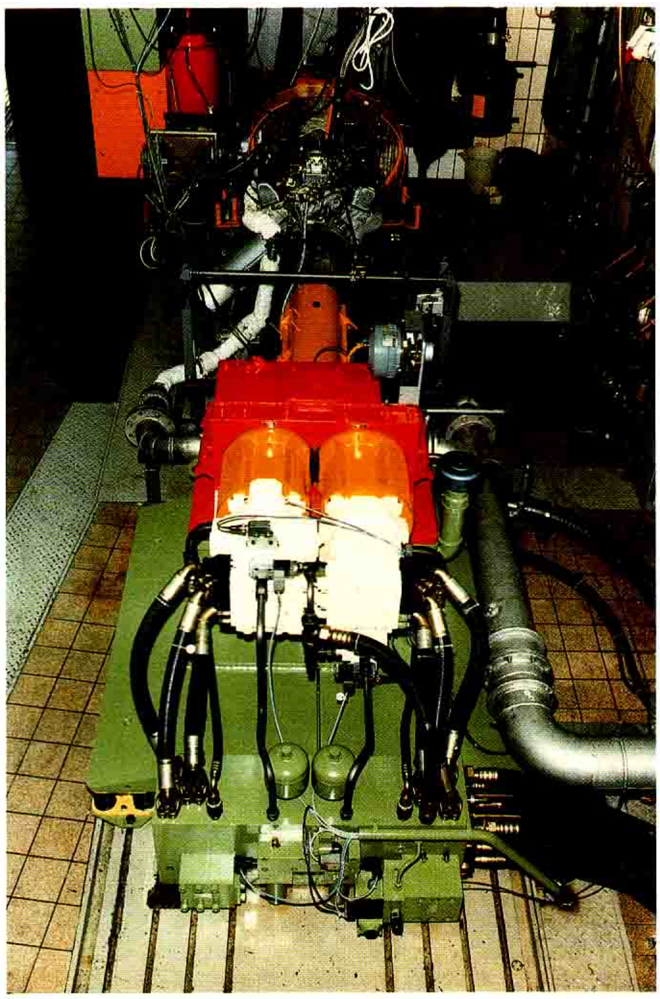


Fig. 52: *Overall view of the loading machine with test specimen*

The engine under test is coupled to the single input shaft of the splitter box via a torque measuring unit.

Two pressure compensated axial piston units of the same size as the loading machines are connected as shown in *figure 54* and form the coupling between the hydraulic circuit and the three phase power lines.

As the power units are some distance from the test cell and are also on a lower floor, a closed circuit hydraulic system had to be chosen.

The torque setting is achieved by setting a specific throttle opening for the engine. The loading machines react accordingly and the engine under test either drives or is driven.

A switching safety circuit ensures that the power feed to the loading machine is broken should a fault occur. Under this protocol, the speed is reduced to zero in 1,2 s. By switching off one of the axial piston units or by lowering the operating pressure, the power of the test stand can be varied to suit the power of the engine under test.

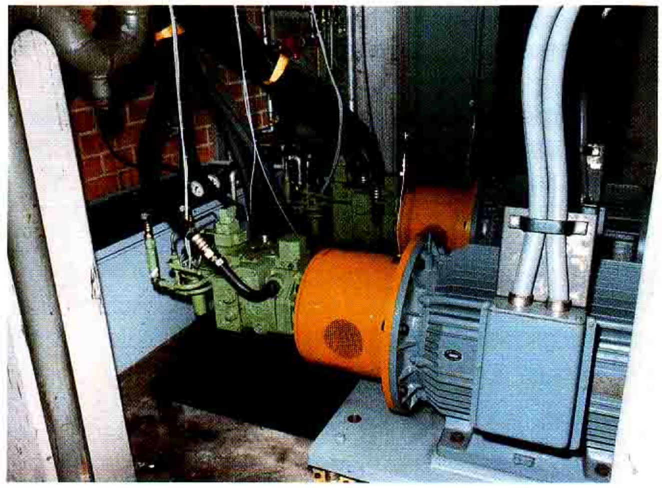
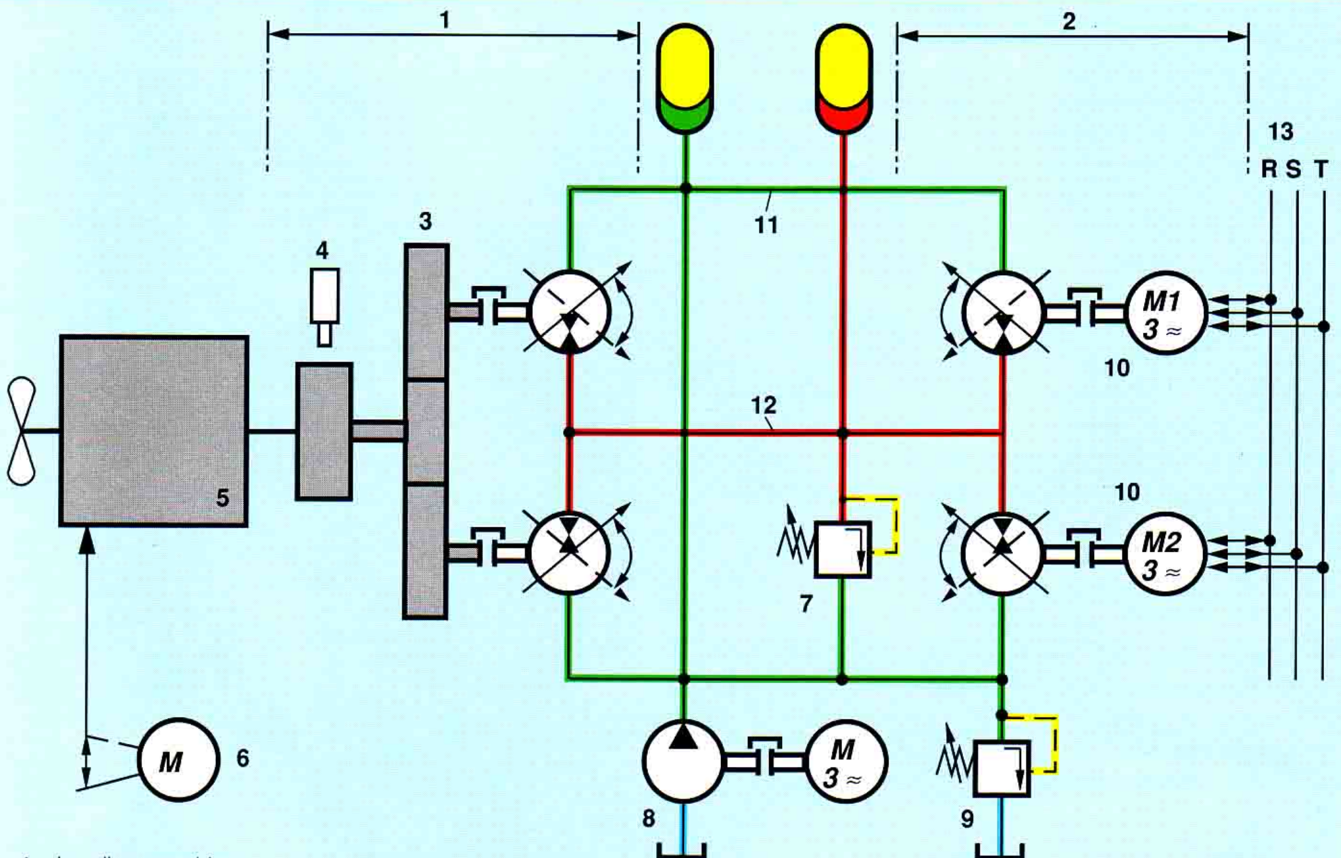


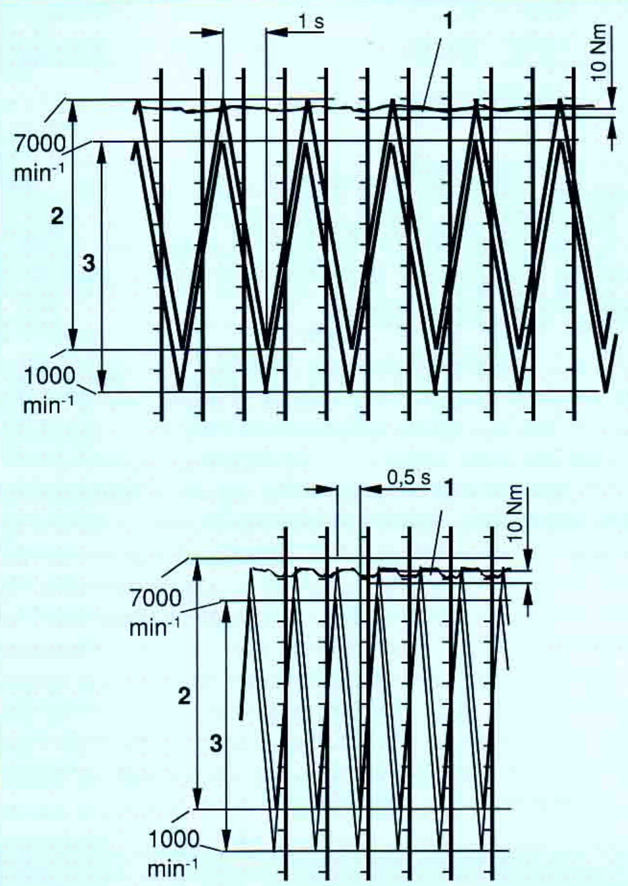
Fig. 53: *AC motor with pressure compensated axial piston unit*

A sketch of the hydraulic circuit is shown in *figure 54*. Due to the imposed pressure, the physical arrangement of the loading machine and the primary unit may be freely chosen as the distance between the primary and secondary units plays no part in the dynamic response of the system.




1 Loading machine
2 Power supply unit
3 Intermediate drive
4 Speed sensor
5 Engine under test
6 Throttle control device
7 Safety valve
8 Boost pump
9 Boost pressure valve
10 Asynchronous machine, P= 160 kW
11 Low pressure
12 High pressure
13 Electrical power lines

Fig. 54: *Sketch showing principle of the circuit*




**1** Acceleration torque of the measuring wheel
**2** Command speed
**3** Actual speed

Diagram 8: *Measured speed curves.*

The energy required to accelerate the loading machine itself is taken from a hydraulic accumulator which restores the energy during the deceleration phase. This in turn means that the measures which would be required to stabilize the electrical power lines are no longer required. Such circuitry is required if asynchronous machines are to return a pure sinusoidal wave form back into the electrical power lines. The efficiency of energy recovery has been measured at 75%.

In order to prove the dynamics of the test stand, the operator of the test stand stated that a speed response curve had to be produced. This curve is shown in *diagram 8.*

This diagram shows the actual and command values for speed against time. and is measured on the high speed shaft of the loading device. The engine under test which could have scarcely endured this procedure for very long was uncoupled during this test.

As can be easily read from the curves, the speed changed from 900 rpm to 7200 rpm in 500 ms. This corresponds to a rate of change of speed of 12 600 rpm per second.

The time offset between the command and actual values arises from the fact that the closed loop speed control was not optimized for this type of operation.

Had the test engine been connected, a curve as shown in *Diagram 9* would have been achieved. The internal combustion engine rotates at 5700 rpm with the throttle full open. The torque at this time is 310 Nm. The engine is then pulled down to 3500 rpm by the loading machine in 0,9 s. This causes the torque to rise to 400 Nm. The loading machine must therefore decelerate both the engine and itself in order to cause the higher torque to occur. It fulfills this duty completely.

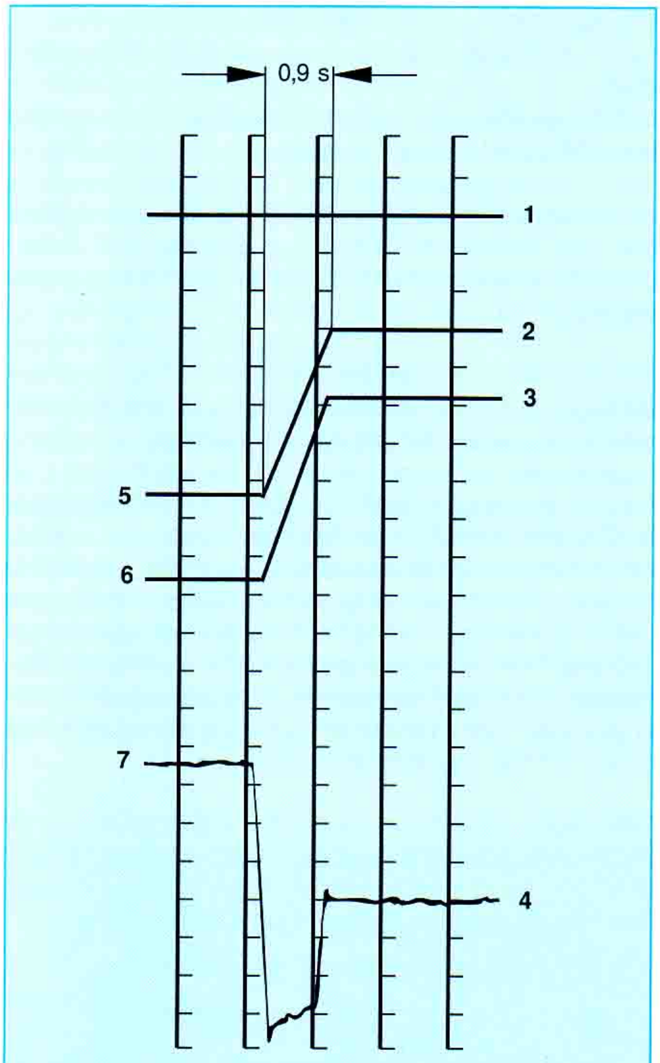
The preset values for torque for the engine control or for the speed control of the machine can be entered at will.

The signals given are derived in one of three ways:

- from a simulation computer given vehicle model
- from actual values obtained during test drives
- from a synthetic programme laid down by the research engineer.

For these to be effective, the closed loop control must fulfil the following duties:

- Speed control of the engine under test and of the loading machine
- Position control of the throttle and thus the torque control of the engine
- The electronic pressure control of the axial piston units and of the primary units via pressure transducers.




1 Throttle open
2 Command speed 3500 rpm
3 Actual speed 3350 rpm
4 Torque
5 Actual speed 5700 rpm
6 Command speed 5700 rpm
7 Torque 310 Nm

Diagram 9: *Speed and torque curves with engine connected.*

The axial piston units which had been running since 1983 and were checked after a running time of 7500 hours. A visual check showed no apparent wear. This was only to be expected, as the functioning of the test rig had not shown any degradation.

On first commissioning, the decision had been made that the main components should receive a basic overhaul after 10 000 hours. This corresponds to approximately 5 years operating time.

## 6 Low loss controls in a hydraulic ring main system with imposed pressure

Low loss power take off from a ring main system with imposed pressure is best achieved with a rotary drive in which the output device can be "swivelled over centre". Utilising such devices, a four quadrant drive is possible in open circuit configuration.

However, fixed displacement units can also be connected to such systems. For this, it is necessary to make certain changes to the circuit as can be seen in *figure 55.* A fixed displacement unit (1) is coupled to a tacho-unit (2) in a conventional manner. The speed information is achieved by flow control (3) and the directional control by means of directional valve (4). Between the pressure line and the main actuator, a 4 way proportional valve (5) (which is piloted via the tacho-unit circuit) is installed. The proportional valve now controls the pressure difference at the secondary unit dependent upon the external torque applied to the unit, so that the pre-set speed set by the flow control can be maintained. The pressure difference between the system pressure and the actuator pressure is generated at the proportional valve.

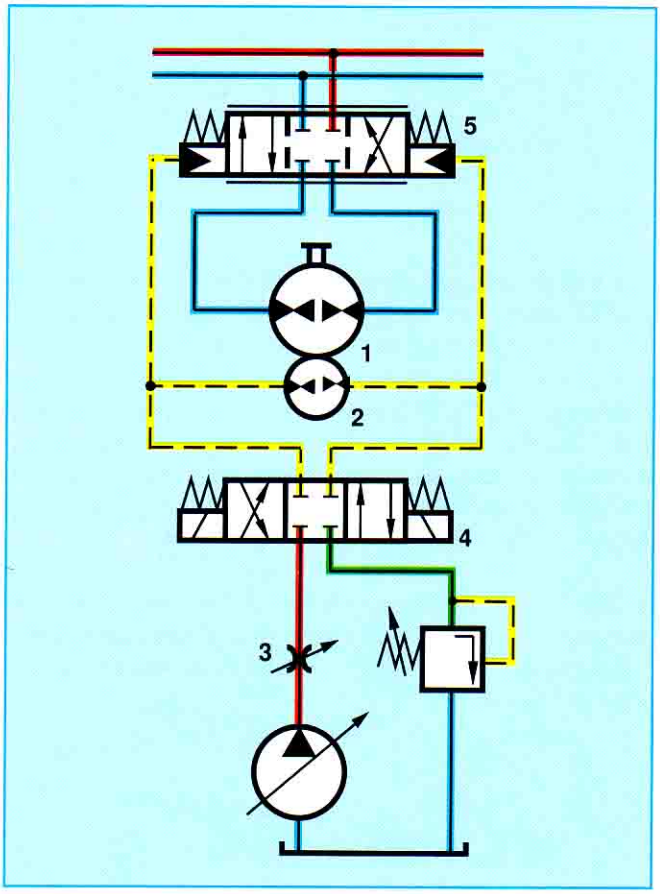



Fig. 55:
*Fixed displacement unit in a system with impressed pressure*