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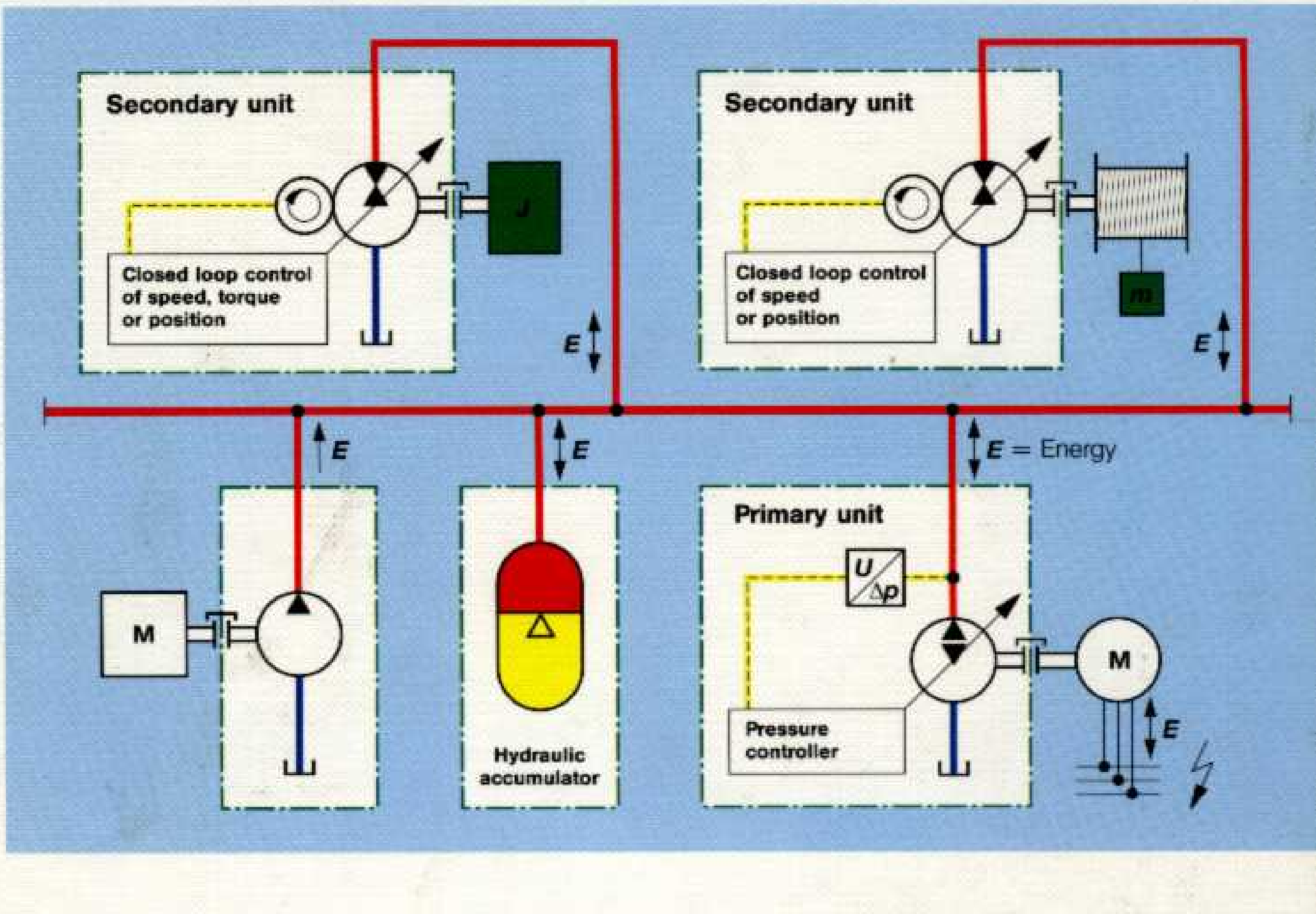
Hydrostatic drives with control of the secondary unit

The Hydraulic Trainer, Volume 6



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Hydrostatic Drives with
Secondary Control

An introduction into the drive concept
and system characteristics

Second edition, completely revised
with 240 illustrations and 7 tables

Edited by Rudi A. Lang

From
"The Hydraulic Trainer" series

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Preface to the Second Edition

It is hard to imagine any transport system or production process today without hydrostatic drives as hydro-mechanical energy converters. The ease of control in open and closed loop circuits makes it possible to meet the increasing demands for flexibility and precision of drives brought about by technical rationalisation.

Secondary control has become increasingly important due to the fact that its characteristics are much more comparable to electrical drives with closed loop control than to conventional hydraulic drives. It is therefore no wonder that secondary control and its associated closed loop control technology has gained considerable impetus through the demands of electrical drives. This has led to the development of a new kind of closed loop control technology. Secondary control fitted the bill, with its already familiar structure and excellent dynamic response characteristics.

Secondary control is another variant of hydrostatic energy transmission.

With conventional hydrostatic energy transmission the secondary unit is moved by open or closed loop control of the flow, either throttle-free via a primary unit with variable displacement, or throttled by means of proportional or servo valves or similar in the energy lines.

With secondary control on the other hand, the torque is generated by a closed loop control process directly at the secondary unit. As with controlled electrical systems the required flow is taken directly from a hydraulic pressure circuit with impressed pressure. The primary side therefore merely has the task of preventing a pressure drop, as is also the case with electrical systems. This type of energy transmission permits unlimited use of hydraulic accumulators to cover short-term peak flow requirements of the actuators or to recover and store potential or kinetic energy from the actuators.

Secondary control, however, has yet to be completely accepted by hydrostatics experts. It cannot completely replace conventional hydrostatic drives. Rather, it should be considered as making those fields of hydrostatics accessible, which were previously inaccessible with respect to dynamic response and accuracy in speed, torque and positioning control, as well as rendering possible energy recovery with or without conversion into another energy form.

This volume "Hydrostatic Drives with Secondary Control", as part of "The Hydraulic Trainer" series, aims to communicate to the reader the fundamental principles of secondary control.

The axial piston units, with slight modifications for secondary control, are introduced and descriptions are given of the electronic components. One chapter covers dynamic response and the particular characteristics of secondary control referred to conventional drive technology.

Selected examples demonstrate the diversity of applications. This should enable the technician to select and design the most suitable hydrostatic drive for the specific application.

The final chapter of this book contains information on project design, which should prove helpful.

Our thanks are extended to all those who have helped in the production of this book by offering information, suggestions and creative ideas.

I would especially like to thank Mr. Rudi Lang for the extensive contributions he has made.

Rolf Kordak, Lohr a. Main

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Terms and Definitions

Impressed pressure
This is a term borrowed from electrotechnology. With power converters in DC intermediate circuits we talk of impressed voltage or impressed current.

With hydrostatic drives using closed loop pressure-controlled hydraulic pumps and hydraulic accumulators the system pressure is dependent on the loading condition of the accumulators, and is therefore not constant but "impressed".

Impressed flow
This is the flow directed to an actuator via variable displacement pumps (displacement control) or valves (valve control) for the generation of a specific output.

Energy utilisation ratio ETA
The term energy utilisation ratio ETA is used when it is unrealistic to refer to the term efficiency, for example with values > 100%.

ETA is the quotient of the mechanical energy output to the machine to that produced by the prime mover.

GTO thyristor
A Gate Turn-Off Thyristor is a power semi-conductor that can be switched off with average switching frequency and blocking voltage and high specific on-state voltage. It is used in frequency converters.

Displacement
The recommended terms "Volumetric displacement" for hydraulic pumps and "Absorption capacity" for hydraulic motors are replaced here by the term "Displacement", as a secondary controlled unit operates in four quadrant mode and it is not possible to completely separate pump and motor modes.

Hydraulic spring
This is the column of oil in a pipe system, between the primary and secondary sides, that is under a varying operating pressure.

Hydraulic isolator
This is the connection point of the secondary unit to the hydraulic system, and is analogous to the electrical plug. It is normally in the form of an electrically piloted check valve.

The hydraulic isolator stops the flow of hydraulic energy to the actuator.

Hydraulic transformer
A hydraulic transformer is a hydrostatic transmission comprising a secondary controlled variable displacement unit and a fixed or variable displacement hydrostatic unit.

The hydrostatic units are permanently coupled together with no mechanical input or output drive.

With the hydraulic transformer single-acting and synchronous cylinders can be operated on the same ring main with impressed operating pressure without throttling losses, and under suitable conditions energy may be fed back into the system.

IGBT
This term stands for an Insulated Gate Bipolar Transistor.
In drive technology transistors are used at high pulse frequencies as contactless electronic switches.

Mooring operation
This is dual-quadrant operation with motor and generator functions in one direction.

MOS
Metal Oxide Semiconductor defines the sequence of layers, (M (gate electrode), O (insulating layer), S (semiconductor channel), set into integrated circuits as field effect transistors.

MOS is the general term for the technology used in the manufacture of discreet and integrated semiconductor components and any associated operational components.

Runge-Kutta-Merson integration method
This is a numerical method for solving differential equations.

It operates with automatic stepped control to a predetermined accuracy limit, taking into consideration any discontinuities that may arise.

The Runge-Kutta method and derived variants of these are of great practical significance.

Self-blocking
Self-blocking is the uncontrolled deceleration of a hydraulic motor to a standstill, caused by the swivel angle dropping to within the zero range. This definition cannot be applied to secondary controlled units.

Stribeck curve
The Stribeck curve represents friction plotted against speed or number of revolutions.

Tandem unit

A tandem unit is the term for two hydrostatic units coupled together.

Translatory analogon

This term is used to describe the inclusion of hydraulic cylinders into the secondary drive unit with energy recovery.

Total loss

This is the sum of all leakages from within a hydraulic system. It is made up of:

- Pressure losses due to mechanical or hydraulic friction and flow losses,
- Leakage losses and
- Throttling losses, which usually account for the greatest loss.

The total loss will always be converted into heat at the position where it occurs.

Four quadrant operation

This is the term given to the operating mode with bi-directional rotation and bi-directional torque.

Efficiency

This is the ratio of output to input of a device, unit or system.

Time factor of the controlled system

This is the factor for the time response of the controlled system for testing system response to the secondary control.

The time factor of the controlled system t_T in seconds characterises both dynamic response of the controlled system and secondary unit.

Formulae

Symbol	Meaning	Statutory unit; practical unit
A_K	Area, piston area	m^2; cm^2
$A_{1,2}$	Amplitude of a control value	
B	Magnetic induction	T
C	Capacity	F
$C_{1/2}$	Spring constant of a column of oil	$\frac{N}{m}$
C_G	Loss factor with generator operation	
C_V	Loss factor with motor operation	
c	Wave speed	m/s
c_S	Spring constant	$\frac{N}{m}$; $\frac{N}{mm}$
D	Diameter	m
D	Electrical flow density	$\frac{C}{m^2}$
d	Diameter	m
d	Damping	
d_H	Hydraulic diameter	m; mm
dB	Amplitude ratio	decibel
E	Elasticity modulus	$\frac{N}{m^2}$; $\frac{N}{cm^2}$; $\frac{N}{mm^2}$
E	Energy, general	J; Nm
E	Electrical field strength	$\frac{V}{m}$
E	„Original voltage"	V
E_{DC}	D.C. voltage	V
E_{hydr}	Hydraulic energy	Nm
E_{kin}	Kinetic energy	Nm
E_{oil}	Elasticity modulus of oil	$\frac{N}{m^2}$; $\frac{kg}{cm \cdot s^2}$
e	Eccentricity	m; cm; mm
e	Unit charge	C

Symbol	Meaning	Statutory unit; practical unit
F	Force	N
T	Amplitude response	—
F	Amplitude ratio	—
f	Frequency	Hz; $\frac{1}{s}$
f_0	Fixed mains frequency	Hz; $\frac{1}{s}$
f_1	Stator frequency	Hz; $\frac{1}{s}$
f_e	Natural frequency	Hz; $\frac{1}{s}$
G	Weight	N
G	Shear modulus	$\frac{N}{m^2}; \frac{N}{cm^2}; \frac{N}{mm^2}$
g	Acceleration due to gravity	$\frac{m}{s^2}$
H	Magnetic field strength	$\frac{A}{m}$
h	Height	m
I	Current, general	A
I_A	Armature current	A
I_B	Base current	A
I_C	Collector current	A
I_E	Emitter current	A
I_H	Holding current	A
I_{pilot}	Pilot current	A
J	Moment of inertia	kgm^2
J_1	Mass moment of inertia	kgm^2
J_{rm}	Mass moment of inertia of secondary unit	kgm^2
J_{red}	Reflected mass moment of inertia	kgm^2
J_{total}	Total mass moment of inertia	kgm^2
J_{add}	Additional moment of inertia	kgm^2
K	Compression modulus	$\frac{m^2}{N}; \frac{1}{bar}$

Symbol	Meaning	Statutory unit; practical unit
K_a	Integration constant	$\frac{1}{kgm^2}$
K_{Rei}	Multiplication factor in control circuit	
L_p	Sound pressure level	dB
l	Length	m
M_a	Idling torque	Nm
M_{acc}	Acceleration torque	Nm
M_E	Electrical torque in air gap	Nm
M_l	Load torque	Nm
M_f	Frictional torque	Nm
M_{comm}	Torque command value, general	Nm
M_T	Torque, general	Nm
M_{T1}	Drive torque; torque of primary unit; torque of hydraulic pump	Nm
M_{T2}	Output torque; torque of secondary unit; torque of hydraulic motor	Nm
$M_{T2,max}$	Maximum torque of secondary unit	Nm
$M_{T,comm}$	Torque command value, general	Nm
$M_{T\,max}$	Maximum torque, general	Nm
M_v	Speed-dependent torque correction value	Nm
M_{add}	Additional loss torque	Nm
m	Mass	kg
n	Speed	rpm/s
$\dot{n}$	Change of speed per unit time	s^{-2}; $\frac{min^{-1}}{s}$
$\dot{n}_2$	Change of speed per unit time of secondary unit; acceleration of secondary unit	s^{-2}; $\frac{min^{-1}}{s}$
$\dot{n}_{max}$	Maximum change of speed per unit time	s^{-2}; $\frac{min^{-1}}{s}$
Δn	Speed deviation of secondary unit; Speed change of secondary unit	rpm/s
n_1	Drive speed of primary unit; Drive speed of hydraulic pump	rpm/s
n_2	Speed of secondary unit; Output speed of hydraulic motor	rpm/s
$n_{2,act}$	Speed actual value of secondary unit; Speed actual value of hydraulic motor	rpm/s

Symbol	Meaning	Statutory unit; practical unit
$n_{2\,act\,M}$	Speed actual value of secondary unit (master)	rpm/s
$n_{2\,act\,S}$	Speed actual value of secondary unit (slave)	rpm/s
$n_{2\,max\,extern}$	Maximum speed of secondary unit, externally set	rpm/s
$n_{2\,max\,intern}$	Maximum speed of secondary unit, internally set	rpm/s
n_{act}	Speed actual value, general	rpm/s
n_{max}	Maximum speed	rpm/s
n_{min}	Minimum speed	rpm/s
n_{mean}	Mean speed	rpm/s
n_N	Nominal speed	rpm/s
n_{comm}	Command speed, general	rpm/s
P	Power, general	$\frac{Nm}{s}$; W; kW
P	Effective power	W
P_1	Maximum power of primary unit	$\frac{Nm}{s}$; W; kW
P_{corner}	Corner power	$\frac{Nm}{s}$; W; kW
P_{max}	Maximum power	$\frac{Nm}{s}$; W; kW
P_{hyd}	Hydraulic power	$\frac{Nm}{s}$; W; kW
P_{max}	Maximum power	$\frac{Nm}{s}$; W; kW
P_b	Blind power	W
P_s	Apparent power factor	W
P_{add}	Additional power	$\frac{Nm}{s}$; W; kW
p	Pressure	$\frac{N}{m^2}$; bar
p	No. of pairs of poles	—
p_{HP}	High pressure	$\frac{N}{m^2}$; bar
p_{act}	Pressure actual value	$\frac{N}{m^2}$; bar
p_{LP}	Low pressure	$\frac{N}{m^2}$; bar

Symbol	Meaning	Statutory unit; practical unit
Δp	Pressure differential	$\frac{N}{m^2}$; bar
Δp_{act}	Pressure differential actual value	$\frac{N}{m^2}$; bar
Δp_{pilot}	Pressure differential in pilot circuit	$\frac{N}{m^2}$; bar
p_{boost}	Boost pressure	$\frac{N}{m^2}$; bar
Q	Electrical loading	C
Q	Flow, general	$\frac{m^3}{s}$; $\frac{dm^3}{min}$; $\frac{L}{min}$
Q_1	Flow of primary unit; Flow of hydraulic pump	$\frac{m^3}{s}$; $\frac{dm^3}{min}$; $\frac{L}{min}$
Q_2	Flow of secondary unit; Flow of hydraulic motor	$\frac{m^3}{s}$; $\frac{dm^3}{min}$; $\frac{L}{min}$
Q_{max}	Maximum flow	$\frac{m^3}{s}$; $\frac{dm^3}{min}$; $\frac{L}{min}$
Q_{min}	Minimum flow	$\frac{m^3}{s}$; $\frac{dm^3}{min}$; $\frac{L}{min}$
Q_{mean}	Mean flow	$\frac{m^3}{s}$; $\frac{dm^3}{min}$; $\frac{L}{min}$
Q_P	Flow of hydraulic pump	$\frac{m^3}{s}$; $\frac{dm^3}{min}$; $\frac{L}{min}$
R	Resistance	Ω
R_1	Stator resistance	Ω
R_2	Rotor resistance	Ω
r	Radius	m
r	Jerk	$\frac{m}{s^3}$
r_B	Geometrical radius of drive flange	m; cm
r_Z	Geometrical radius of cylinder drum	m; cm
S	Surface	m^2; cm^2
s	Slip	
T	Temperature	K; °C

Symbol	Meaning	Statutory unit; practical unit
T	Length of time	s
t_1	Ramp time of load moment	s; ms
t_r	Time factor of control area	s
t_{swivel}	Swivel time; control time	s; ms
t_{delay}	Time delay	s; ms
t	Time	s
t_a	Control time	s
t_r	Setting time	s
U	Range, circumference	m
U	Voltage, general	V
U_1	Stator voltage	V
U_2	Rotor voltage	V
U_{CE}	Collector-emitter voltage	V
U_E	Analogue input voltage	V
U_a	Correction factor	V
U_{di}	Mean value of voltage in idle running	V
U_i	Analogue input voltage, current dependent	V
U_{act}	Voltage actual value	V
U_n	Analogue input voltage, speed dependent	V
U_{comm}	Voltage command value	V
U_{pilot}	Pilot voltage	V
$V_{\text{öl,m}}$	Volume of oil column	m^3; cm^3
V_g	Displacement, general	m^3; cm^3
V_{g1}	Displacement of primary unit; Displacement of hydraulic pump	m^3; cm^3
V_{g2}	Displacement of secondary unit; Displacement of hydraulic motor	m^3; cm^3
$V_{g2,max}$	Maximum displacement of secondary unit; maximum displacement of hydraulic motor	m^3; cm^3
v	Speed, general	$\frac{m}{s}$
v_{max}	Maximum speed, general	$\frac{m}{s}$
v_{pos}	Positioning cylinder speed	$\frac{m}{s}$; $\frac{mm}{s}$
W	Work	J; Nm

Symbol	Meaning	Statutory unit; practical unit
X	Torque relationship	—
x_a	Output value	
$\hat{x}_a$	Amplitude of output value	
x_e	Input value	
$\hat{x}_e$	Amplitude of input value	
y	Positioning cylinder distance	mm
Z	No. of pistons	
Z	Impedance	Ω
Z	Disturbance torque	Nm
α	Swivel angle, general	rad; degrees; °
α	Control angle	rad; degrees; °
α	Firing delay angle	rad; degrees; °
$\dot{\alpha}$	Change per unit time of swivel angle	$\frac{rad}{s}$; $\frac{°}{s}$
α_1	Swivel angle of primary unit; Swivel angle of hydraulic pump	rad; degrees; °
α_2	Swivel angle of secondary unit; Swivel angle of hydraulic motor	rad; degrees; °
$\alpha_{2,max}$	Maximum swivel angle of secondary unit; maximum swivel angle of hydraulic motor	rad; degrees; °
α_{act}	Swivel angle actual value, general	rad; degrees; °
$\alpha_{act,load}$	Swivel angle actual value of load unit	rad; degrees; °
$\alpha_{act,sec}$	Swivel angle actual value of secondary unit	degrees, °
α_{corr}	Swivel angle correction factor	rad; degrees; °
α_{max}	Maximum swivel angle	rad; degrees; °
α_{comm}	Swivel angle command value	rad; degrees; °
α_{per}	Maximum permissible swivel angle	degrees, °
δ	Degree of uniformity	%
ε	Relative permittivity	$\frac{F}{m}$
η	Efficiency, general	%
η_{total}	Total efficiency	%
η_{mh}	Mechanical-hydraulic efficiency	%
η_{vol}	Volumetric efficiency	%
κ *	Adiabatic exponent	%
λ	Tube resistance coefficient	—

Symbol	Meaning	Statutory unit; practical unit
λ	Wavelength, electrical	m
μ	Permeability	$\frac{H}{m}$
Φ	Magnetic flow	$V \cdot s$
Φ	Stator flow	$V \cdot s$
φ	Angular position	rad, °
φ	Phase shift	rad, °
φ	Phase response	
$\dot{\varphi}$	Angular velocity	s^{-1}
$\ddot{\varphi}$	Angular acceleration	s^{-2}
φ_2	Angular position of secondary unit	rad; degrees; °
$\dot{\varphi}_2$	Angular velocity of secondary unit	s^{-1}
ω	Angular velocity, general	s^{-1}
ω_1	Angular frequency	s^{-1}
$\dot{\omega}$	Angular velocity	s^{-2}; $\frac{1}{s^2}$
ω_2	Angular velocity of secondary unit	$\frac{1}{s}$
ωt	Angular position	degrees; °
ω_{corner}	Corner frequency	$\frac{1}{s}$

The Development of Secondary Control at Mannesmann Rexroth

Introduction

The development of secondary control began for Mannesmann Rexroth in September 1977 when Professor Nikolaus of the Army University in Hamburg applied for a patent on the fundamental principles of secondary control.

Simultaneously, yet independently the author began looking into circuits for the operation of rotary hydrostatic units in a constant pressure system without the need for throttling elements in the energy lines.

The result of these tests was the speed control of a variable hydrostatic unit at constant pressure by means of torque adjustment, achieved by a change in displacement volume (see **Fig. 12**).

Although contradicting the flow coupling laws, this circuit was technically highly complex.

The first attempts by Professor Nikolaus to find a partner to work on his idea were initially unsuccessful. The reasons were as follows:

- Secondary control was already a familiar term within the industry, but at that time it meant simply an open loop control process on the motor side.
- The unproven scientific claim that "self-blocking" would occur with motor operation or a small swivel angle was also considered a point against the introduction of secondary control. A continuous zero position flow was not considered possible.
- The control experts raised the objection that a dual-integrated system, of which secondary control is one, is virtually impossible to stabilise.

These reasons led to a rejection of secondary control by drive technicians not familiar to the industry on the grounds of technical impossibility. Not until 1979, when Professor Nikolaus took up contact with the author, was the basis formed for a co-operation and potential marketing of the concept.

A decision was made to work together on a joint project for the technical development of a drive based on this concept.

Before work was started on the actual development all patentable ideas were collated.

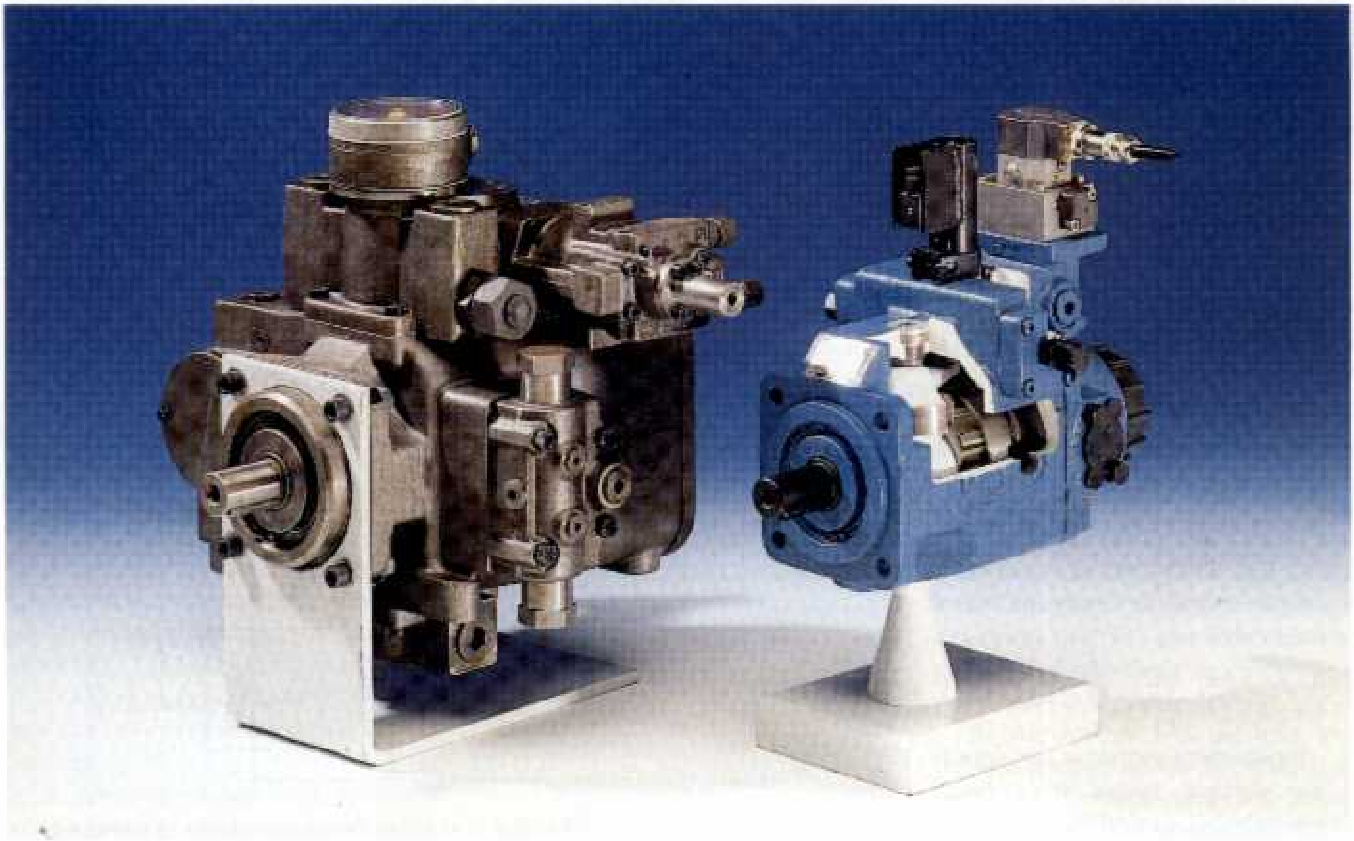


Fig. 1: The first ever secondary controlled unit, axial piston unit type 71EW in bent axis design, with hydraulic tachometer connected (left), and an up-to-date secondary controlled axial piston unit type A4VSO40DS1 in swashplate design, with electrical tachometer (right)

Patent protection

This resulted in a patent application by Mannesmann Rexroth in 1980 with 38 claims under protection of the earlier priorities. This was to keep competitors away from this high-dynamic drive technology.

It came as a complete surprise however, when, at the end of 1980, a registered patent from May 1962 by engineers Pearson and Burrett was unearthed in England, which forestalled all these basic ideas of secondary control with the aid of a hydraulic tachometer. Only the fact that technology at that point was not sufficiently advanced to be able to control high dynamics, and also that there were no suitable machines available, prevented this from reaching the industrial application stage.

In September 1992, after a period of 12 years, the Mannesmann Rexroth application of 1980 was finally patented. The concept of the combination of an electrical tachometer and electrical swivel angle feedback for stabilisation of a system was acknowledged by the patent office as an invention. Secondary control as defined and designed by Mannesmann Rexroth is thus protected by patent right until the year 2000.

In addition to this fundamental patent there are approximately 30 other patent applications or patent rights already granted, pertaining to secondary control. These include:

- hydraulic transformers,
- the inclusion of hydraulic cylinders in a secondary controlled drive system with energy recovery,
- torque control by means of swivel angle feedback,
- a so-called trick circuit for fast torque built-up,
- displacement control at low speeds for uniform torque transmission (creep speed control),

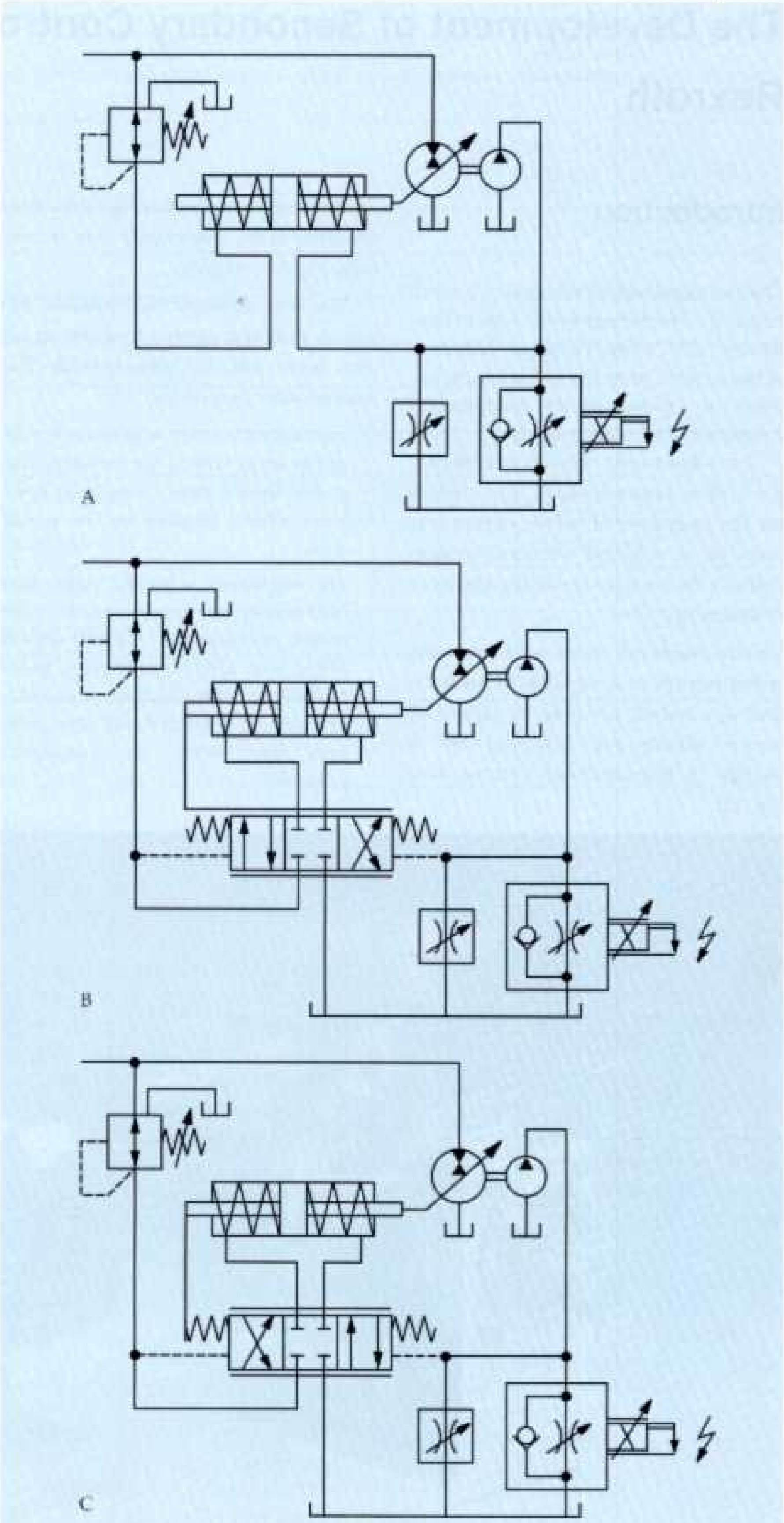


Fig. 2: Possible variants for signal feedback with hydraulic speed control in an impressed pressure system

- a safety circuit to prevent excessive speeds or acceleration and
- electronic power limitation for up to several actuators.

Initially repeated attempts were made to find another term for this new drive concept, especially as this English translation "secondary control" intimates rather second rate. Terms such as "Motor operation in a constant pressure system" were considered. Such a term, however, cannot not be used as it is not strictly correct. Secondary control works in four quadrant operation i.e. bi-directionally as a hydraulic pump and hydraulic motor. The operating pressure is not constant, but impressed i.e. it changes by means of the loading condition of the hydraulic accumulators that are normally part of the drive system.

The term "impressed" was borrowed from electrotechnology. In electrotechnology it is a known fact that with power converter fed motors the DC intermediate circuit works either with impressed current or impressed voltage.

Today the term "Secondary control" is widely known. In November 1979 a contract was signed between Professor Nikolaus and Mannesmann Rexroth, Lohr, enabling the Institute of Hydrostatic Drives and Controls of the Army University in Hamburg to participate in the system development using the test equipment available there.

In June 1980 initial tests were carried out on an axial piston pump in bent axis design, type 71EW, with mounted hydraulic tachometer and flow-dependent positioning cylinder (**Fig. 1**). The first test results were not very encouraging. The basic structure was found to be unstable (**Fig. 2** and **Fig. 3**, A).

As a result of this the flow-dependent positioning cylinder was replaced by a hydraulic pressure-dependent pilot control with mechanical feedback of the positioning cylinder to the pilot valve (**Fig. 2** and **Fig. 3**, B).

This test piece was used to research into the basic requirements for system stability.

It was determined that optimum system stability can be achieved in the secondary controlled circuit if:

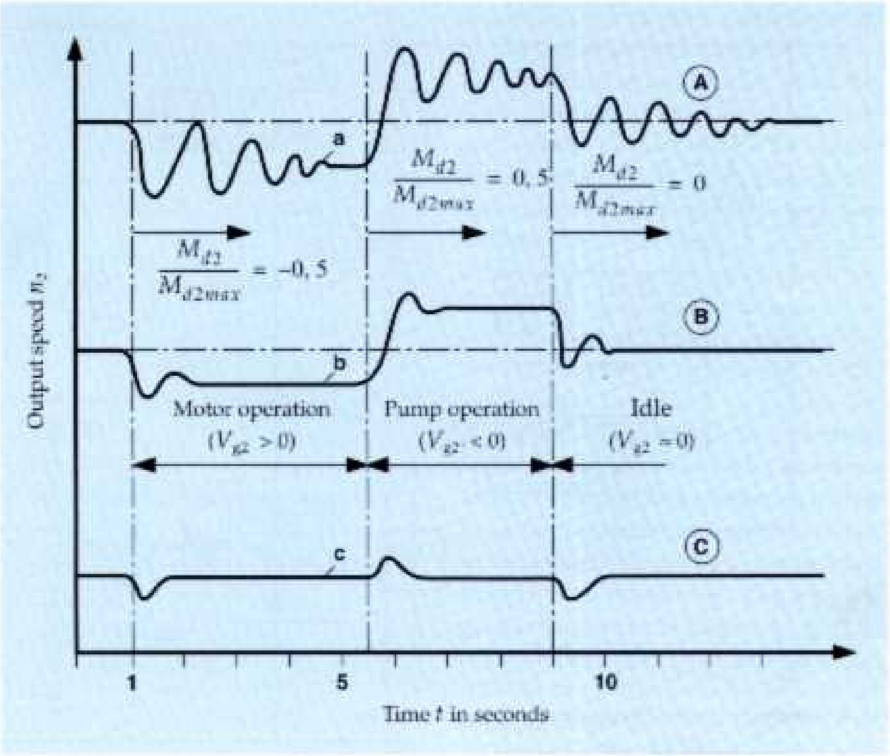


Fig. 3: Reaction of the secondary unit with hydraulic tachometer to stepped changes in torque; circuit to Fig. 2

- the secondary unit possesses sufficient acceleration and deceleration reserves i.e. the unit must be reversible and sufficiently large,
- on reaching the command speed the acceleration or deceleration part of the torque is equal to zero.

These requirements can only be fulfilled if:

- the measurement of speed deviation is free from backlash and with no interference and
- the change of torque is without delay.

The more precisely these requirements are fulfilled, the more stable and free from problems the control concept is likely to be.

After evaluation and analysis of all test results with this new "pressure-dependent positioning cylinder with mechanical positional feedback", the specification was produced for a more suitable positioning system. It came as something of a shock that the HD version (hydraulic adjustment, pressure dependent) of the mobile type A4V fulfilled all the requirements of this specification.

As from June 1982 for comparative measurements with the 71EW / S3041 unit used up until then, the axial piston unit A4V90HD in swashplate design with force feedback of the positioning system was used for the test programme (**Fig. 2** and **Fig. 3**, C).

The evaluation of this test series underlined the technical superiority of the swashplate over the bent axis design for secondary control, in addition to the positioning system with force feedback, which is more suitable for secondary control.

As expected, system stability increases noticeably as we progress from version A through B to C (**Figs. 2** and **3**). Also as part of these tests the influence of the control time on the system characteristics was recognised and evaluated. Control time needed to be reduced drastically, and this was achieved most quickly and effectively by mounting a servo valve in the control circuit.

By the end of 1982 all tests and planned research on the A4V90HS axial piston unit had been completed. Swivel angle feedback was effected electrically and the tachometer was fitted with sig-

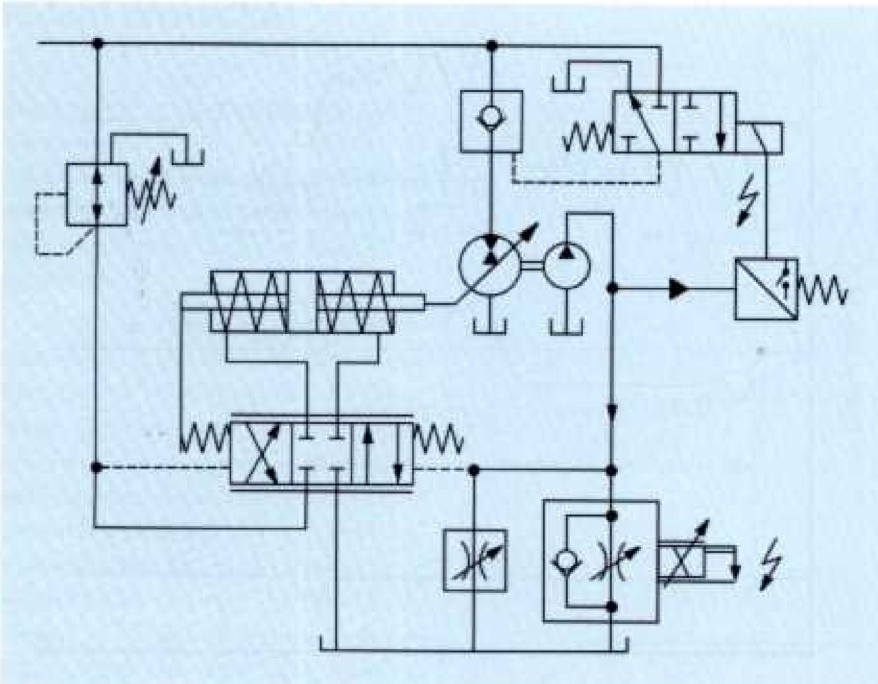


Fig. 4: Secondary control circuit with speed safety device

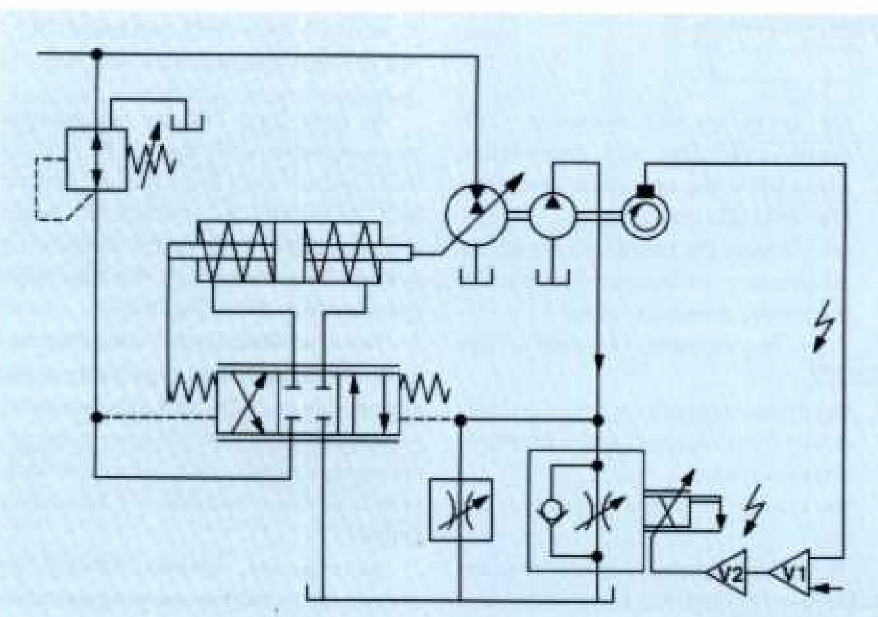


Fig. 5: Secondary control circuit with hydraulic and electrical determination of speed

nal processing, initially analogue, and later digital.

All closed loop circuits shown in **Figs. 2** and **3** depend on perfect functioning of the speed tachometer (hydraulic or electrical), as is the case with power converter fed tachometers.

If the tachometer breaks down the control circuit will become unstable and "crash". A speed limiting system is built in to prevent this. Even in the early stages of development a safety system was considered for protecting the control circuit in the event of the hydraulic tachometer failing (**Fig. 4**). The signal pressure in the speed control circuit is monitored by means of a pressure switch. If this pressure drops to below a pre-set minimum value the electrically operated check valve will be activated. The energy supply to the secondary unit will be cut off, the brake function however being retained, as the braking energy can drain off into the hydraulic system.

In the circuit illustrated in **Fig. 5** the speed of the secondary unit is determined both by hydraulic and electrical means. The current signal generated by the electrical control circuit acts on the electrically operated flow control valve.

This process also increases
- the positioning speed,
- system stability,
- operational safety and
- control precision of the drive.

If the electrical tachometer fails the control function of the drive will be maintained, albeit with reduced control accuracy and system stability.

The circuit designs represented in **Figs. 2** and **4** do not permit change of direction of rotation, as the hydraulic tachometer has to work as a pump to generate the energy required to maintain the control pressure. In order to be able to travel in both directions, the hydraulic tachometer must effect flow in the opposite direction on a change of speed sign. This is achieved electrically, as illustrated in **Fig. 6**. The hydraulic tachometer (1) operates here in closed circuit operation.

Control pressure line (2) is used as a level of comparison, fed by means of the pressure relief valve. In order to receive positive and negative pressure differential signals in both directions, the boost pressure must be reduced even further by means of an additional pressure relief valve (3).

The required flow volume only has to replace the leakage loss from the hydraulic tachometer (1) and is therefore negligible. For pre-setting each direction of rotation a separate control valve (flow control valves (4) and (5)) can be used in the control circuit. Another alternative is a proportional valve with flow control function, acting in both directions. This combines the functions of valves (4) and (5). The tachometer (1) generates pressure differential signals with the relevant sign and proportional to the control deviation, by means of the throttle (6) in both directions of rotation of the secondary unit. The circuit shown in **Fig. 6** can, in other words, work in four quadrant operation.

Before development was concentrated entirely on the electrical tachom-

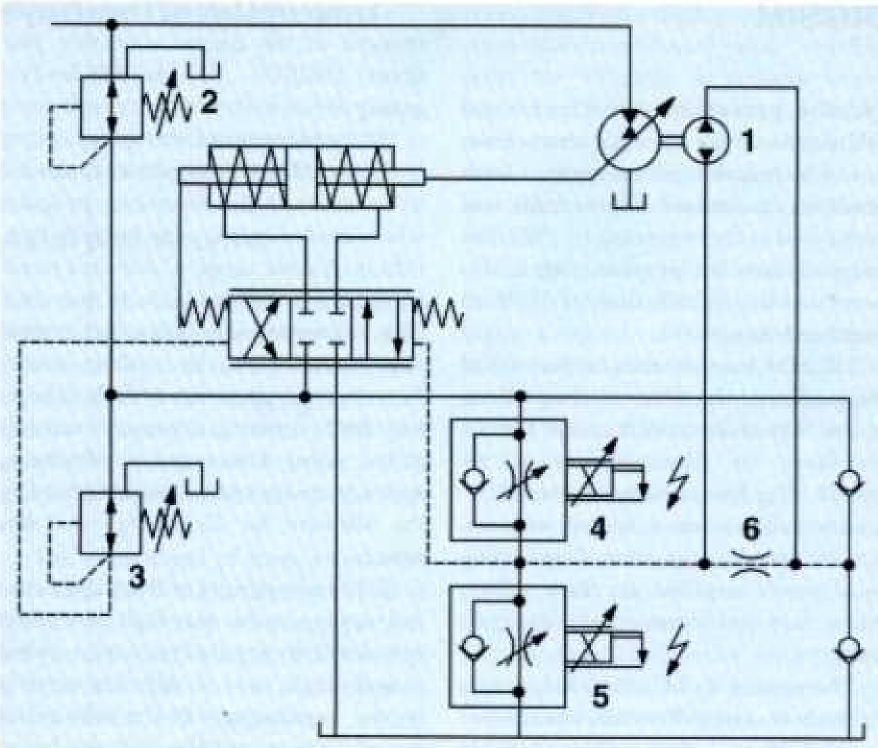


Fig. 6: Secondary control circuit with reversable direction of rotation

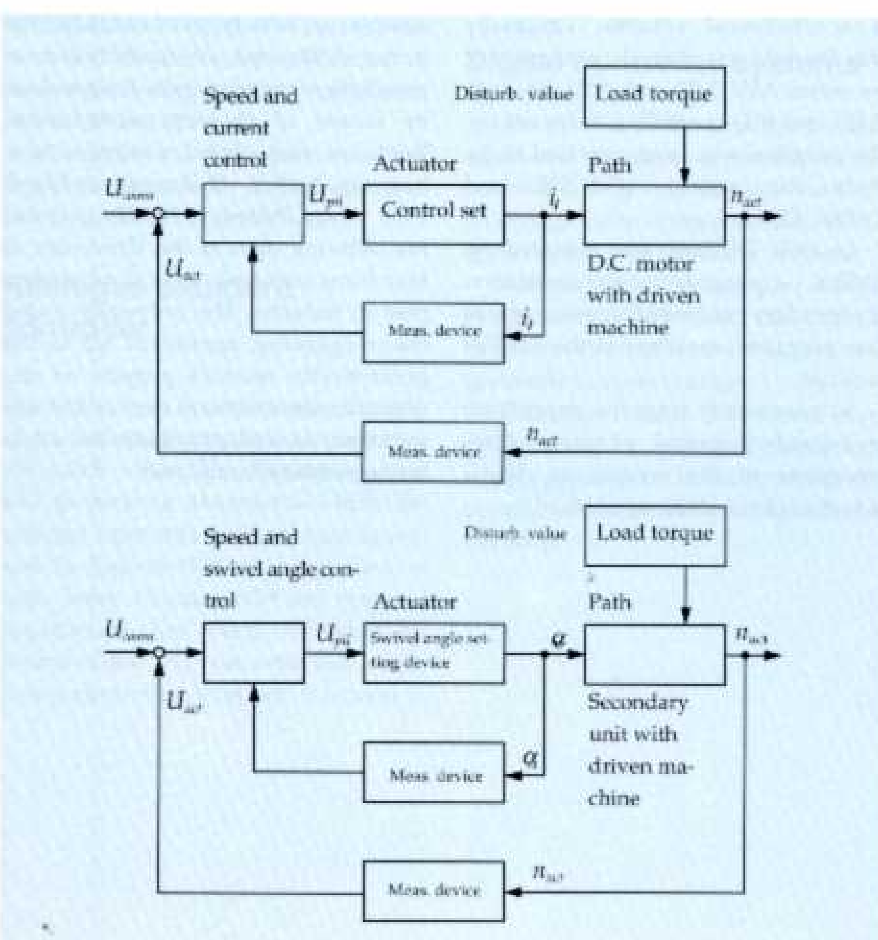


Fig. 7: Signal flow chart for speed control of a D.C. motor (top) and a secondary unit (bottom)

eter all basic tests were carried out using the hydraulic tachometer.

After this standard had been defined, concrete offers could be made and within a short time four orders were booked in. Even at this stage it was clear that this drive concept was destined for great things, being ideally suited for the high demands placed on it in terms of dynamic response, precision and reproducibility required for test and simulation techniques in automobile manufacture.

In this context we must mention Mercedes Benz, as they were prepared, without any previous references, to take a risk by introducing secondary control, as the advantages associated with it had already been recognised. The result of good co-operation was initially two test stands, which were put into operation without any problem and are still in operation today.

However, the development of a control electronics system geared to secondary control proved somewhat difficult. The experts in hydrostatic drives were also not easily convinced.

There was therefore initially no choice but to work together with electrical control experts on the transition from electrical machines to secondary controlled axes.

Signal flows of both drive concepts are basically the same. With an electrical drive the current is fed back, with a hydraulic drive it is the swivel angle (**Fig. 7**). There are, however, considerable differences in the control characteristics.

The dynamic characteristics of the controlled electric motor is determined by high natural moments of inertia. Damping is better with electrical drives and non-linearity is not so pronounced. As the dynamic parameters are only to a small extent dependent on load, it is sometimes possible to pre-set the control circuit without detailed knowledge of the particular application. It is therefore not surprising that with simple control concepts the true capacity of hydrostatic machines, especially when they are secondary controlled, cannot be fully utilised.

There is one characteristic, however, that is peculiar to the conventional hydrostatic drive that is of no significance with secondary control. The volume of oil enclosed in the compression chambers and lines, which, due to its elasticity, forms a second degree oscillation element together with the mass load, has no influence on the system dynamics because of its impressed pressure.

There are, however, problems caused by a frequent lack of damping of the controlled system and also by the presence of dominating non-linearities. In addition to this the natural moment of inertia of the hydrostatic unit is small in comparison to the moment of inertia of the load, so that the load associated with the drive is the main factor in determining the dynamic characteristic of the drive.

A good knowledge, both of the machine to be driven and of the operating procedure, is thus required for the technical design of the control system.

In addition to the demand for a drive-specific control electronics system, a simulation computer program was therefore also sought. This was to enable static and dynamic transfer characteristics to be estimated as early as the project design stage.

This led to the laboratory for hydrostatic drives and controls at the Army University in Hamburg being given the job of developing a simulation program for secondary control at the beginning of 1984.

DIGSIM

DIGSIM, a program system for DIGital SIMulation of the dynamic characteristics of technical machine systems with results in tabular and graphic form, was completed at the beginning of 1985. The program item for graphic output also permitted the reproduction of digitised measured values.

DIGSIM was suitable for numerical integration of ten differential equations of the first order, which could also be non-linear or discontinuous as required. The Runge-Kutta-Merson integration process was followed with automatic step-by-step control according to a pre-determined accuracy range, taking into consideration any discontinuity.

The system to be simulated was in the form of state differential equations.

DIGSIM was thus mainly suitable for the user in technical and scientific areas. However, due to its flexibility, it could easily be used for the simulation of non-technical systems. Originally only the data sets of axial piston units of the series A4V...HD and A4V...EL with P, PI and PID controllers were set up. The program was later extended to include the series A4VS...DS and A10VS...DS.

In 1990 DIGSIM was replaced by HYSEK - a program for the simulation of secondary controlled drives - one of four programs making up the HYSYS package.

At a very early stage the importance for secondary control, of using microprocessors in the control of digital swivel angles, was recognised.

October 1984 saw the start of development of the digital controller program DIGREG (DIGital-REGler-Program).

The advantage of this program lay in the fact that no amendments needed to be made to the assembler program when implementing new control algorithms. A wide range of different transfer blocks were available to the user. This microprocessor-controlled system was developed by Mannesmann Rexroth in co-operation with the laboratory for hydrostatic drives and controls at the Army University in Hamburg, and was sponsored in the first phase by the Ministry for Technology and Research.

Other advantages of this digital control system were the high resolution achievable in the determination of the swivel angle, which depends entirely on the impulse rate of the transmitter per revolution, and the high resolution of the hardware counter module used.

In addition it offered great flexibility with respect to the implementation of complex or new types of controller, together with ease of modification or matching of existing control algorithms by means of program amendments. Work on this digital controller was, however, broken off abruptly in March 1986 when Professor Nikolaus ended his lecturing days at the University of Hamburg and took on a development post in industry. The university ended the co-operation agreement. Up to this point twelve research projects on secondary control, several studies and dissertations as well as a thesis for a doctorate had been completed.

Further Development

From this point all further development in the field of secondary control lay in the hands of Mannesmann Rexroth.

Axial piston units

Although the use of any compressors with variable displacement over centre was considered feasible with secondary controlled hydrostatic drives in four quadrant operation, since 1985 the swashplate design with series A4VS and A10VS has been the one in common use.

The advantages of these two series outweigh the disadvantages, in spite of the fact that start-up efficiency, particularly under load, is up to 8% lower in comparison with the bent axis design.

In March 1987, after axial piston units were adopted as standard in secondary control, product information sheet API 160 was published. This data sheet contained not only the relevant technical data, but also stipulated the rules and directions for application of models with displacement volumes of 40, 71, 125 and 250 cm^3.

The series has since been extended to include units with displacements of 180, 355, 500, 750 and 1000 cm^3.

Analogue standard controller

Since 1986 there has been a specification of an analogue standard controller for the A4VS series. The VT12000 control and monitoring electronics which developed from this was put into operation in August 1988 on completion of tests. Since October 1989 the range of applications has been extended by means of the f/U converter and monitoring electronics VTS0102. It is used in conjuction with a digital tachometer (incremental transducer) and, together with the VT12000, is suitable where high speed and positioning accuracy are required. These electronics are also used for monitoring the signal lines of the incremental transducer for cable breaks.

As the control with the combination of VT12000 and VTS0102 is still analogue, a digital control card developed in 1992 offers a suitable solution, as this can be used for various hydraulic applications but with the same hardware e.g. for secondary control.

This "electronic matching" to the relevant work process is carried out exclusively by the software.

The DSR digital open and closed loop electronic control has been available for the A10VS series since the beginning of 1989. This is an open Eurobus based hardware system with input via menus. The digital positional controller operates with state feedback. The subordinate swivel angle control circuit is still of analogue design.

Digital controller systems

The control of the smallest possible additional moment of inertia on the controlled axis has always been the greatest priority in the continued development of digital controller systems.

A digital adaptive controller is being sought after, which recognises changes in parameters and is able to adapt itself optimally to these changes.

This requirement is not new, especially as digital open and closed loop control systems are in the forefront on a broad basis in virtually all branches of industry.

Summary

The experience that we have gained from the past in the implementation of secondary controlled units has placed us in the position today of being able to make definite statements on the technical limits and possibilities of this system technology. These are essentially determined by the achievable speed of the control system, characterised by the control time. Other decisive factors are the technical data of the controlled area such as moment of inertia and load torque recognition, which are able to give us the application limits in advance.

Future development of secondary control will be aimed mainly at system-oriented transmission units for reducing control time.

Other aims are:

- increasing response sensitivity of the control device with small signals,
- controlling the system from extremely fast controlled areas e.g. with low moments of inertia,
- increasing system safety and accuracy,
- inclusion of translatory analogons into the secondary controlled drive system and
- reducing losses by improving volumetric efficiency.

No doubt some of these characteristics which have been developed for secondary control will also be used to advantage with conventional drive systems with flow coupling, as has repeatedly been the case in the past.

Basic Principle of Secondary Control

Introduction

Secondary control, a hydrostatic drive concept in a hydraulic system with impressed operating pressure, has been in worldwide use since 1980. As the special characteristics can be more easily compared with electrical drives with closed loop control than with conventional hydraulic drives, project design together with the application of secondary controlled drives is still only used by a small group with specialist knowledge.

Secondary control is used predominantly where a conventional drive is no longer able to fulfil the technical requirements in terms of dynamic response, positioning and precision control of speed and torque.

One important criterion is energy recovery with or without conversion into another energy form.

In drive technology two power transmitting parameters are of importance:

- torque M_T in Nm and
- speed n in rpm.

These mechanical parameters correspond to the following parameters in hydrostatic drives:

- pressure p in bar corresponds to torque M_T and
- flow Q in L/min corresponds to speed n.

Depending on the coupling of the mechanical and hydraulic parameters we differentiate between two types of drives. These are:

- drive systems with flow coupling (conventional systems) and
- drive systems with pressure coupling (systems with control of the secondary unit).

Drive systems with flow coupling

Before looking at the system-dependent characteristics of secondary control it is a good idea to start by demonstrating the basic principle of the conventional hydrostatic drive system that operates with a flow coupling.

In hydrostatics pressure differential Δp and flow Q are modified in order to influence the power to be transmitted:

$$P_{hyd} = \frac{\Delta p \bullet Q}{600}$$

where
P_{hyd} = hydraulic power in kW,
Δp = pressure differential in bar,
Q = flow in L/min.

The conventional hydrostatic drive consists of a hydraulic pump with variable displacement V_{g1} and a hydraulic motor with fixed volume V_{g2} (**Fig. 8**). By varying the displacement V_{g1} e.g. by means of swivel angle a_1 or input drive speed n_1, volumetric flow Q of the hydraulic pump

$$Q_1 = V_{g1} \bullet \frac{\alpha_1}{\alpha_{1\,max}} \bullet \frac{n_1}{1000}$$

will be changed and thus the output speed n_2 of the hydraulic motor determined.

Flow Q, determined by drive speed n_1 and volume V_{g1}, causes the hydraulic motor to achieve an output speed n_2 depending on its displacement V_{g2}:

$$Q_1 = Q_2 = \frac{V_{g1} \bullet n_2}{1000}$$

From this it follows that

$$n_2 = n_1 \bullet \frac{V_{g1}}{V_{g2}} \bullet \frac{\alpha_1}{\alpha_{1\,max}}.$$

where
M_{T1} = drive torque in Nm,
M_{T2} = output torque in Nm,
n_1 = drive speed of hydraulic pump in rpm,
n_2 = output speed of hydraulic pump in rpm,
Q_1 = flow of hydraulic pump in L/min,
Q_2 = flow of hydraulic motor in L/min,
V_{g1} = displacement of hydraulic pump in cm^3,
V_{g2} = displacement of hydraulic motor in cm^3,
α_1 = swivel angle of hydraulic pump in °,
α_2 = swivel angle of hydraulic motor in °.

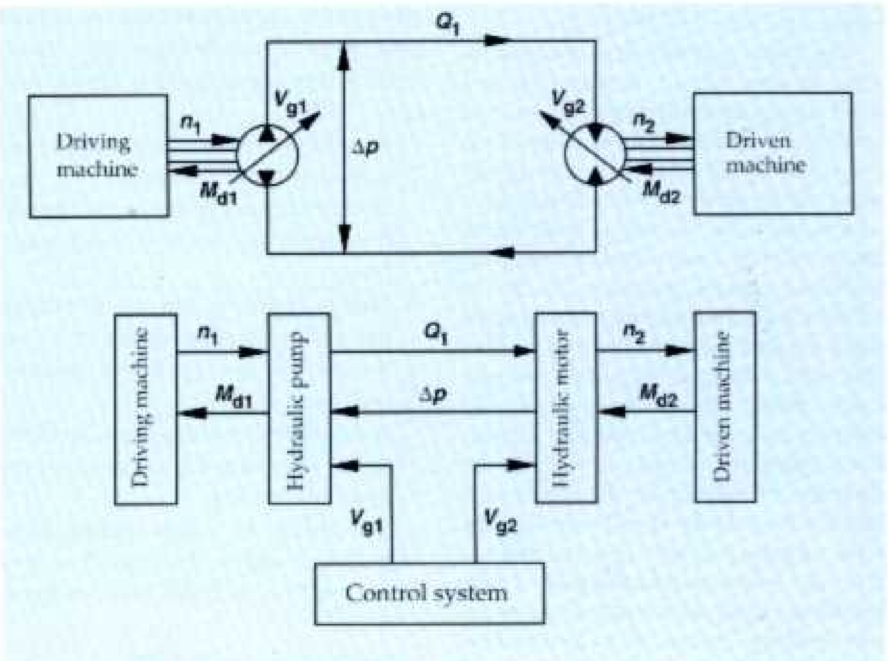


Fig. 8: Basic circuit of a conventional hydrostatic drive

A pressure differential Δp, corresponding to the load characteristic of the drive performance graph, will occur at the hydraulic drive:

$$\Delta p = \frac{20 \cdot \pi \cdot M_{T2}}{V_{g2}}.$$

With any change in torque required at the output end the system will respond with a change in pressure Δp. The oil column between the control element and actuator will be either compressed or expanded. With dynamic response operations the "hydraulic spring" may have a considerable adverse effect on system stability. It is advisable in practice to damp the system by increasing control time of the hydraulic pump. This enables pressure build-up or reduction and thus system stability to be kept under control. This applies in particular to the closed circuit represented in **Fig. 8**.

In heavy engineering and shipbuilding it is common practice to install hydraulic systems with a central oil supply. These operate at a constant pressure using pressure regulated pumps and can have several actuators connected in parallel. In order to ensure that all the fluid does not flow through the actuator with the lowest level of resistance, it is necessary to install throttling elements in the energy transmission lines. These ensure that the required flow reaches the individual actuators. Thus, in order to obey the laws of flow coupling, the constant pressure system is converted into a constant flow system.

Fig. 9 shows an example of two actuators in open circuit. Actuator (1) is a fixed displacement motor with an electrical tacho-generator for a closed loop speed-controlled drive. The control element can be either a proportional valve or a servo valve. Energy is supplied to the system by two pressure regulated hydraulic pumps. Actuator (2) can be either a fixed displacement or a variable displacement motor. Maximum flow to this unit is limited by a flow control valve. Below this maximum flow the motor is controlled by means of a directional valve, which can control both the direction of rotation of the motor and can also throttle the speed even further. If the output units tend to act as generators i.e. when decelerating or when lowering a load, the energy is converted into heat by means of a deceleration valve.

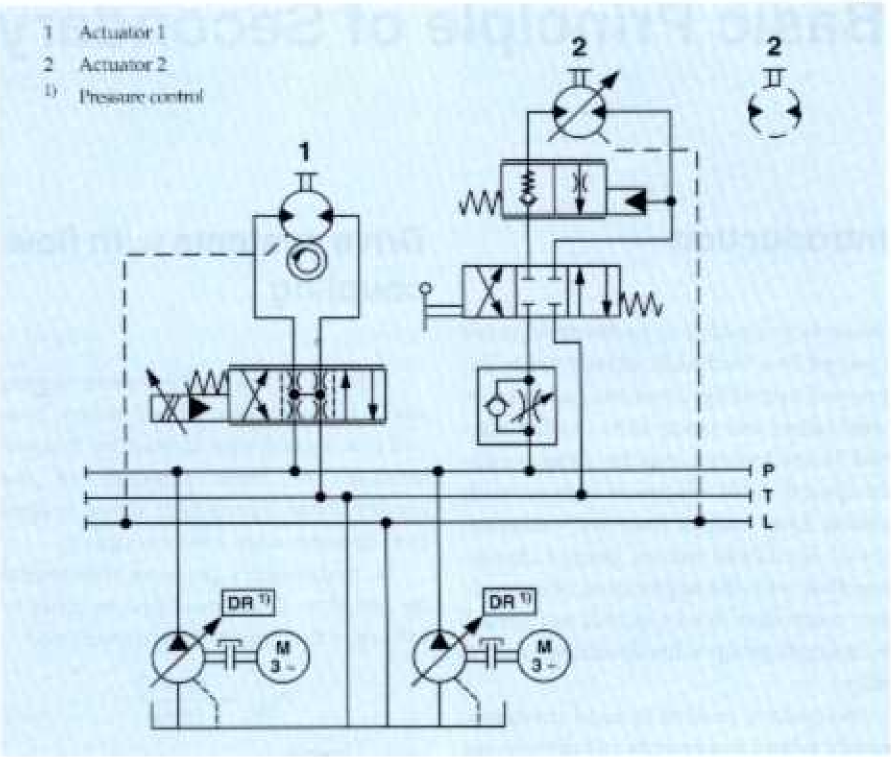


Fig. 9: Conventional hydrostatic drive

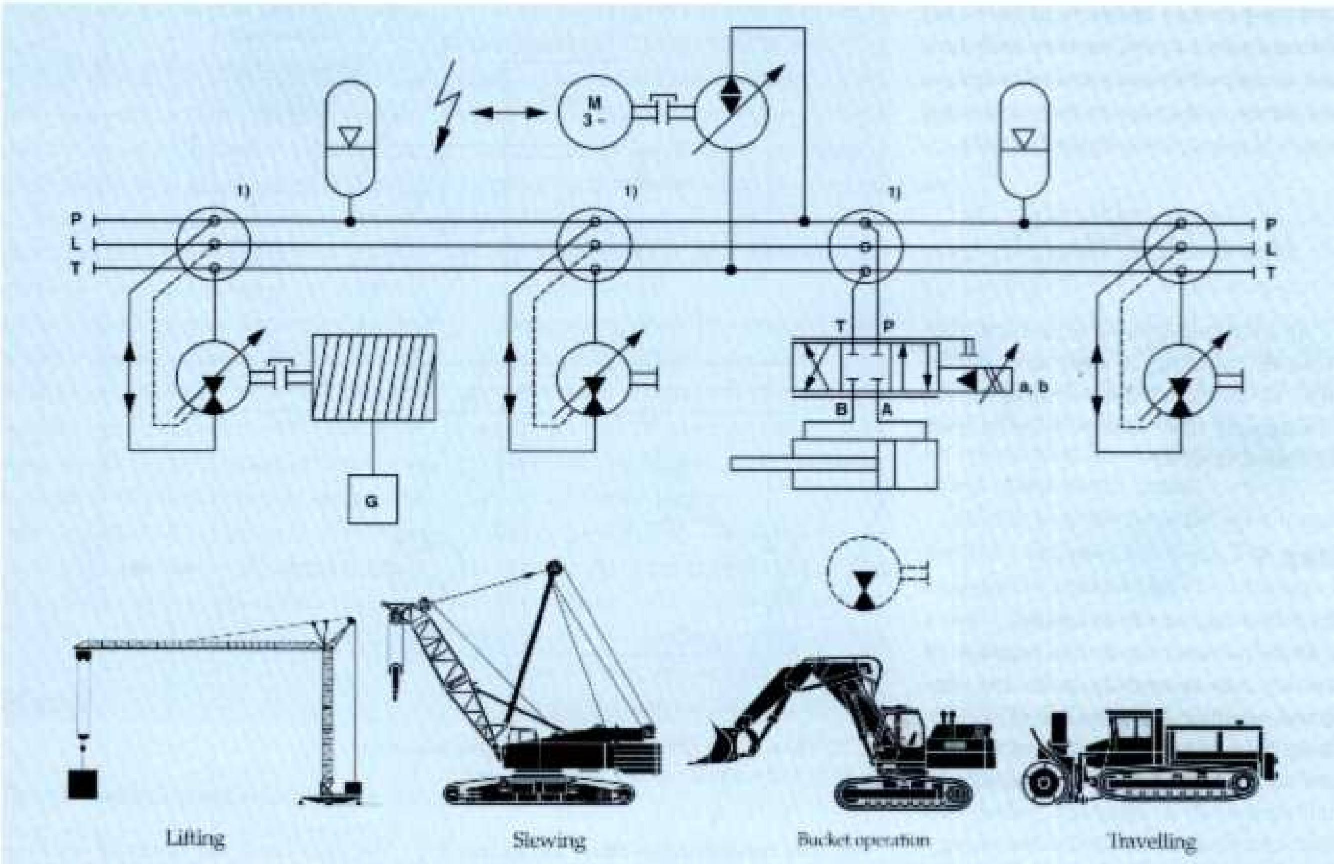
In the throttling operations illustrated here and especially under partial load conditions, a considerable amount of heat is lost due to the proportion of pressure generated by the pump which is not required by the actuator for a given flow volume. The energy balance leaves much to be desired, energy usage at the primary end being correspondingly high.

It was thus necessary to look for another drive concept which would not have these disadvantages and which would have the following characteristics:

- Parallel operation of a number of actuators without limitation.
- Energy transfer from the primary to the secondary units without throttling.
- Energy recovery without throttling, for use by other actuators or by returning the energy to the primary unit.
- A constant operating pressure in order to eliminate the influence of the hydraulic spring.
- The ability to accommodate accumulators within the system at any required point, again without throttling.
- Four quadrant operation

These drive concepts can be illustrated as in **Fig. 10**. A pressure differential Δp is generated in a hydraulic system by the input of energy. The pressure level in the system is also determined by the loading condition of the accumulator. By means of isolating elements as many actuators as required can be connected to the ring main. There are no throttling elements in the energy transmission lines. When the actuators are working as motors, energy is drawn from the system. When they are working as generators, energy is returned to the system. The energy which has been recovered may be used by other actuators or may be stored for later use or even returned to the energy supply unit and converted into another form of energy e.g. electrical energy. As the operating pressure remains virtually constant, the influence of the hydraulic spring is no longer of importance, the dynamics of the system have free rein, the energy balance is improved and the primary energy usage is considerably reduced.

It now remains to find a technical solution which permits such a system to be controlled in every operating condition, so that any influence from interaction of the actuators is excluded from the system. This drive concept we call secondary control.



1) „Hydraulic isolator" (see Fig. 18)

Fig. 10: Drive concept with some advantages over a conventional system

Drive systems with pressure coupling

If a hydrostatic transmission consists of three energy carriers, comprising at least one hydraulic pump, one accumulator and one actuator element, it is possible to store energy in the operating circuit (**Fig. 11**).

This type of transmission design differs from conventional ones in that the system pressure is dependent on the loading condition of the accumulator and can no longer be freely matched to output torque M_{T2}.

Such hydrostatic drive systems are known as hydrostatic transmissions with "impressed" operating pressure, a term taken from electrotechnology.

Output drive torque M_{T2} is determined by varying displacement V_{g2} of the secondary unit.

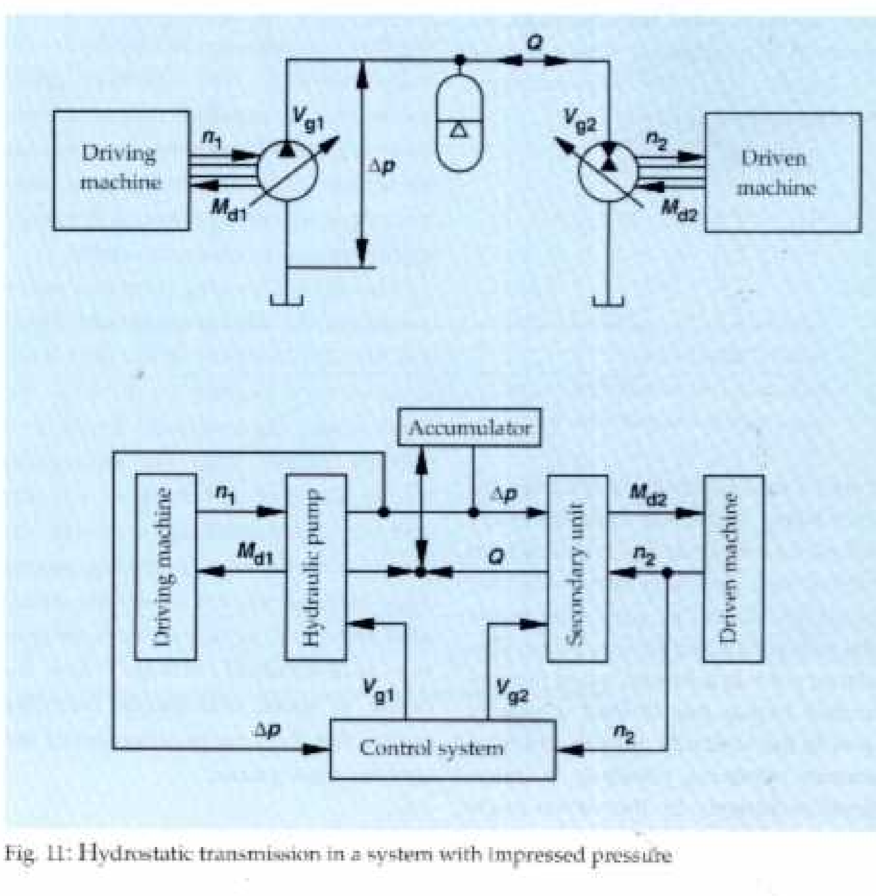


Fig. 11: Hydrostatic transmission in a system with impressed pressure

At a pressure differential Δp and constant speed flow Q to the secondary unit is proportional to swivel angle α_2 and hence also proportional to output torque M_{T2} according to the formula:

$$M_{T2} = \frac{\Delta p \bullet V_{g2max}}{20 \bullet \pi} \bullet \frac{\alpha_2}{\alpha_{2\,max}}$$

An easy step-by-step introduction to pressure coupling is demonstrated in **Figs. 12** to **15**, as this method differs considerably from that of systems with impressed flow.

Step 1

The following conditions apply.

In a hydraulic system a number of primary and secondary units are connected together in parallel (**Fig. 12**). These may operate either as motors or generators. The operating pressure is maintained at a constant value. No throttle elements are fitted in the energy transmission lines. The hydraulic circuit can be either open (**Fig. 12**) or closed. The displacement of the hydrostatic units, with either axial or radial piston units, shown here as a winch drive, can be separately adjusted over the zero point i.e. both value and direction, by means of a mechanical screw control with a hand wheel. The torque changes proportionally as follows:

$$M_{T2} = \Delta p \bullet V_{g2} = \Delta p \bullet f(\alpha_2)$$

where

M_{T2} = output torque of hydraulic motor in Nm,
Δp = pressure difference in bar,
V_{g2} = displacement of hydraulic motor in cm^3,
α_2 = swivel angle of hydraulic motor in °.

If, after a load is applied to a winch, the hand wheel is turned experimentally backwards and forwards, it will be seen that the load rises and falls at varying speeds and it will be easy after a few tests to ascertain the balance point. This balance point is achieved when the mechanical torque due to load is exactly equal to the hydraulic torque which, at constant operating pressure, is determined exclusively by the swivel angle of the unit.

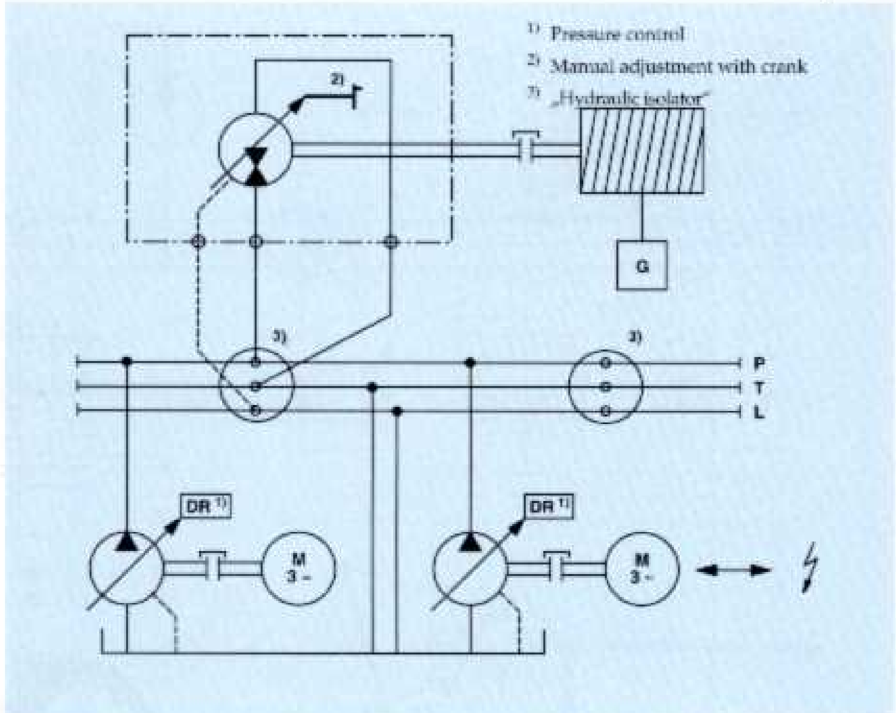


Fig. 12: Hydrostatic drive in a constant pressure system

In this condition no other forces are in operation. The load speed is zero and the load is held in position regardless of the volumetric efficiency of the unit and without the need to apply a mechanical brake. This characteristic is somewhat perplexing if one is accustomed to saying that a load cannot be held hydraulically. However, this statement refers solely to conventional hydraulic systems with flow coupling. In the case presented here, the volumetric efficiency has no effect on torque if the operating pressure remains constant.

However, if moving from this stable condition, the swivel angle and therefore the displacement of the unit is increased very slightly by turning the hand wheel, the hydraulic torque will become greater than the mechanical torque, the unit will work as a motor and the load will be raised slowly. Velocity increases with increasing swivel angle (torque) so that an exactly definable "flow" will be drawn from the system at a constant "voltage". Any increase in speed and torque therefore means that the flow requirement of the unit becomes greater.

We thus have the unusual relationship that a change in torque produces a similar change in flow. The analogy to closed loop electrical drives thus becomes clear.

If, starting once more at the balance point, the swivel angle is reduced, the mechanical torque will become greater than the hydraulic torque. The unit will then work as a generator i.e. a pump, and the load will fall. The potential energy will then be fed back into the hydraulic ring main.

The direction in which the pressure is acting will remain the same in this condition, even though the direction of rotation has been reversed.

If the winch is to be lowered with an empty hook, the swivel angle must be reversed over zero into the negative area. This means that the unit must once more act as a motor, as the load due to the empty hook is too small to cause the unit to act as a generator.

The first step has little in common with secondary control. It has been described only to show the possibility of achieving a throttle-free drive of a number of units in parallel with pressure coupling via a constant operating pressure.

The overall characteristics of this drive system in which a torque is generated hydrostatically and the driven machine reacts with an appropriate speed is totally different from the requirements in practice. Normally a pre-set speed is demanded of a hydrostatic drive. The latter then generates the required torque automatically via a closed loop system in order to be able to hold the speed constant under the operating speed being applied at that time. In order to be able to operate such a system economically, technical modifications and additions must be made to the axial piston unit shown in **Fig. 12**.

Step 2

The mechanical positioning device has been replaced by a flow-dependent positioning cylinder. As no statement about output speed n_2 can be made from the flow alone, the lost speed information must be determined another way and fed to the positioning system as an overriding signal value. This determination of speed and the feedback of the signal into the positioning function is assumed in a hydraulic tachogenerator in the control circuit. This tachometer can act either as a pump or as a motor. It feeds the positioning signal to the volume-dependent cylinder and regulates its position such that the speed of the secondary unit is reduced to zero and the required torque is built up.

An equilibrium position is once more achieved when the mechanical and hydraulic torques at the coupling point are the same. This is exactly the case when the pressure differential in the control circuit at the positioning cylinder and the tachometer are zero i.e. when $\Delta p_{pü} = 0$.

If the loading on the winch drum is now changed, equilibrium is lost. The load starts to move and the winch drum turns. At the same time pilot oil from the tachometer is fed to the positioning cylinder so that its position is changed to suit the new torque levels. At the end of this adjustment, a new state of equilibrium is achieved.

Under conditions of constant load, the same process is repeated if the operating pressure is changed. This may occur, for example, if other actuators draw power from the system, thus emptying the accumulator.

In contrast to **Fig. 12**, additional accumulators have been included in **Fig. 13**. This has an influence on the operating pressure depending on the loading conditions. We speak here of impressed operating pressure i.e. the pressure remains nominally constant. However, the pressure varies somewhat due to the amount of charge in the accumulators. This process has no effect on the system characteristics. The circuit presets a speed of zero which is not dependent on the load or the operating pressure.

The load holding process is once more completely independent of the volumetric efficiency of the axial piston unit in the driving circuit but is not entirely independent of any leakage from the control circuit. For example, a leakage in the control circuit at the tachometer would lead to a corresponding creep speed of the winch drum.

In this way we have set up a torque control with speed feedback. The swivel angle of the unit and its displacement is a free value and is not uniquely defined. It changes with the required torque in order to maintain zero speed.

In order to be of use, a drive must be able to be operated. Further additions are therefore necessary.

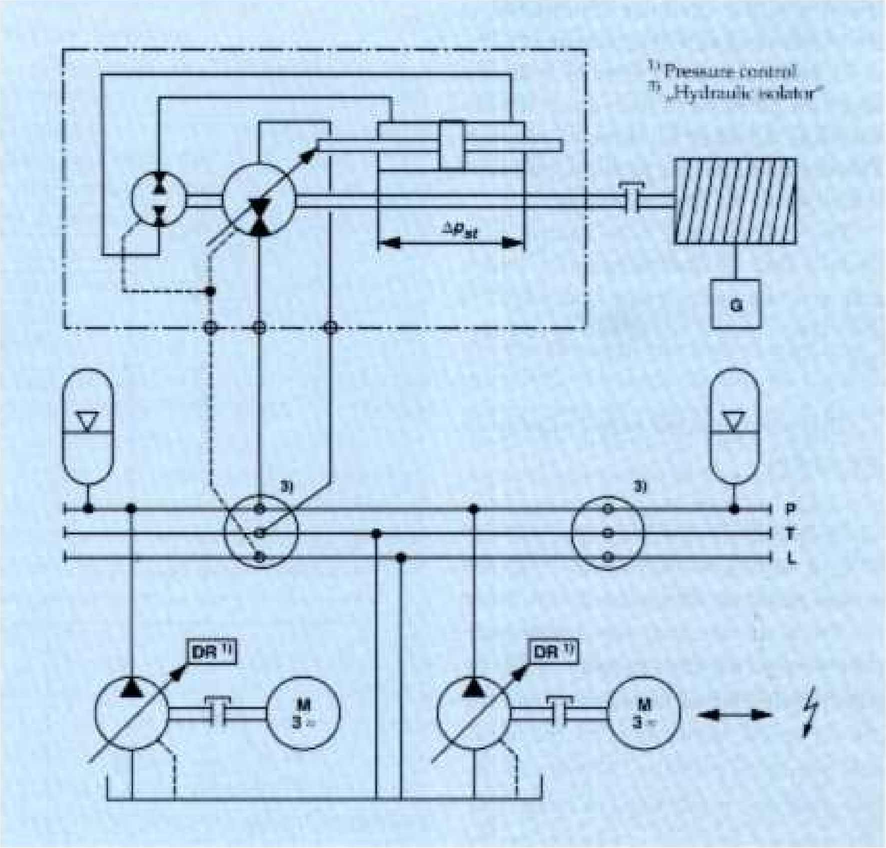


Fig. 13: Secondary control in a system with impressed pressure

Step 3

As opposed to **Fig. 13**, in **Fig. 14** a valve has now been added in the control circuit.

This valve can be either a proportional valve or a servo valve depending on the dynamic response required. This valve is used to select the direction of rotation and to meter the required flow into the control circuit.

With the valve in the neutral position, the characteristics are the same as for **Fig. 13** and the speed is zero. If a flow is now passed via the valve into the pilot circuit, the balance across the positioning cylinder is destroyed with pressure being built up at one side. The positioning cylinder then changes its position and thus alters the torque available at the secondary unit. An imbalance now occurs between the mechanical torque at the winch drum and the hydraulic torque available. The load starts to move and the drum turns. The pilot oil requirement of the tachometer rises proportional to the rotational speed and the pressure differential at the positioning cylinder is reduced.

Equilibrium is once more achieved when the pressure differential at the positioning cylinder is reduced to zero. This occurs when the total pilot oil flow is bled away via the tachometer.

The proportional flow control thus feeds a flow to the tachometer which acts as a command value and which is then taken as the command speed signal.

This process is the secret of secondary control.

In secondary control we therefore have a speed control loop with the swivel angle or torque as a free value i.e. the hydraulic unit seeks automatically to meet the pre-set speed input by adjusting its torque in order to hold the pre-set speed, depending on the available operating pressure. A hydraulic tacho-generator is still used in certan special cases e.g. in fire or explosion hazard zones.

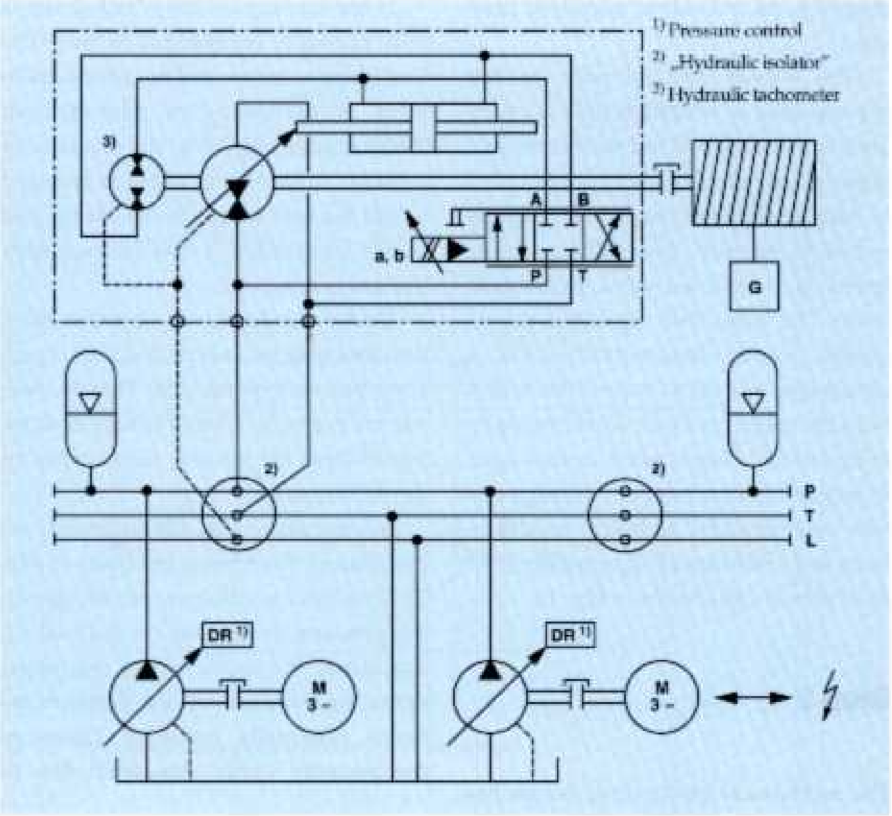


Fig. 14: Secondary control in a system with impressed pressure with hydraulic tachometer

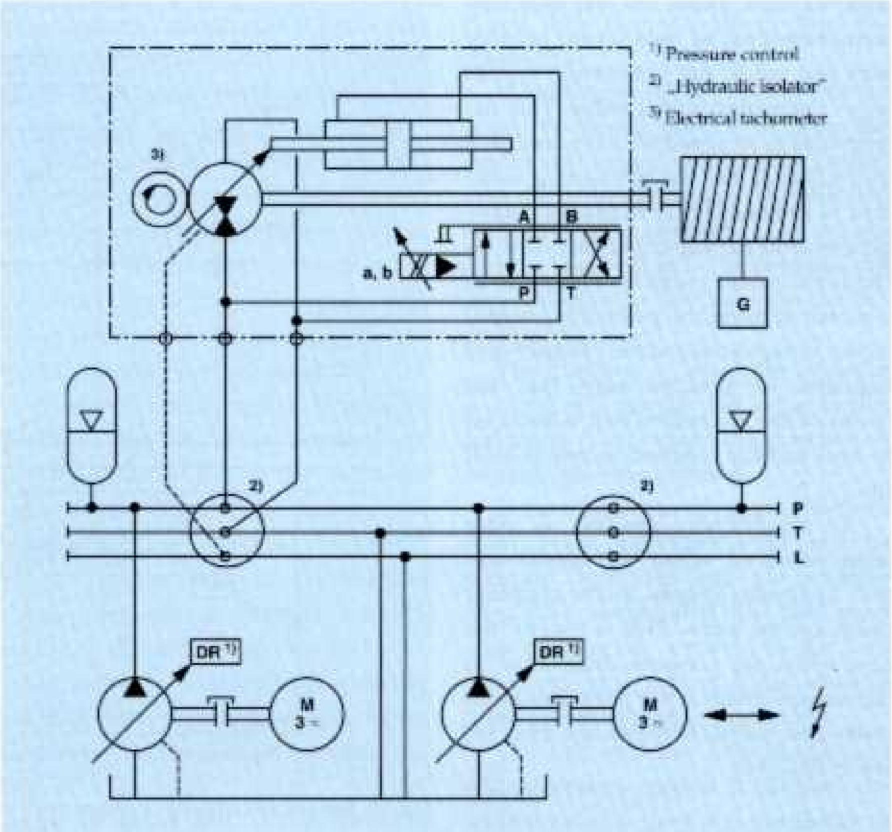


Fig. 15: Secondary control in a system with impressed pressure

Step 4

In the vast majority of applications the hydraulic tacho-generator is replaced by an electro tacho-generator (**Fig. 15**) capable of producing either analogue or digital signals. As leakage in the control circuit is also completely avoided in these cases, load holding may be achieved without difficulty.

Summary

In contrast to conventional drive systems, the swivel angle of the secondary unit is no longer co-related with an exactly defined drive speed n_2 in rpm. Instead, in a system with impressed pressure, it is defined by a definite torque M_{T2} in Nm. When the system pressure changes, the swivel angle is so regulated as to maintain the required torque, and the speed is held constant. In a system with secondary control there is almost loss-free conversion of hydraulic energy into mechanical energy (motor operation) and of mechanical energy into hydraulic energy (pump operation).

A four quadrant drive can even be achieved in open circuit without problem. With the help of secondary control, in a similar manner to an electrical power line operating at constant voltage, as many independent actuators as required can be coupled in parallel in most motor and pump operations. The possibility of almost loss-free energy storage by means of piston or bladder accumulators permits a new energy-saving drive concept to be achieved on a universal basis.

The following characteristics, peculiar to the conventional drive with flow coupling, do not apply with secondary control with pressure coupling, although the laws of physics are not affected.

a) Flow coupling:

"With flow coupling the operating medium flows to the actuator with the least resistance, other actuators connected in parallel remaining in a rest condition".

This statement does not apply to pressure coupling.

b) Flow coupling:

If the displacement of a motor in a conventional hydrostatic drive is reduced, the speed will increase".

With secondary control an increase in speed is only possible if the displacement is increased.

c) Flow coupling:

"If the swivel angle is sufficiently small, "Self-blocking" will occur in motor operation i.e. the hydraulic motor will remain still."

A machine with secondary control will have constant zero flow if the swivel angle is decreased. In the unloaded condition a swashplate unit with 500 cm^3 displacement at 2000 rpm can be operated in the motionless state and motor operation with a swivel angle of less than 1.5°.

d) Flow coupling:

"A change in torque within a conventional system is directly proportional to a change in operating pressure".

With secondary control a change of torque at constant speed means a change in the flow volume required by the axial piston unit.

e) Flow coupling:

"Instability or vibrations within a system can be eliminated either by conventional methods or by the use of throttling elements".

Throttling within the control circuit of a secondary controlled machine will directly cause vibrations - the smaller the reduced moment of inertia where coupled, the greater the vibrations.

f) Flow coupling:

"If a hydraulic pump is used for open or closed loop speed control, only ever one actuator can be used".

With a pressure coupled drive system with impressed operating pressure any number of actuators can be connected in parallel without their interacting on each other.

g) Flow coupling:

"The use of hydraulic accumulators for energy recovery is by conventional methods only possible with a complex control system."

This is not a problem with secondary control. The hydraulic accumulator can be situated in any position without necessitating extra valves.

h) Flow coupling:

"In a conventional system four quadrant operation is only of any use in a closed circuit."

With secondary control this operating mode can easily be applied to an open circuit.

i) Flow coupling:

"During the deceleration process a conventional motor uses the drive unit or valves for support".

With secondary control the drive unit can be decoupled even in open circuit operation. Under certain conditions it may even be switched off, without having any effect on the control process. All that is required is a pressure valve for build-up of pressure.

j) Flow coupling:

"An external load cannot be held by conventional hydraulic means as it will move under the influence of external leakage".

With secondary control the size of a machine has no effect on the load holding process; the load will remain stationary even if there is no mechanical brake.

In the past, for a conventional hydrostatic drive system with flow coupling, the closed loop control concepts required to achieve a good time response and high precision were worked on regardless of whether the system was designed to be pump or valve controlled. This drive system will suffice for both rotary and linear operating actuators.

The pressure difference not required by the actuator becomes energy lost. If throttling elements need to be used for energy recovery when energy is drawn from the system for flow control, the natural advantage of the constant pressure system cannot then be utilised to the full. This is not a problem with secondary control and there is thus no need for throttling elements in the energy lines.

With linear movement by means of a hydraulic cylinder it is not possible to influence the effective piston area. So-called hydraulic transformers must be used to draw energy without a basic loss from a hydraulic system with impressed pressure. The infinite variation in piston diameter will thus occur in an intermediate step, which is described in more detail in a later chapter.

Axial Piston Units Designed for Use with Secondary Control

Swashplate design A4VS

As initially insufficient machines were manufactured which were specifically for secondary control, machines had to be used that were really designed for use with a conventional system. However, the increasing demands placed on this system technology in the course of time have required some design modifications to be made. The result of these modifications was essentially a longer service life of components, a reduction in control time and monitoring of operational safety.

Fig. 16 shows the standard design of an axial piston unit of the type A4VS in swashplate design, for use in a secondary control drive system.

The incremental transmitter with 1000 or 2500 increments per revolution, which is coupled to the second shaft end free from backlash, is also available with an integral analogue output. A mechanical centrifugal switch is fitted as a special accessory between tachometer and the axial piston unit to increase safety. This can, however, only be used to advantage with a sufficiently large reduced additional moment of inertia due to the associated switching delay.

As the speed range per unit time can be extremely high this is discussed in greater detail later on. An inductive positional transducer for electrical feedback of the swivel angle is fitted at the control piston and is connected to this by means of a non-positive inclined plane. This swivel angle feedback (cascade), which is subordinate to the closed loop speed circuit is essential for

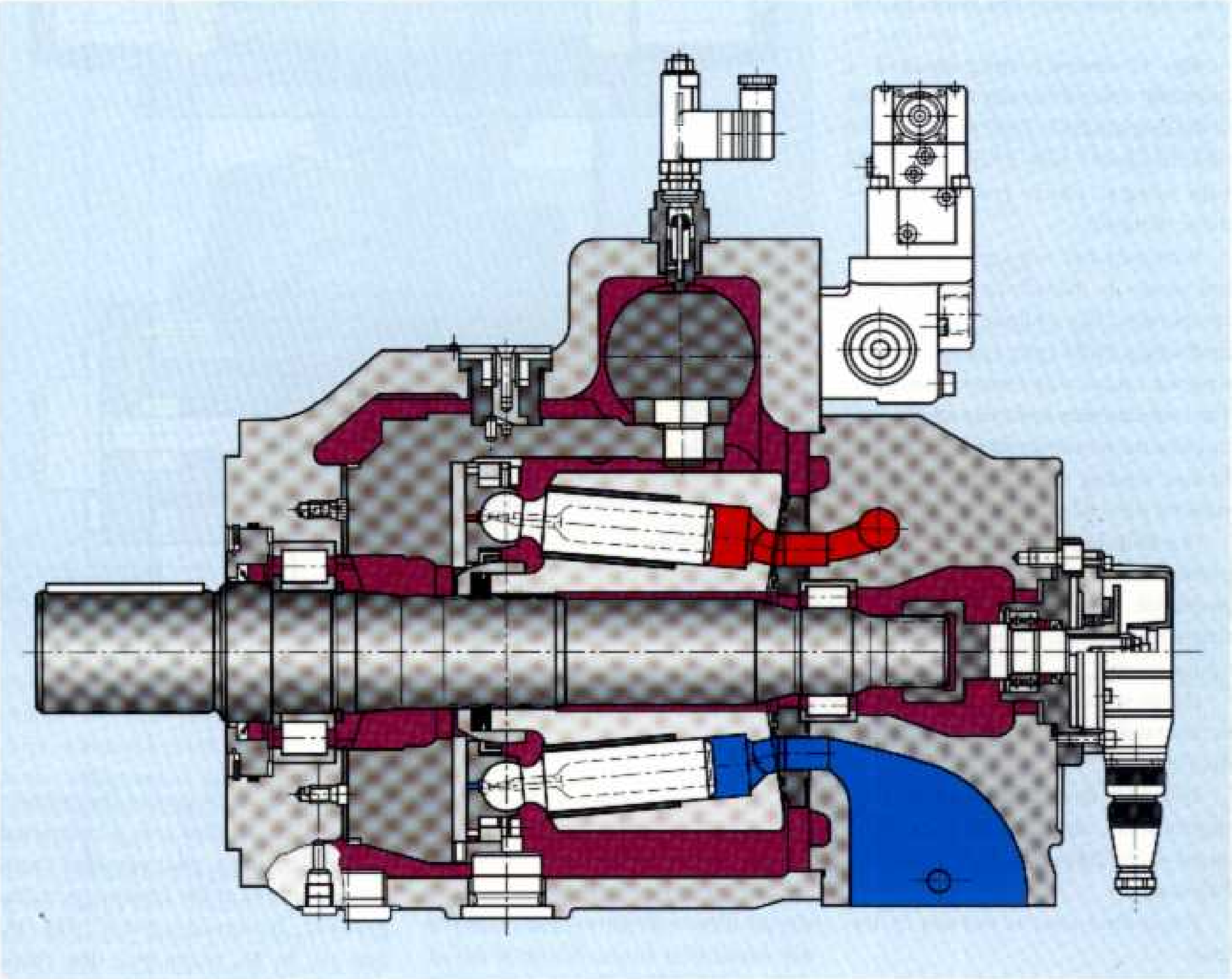


Fig. 16: Axial piston unit type A4VS, swashplate design, for secondary control

system stability, as secondary control is a double-integral system, renowned as being difficult to stabilise. With the feedback an indicator is given for the swivel angle or displacement, so that this can be taken into account with the operating pressure for torque control (see also the chapter on "Special Characteristics of Secondary Control").

Control of the positioning cylinder is by means of either an electrohydraulic servo valve or, with limited dynamic response, a proportional valve.

In principle with secondary control the pilot flow is taken from the operating pressure system. This not only dispenses with the need for a separate pilot pressure system, the increasing pilot forces proportional to the operating pressure are also compensated this way and the control surfaces and hence also the control volume can be reduced. This is not the case with the standard version.

Fig. 17 shows a comparison of a standard control system series (top) with the standard version for secondary control (centre) and a special version with integral piston positional transducer (bottom).

With the latter two versions the control piston is manufactured from one component. The oblique part for the feedback is inwrought, instead of being screwed on, as was formerly the case. The simultaneous reduction of the control area decreased positioning time still further, leading to even greater system stability (see also pages 45 to 51).

Due to its high resolution the control piston version with integral inductive positional transducer can be used where high accuracy is required and for torque control.

However, unlike the inductive positional transducer, this version is dependent on size.

With both these positioning systems control times can be realised in accordance with **Table 1**, which guarantee 20% safety.

The control time is defined as follows:

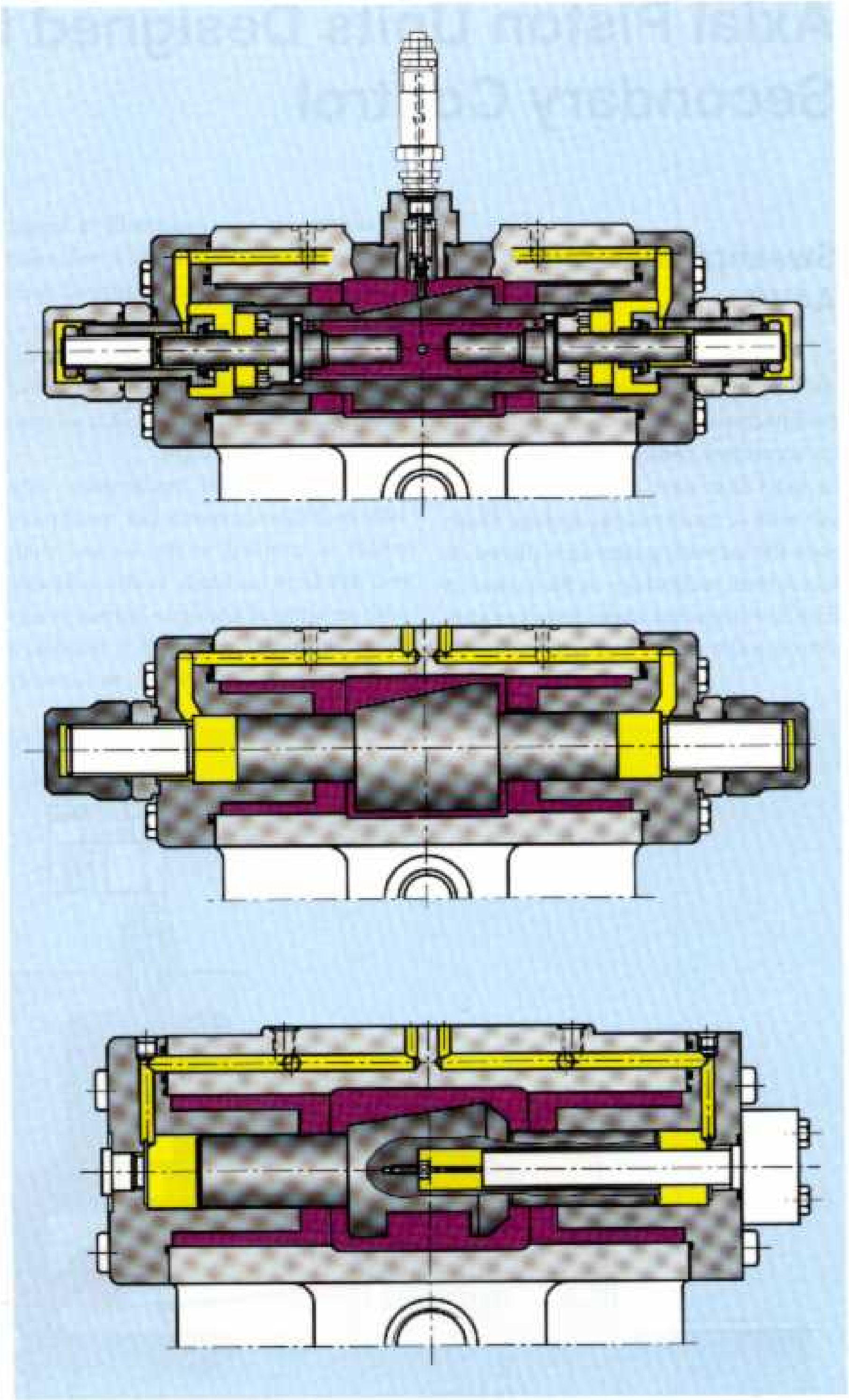


Fig. 17: Positioning systems: Standard series (top), standard version for secondary control (centre), special version with integral piston positional transducer (bottom)

Displacement change from 0 to V_{g2max} or V_{g2max} to 0.

Any further reduction of control time with this type is virtually impossible, as these current figures extend to the mechanical fatigue limits of the associated components.

The mechanical part of the system's secondary controlled axis is completed by the electrically pilot operated check valve mounted at the pressure port, the so-called "hydraulic isolator" (**Fig. 18**, item 4). In an emergency this check valve cuts off the energy supply from

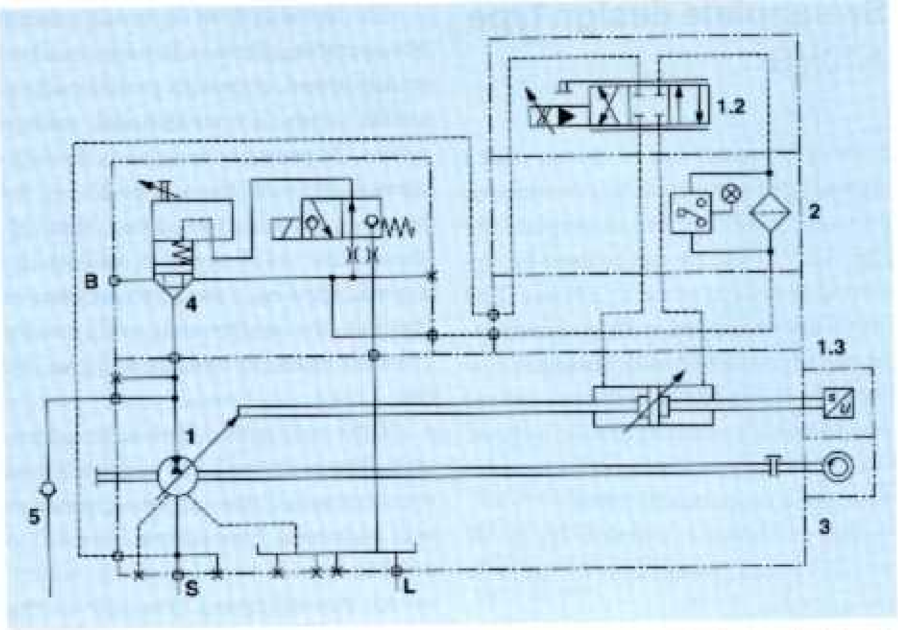


Fig. 18: Circuit diagram for an axial piston unit for secondary control with „hydraulic isolator"

Displacement in cm³	Control time in ms	Pilot oil flow in L/min
40	30	12
71	40	16
125	50	23
180	50	23
250	60	36
355	60	36
500	80	48
750	90	70
1000	100	77

Table 1: Minimum control times and associated flow requirement of A4VS series

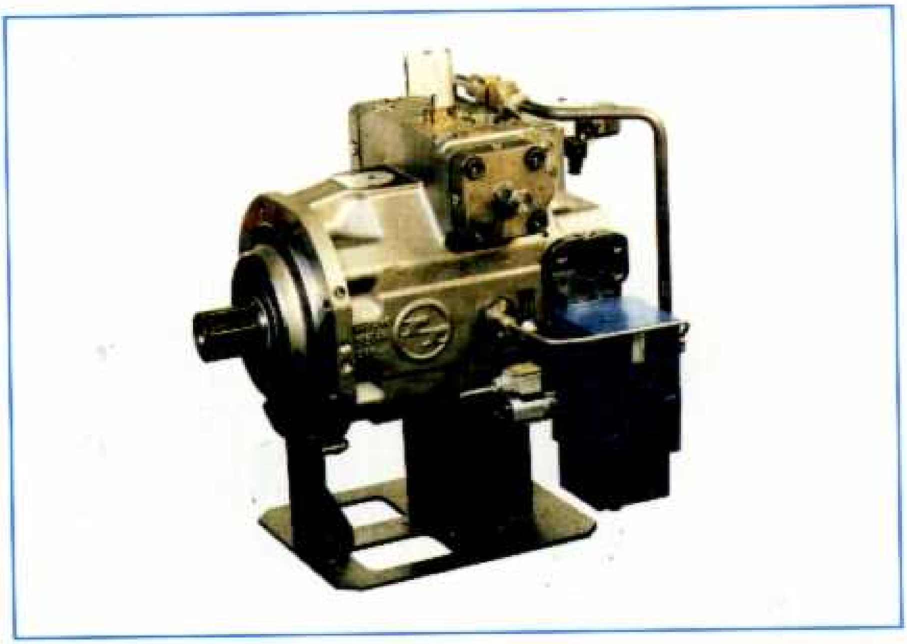


Fig. 19: Secondary unit A4VS500DS1 with „hydraulic isolator"

the primary side to the secondary unit. Unlike the conventional hydrostatic drive, where the overload protection remains safeguarded via pressure relief valves or flow control valves situated in the energy line, with secondary control the secondary unit is connected directly to the energy side. There is thus a virtually unlimited supply of energy. In the event of a breakdown e.g. control system failure, the drive must be cut off from the energy supply.

In the emergency operating mode the secondary unit can only help by generating a power supply.

The "hydraulic isolator" has another important function. During commissioning at maximum operating pressure and with an interrupted energy supply to the drive unit i.e. at zero speed, the positioning system (servo valve, positioning cylinder) can be optimised without risk.

The complete secondary control drive system comprises the following components, which are tested in the factory as a complete unit (**Fig. 18** and **Fig. 19**):

- Axial piston unit A4VS...DS1, (1)
- 4-way servo valve 4WS2EM10 (1.2) (optional proportional valve),
- Inductive positional transducer IW9 (1.3) for swivel angle feedback (optional integral piston positional transducer),
- Pressure filter DFBH/HC, (2) sandwich plate version,
- Tachometer (digital or analogue), (3)
- "Hydraulic isolator", (4)
- Anti-cavitation valve (if required) (5)
- Closed loop control and monitoring electronics.

Bent axis design

In addition to the A4VS series swashplate version as described it is perfectly feasible for a bent axis design of the A2V or A2P series to be used as a secondary controlled unit. This has frequently occurred in the past, as this design principle offers the least friction for movement of the piston in the cylinder drum and the tank is free from torque.

However, when used in a closed circuit system the bent axis, adjustable over the zero point, has some disadvantages compared with the standard swashplate design. These include:

- mounting of tachometer is complex,
- fluid path is complicated via rotary transmission leadthrough,
- large mass to be moved in swivel operation and
- shaftless transmission with transduced torques.

The swashplate design, which is more economic to produce, has come to be more commonly used in this type of application.

Radial piston design

Radial piston units with an eccentricity variable over the zero point are equally as suitable for use in secondary control, as any ensuing forces only act on one level and the positioning time can thus be further reduced.

Moreover radial piston units have a favourable ratio of displacement to natural moment of inertia, which has a beneficial effect on the system dynamics.

Swashplate design type A10VS

If the A4VS series can be driven with a nominal pressure equal to a continuous pressure of 350 bar, the swashplate design A10VS (**Fig. 20**) can be used for operating pressures of up to 250 bar. This unit is also equipped with an incremental transducer, albeit with analogue output only. It also has an inductive swivel angle feedback and a hydraulic isolator. Control of the positioning cylinder is by means of a proportional valve.

Fig. 21 shows a complete layout of an A10VS axial piston unit for secondary control.

As the A10VS is large-scale manufactured, this secondary controlled version is available at a reasonable price. It must, however, be mentioned here that the A4VS series offers higher dynamic response and greater precision can be achieved with speed control and positioning. The nominal calculable service life is also longer with the A4VS.

The fact that with secondary control the operating pressure remains constant or impressed, even with partial loading, unlike with a conventional system where the pressure drops, has no detrimental effect on the service life of the bearings. It must be noted that, although the A4VS series is usually designed based on a standard pressure of 280 bar, the system frequently attains the maximum permissible value of 350 bar.

If the torque of a hydraulic system with flow coupling is reduced by 50% at constant speed, the operating pressure will be halved, for example from 300 to 150 bar.

As the operating pressure to the power three is included in the calculation of the service life of bearings, the calculated service life would be increased by a factor of 8.

With secondary control with pressure coupling a torque reduction of 50% at constant speed will cause the swivel angle also to be reduced by 50% e.g. from 15° to 7.5°. As we can see from **Fig. 22** the bearing service life carries an ex-

Fig. 20: Axial piston unit A10VS, swashplate design, for secondary control

tension factor of 10. This means that at constant operating pressure the swivel angle with swashplate units has a greater influence on the bearing service life than changes of pressure.

This does not take into account the fact that the hydrostatic bearings on the guide shoe, control plate and swivel cradle, which have only a limited service life, are always pressurised by impressed operating pressure with a swivel angle-dependent load. The thickness of lubricating film, which is dependent on the flow throttled by the bearings, remains mainly constant.

The service life of a secondary controlled drive therefore depends to a great extent on the cleanliness of the operating medium.

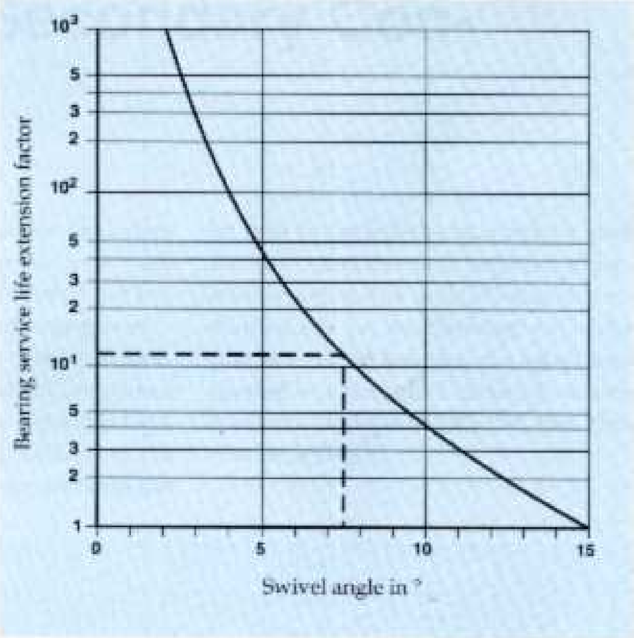


Fig. 22: Influence of swivel angle on the service life of swashplate drives

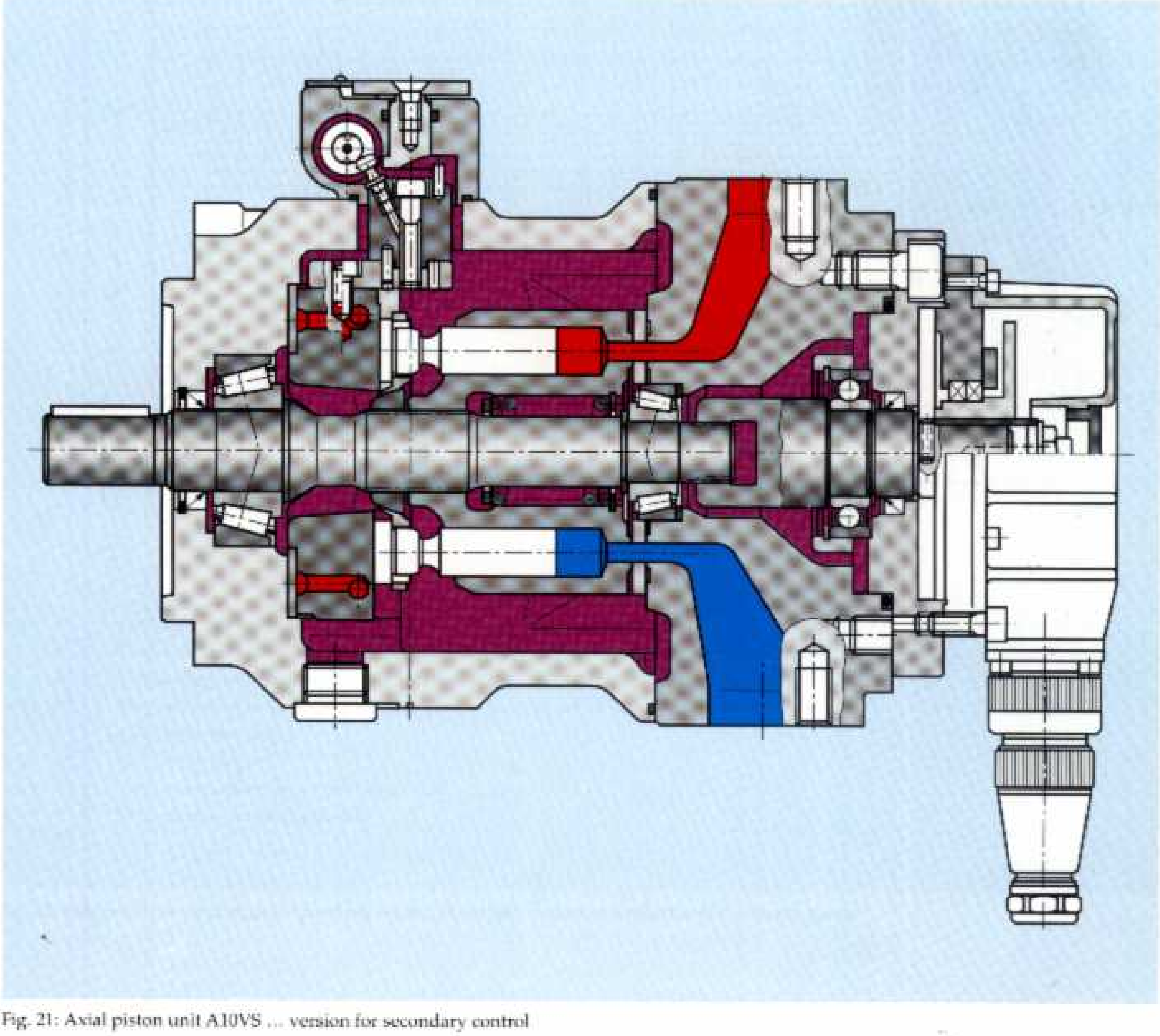


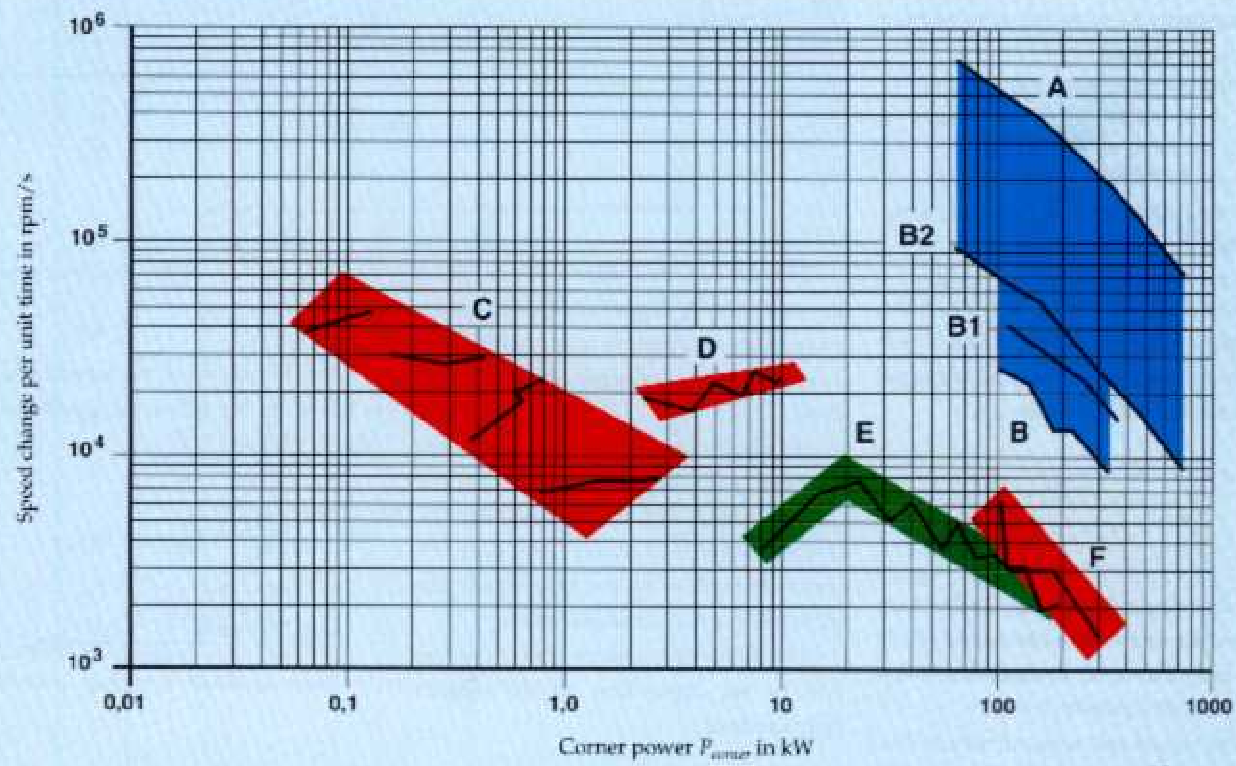
Fig. 21: Axial piston unit A10VS ... version for secondary control

Dynamic Response of Secondary Controlled Drives

Comparison of electric motors and secondary controlled units

Fig. 23 shows a comparison of various types of controllable electric motors with hydrostatic units under secondary control, with respect to the maximum possible rate of change of speed per second as a function of the corner power.

The actual values at present achievable with secondary control are shown in the ranges B/B2 (on a double logarithmic scale). Looking back over the last ten years an enormous increase can be seen in controllable acceleration and the associated dynamic response of secondary controlled units. This is dependent on size i.e. the displacement of the unit, the swivel time and the signal processing in the control circuit. It works, however, independently of the "hydraulic spring".



A	Hydraulic motor, possible theoretical value
B	Hydraulic motor, secondary speed control with axial piston unit type A4VEL, 1984
B1	Hydraulic motor, secondary speed control with axial piston unit type A4VS, 1987
B2	Hydraulic motor, secondary speed control with axial piston unit type A4VS, 1990
C	D.C. servo motors
D	A.C. servo motors
E	A.C. motors, frequency controlled
F	D.C. motors, externally cooled

Fig. 23: Maximum rate of change of speed per second of various motors as a function of the corner power

As can be seen from the graph in **Fig. 24** the speed of a secondary controlled unit with a displacement of 250 cm^3 changes at a rate of 463 rpm in 10 msecs. This represents a speed change of 46,300 rpm/sec or 4850 rad/sec^2. The maximum value theoretically obtainable with negligible loss is 5100 rad/sec. In the last ten years the rate of speed change has been increased five-fold, still keeping to within the limits of technical possibility.

The test measurements were carried out under the following conditions:

Pressure differential:
Δp = 250 bar

Boost pressure:
p_{boost} = 5 bar

Displacement:
$V_{g2\,max}$ = 250 cm^3

Minimum swivel time:
t_{swivel} = 35 msecs

Natural moment of inertia:
J_{nat} = 0.0959 kgm^2

Additional moment of inertia (coupling):
J_{add} = 0.0991 kgm^2

Under the specified conditions, if the operating pressure was increased to 300 bar the system would become unstable in accordance with the formula

$$\dot n_{2max} = 62000 \bullet \frac{rpm}{s}$$

We can also see from **Fig. 23** that further developments must be made in order to push curve B2 towards the theoretical limit of curve A.

There are several ways in which this can be achieved:

- by producing a system-oriented rotary group with the object of reducing the control times of the units.
- by improving the electronic signal processing in the tachometer, swivel angle feedback and at the servo valve.
- by developing advanced digital closed loop control concepts with specially adapted algorithms for secondary control.

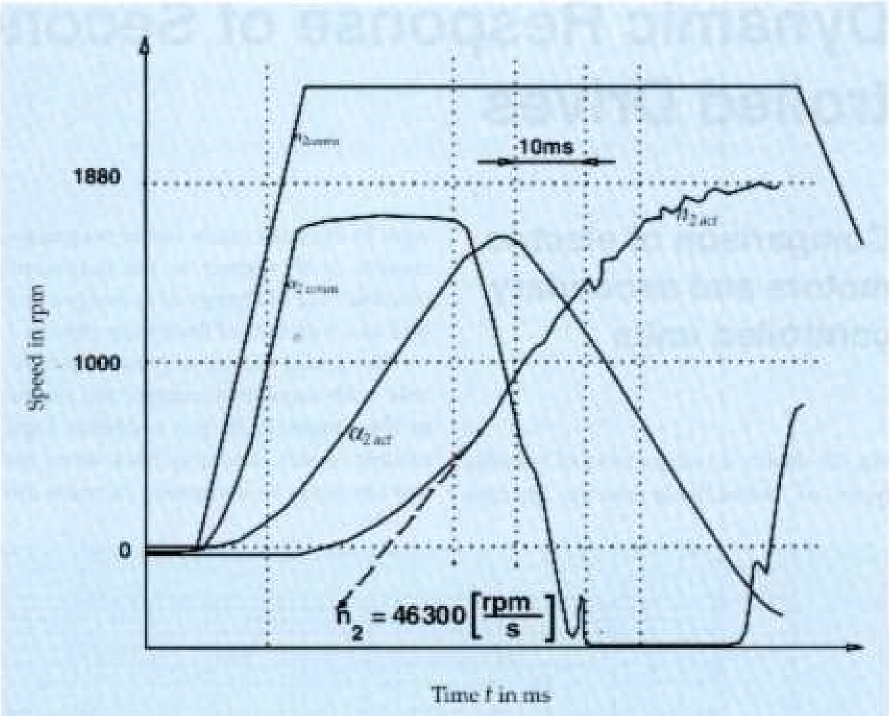


Fig. 24: Determination of acceleration of an A4VSO250DS1 unit

The aim of this development must be an adaptive digital controller which recognises changes in parameters and automatically optimises itself to suit these.

If displacement V_{g2} of a motor changes with impressed pressure the output torque M_{T2} will also be affected. If the torque and swivel angle α_2 increase then flow to the unit must also increase, even at constant speed.

The following rate of speed change will be set depending on the system characteristic:

$$\dot n_2 = \frac{60}{2\pi \bullet J_g} \bullet \left(\frac{\Delta p \bullet V_{g2max}}{20\pi} \bullet \frac{\alpha_2}{\alpha_{2max}} - M_r - M_1\right)$$

A decisive factor in the quality of a secondary controlled unit is the control time t_{swivel} which, according to definition, is the time from 0 to α_{2max} in seconds or milliseconds.

Fig. 25 shows the influence of the control time and the reflected moment of inertia of the driven axis on the speed variation for a jump in torque from light running to 70% of the maximum possible torque.

The derivation of the formula for Δ_{n2} is based on the following mathematical relationships:

$$t = X \bullet t_{swiv} + t_{delay}$$

$$\Delta n_2 = \int_0^t \frac{M_1(t) - M_{d2}(t)}{2 \cdot \pi \cdot J_g} 60 \bullet dt$$

$$X = \frac{M_1}{M_{d2max}}$$

$$n_2 = \frac{M_1(t) - M_{d2}(t)}{2 \cdot \pi \cdot J_g} 60$$

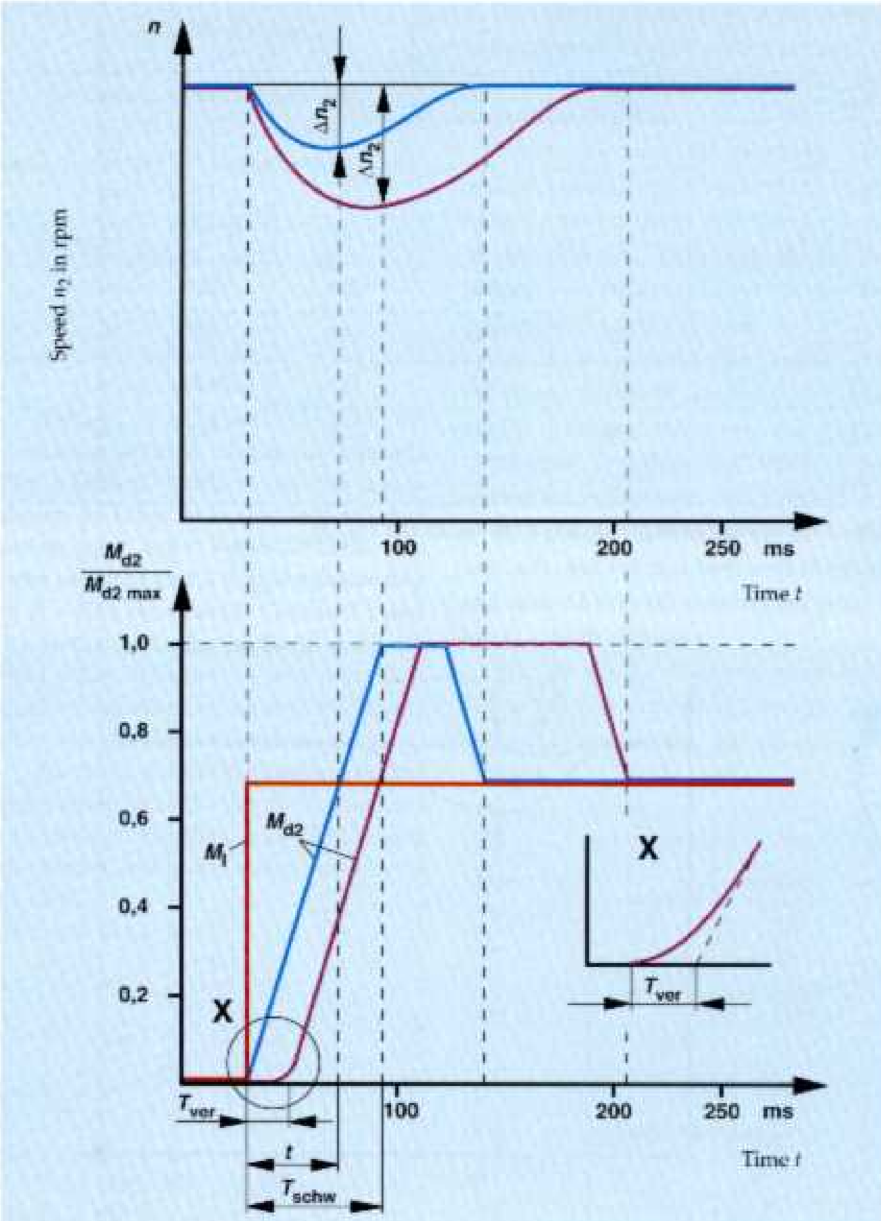


Fig. 25: Min. amount of maximum speed variation with load step

where

I_g = Reflected moment of inertia in kgm^2,
M_2 = Torque of secondary unit in Nm,
M_{2max} = Max. torque of secondary unit in Nm,
M_L = Load torque in Nm,
M_F = Frictional torque in Nm,
n_2 = Acceleration in rpm/sec,
Δ_p = Pressure differential in bar,
α_2 = Swivel angle of secondary unit in °,
α_{2max} = Max. swivel angle of secondary unit in °,
t = Time in sec,
t_{swivel} = Control time in sec,
t_{delay} = Time delay in sec,
$V_{g2\,max}$ = Max. displacement of secondary unit in cm^3,
x = Torque ratio

As soon as the change in load torque is recognised, and without a servo valve in the control circuit, the unit swivels immediately towards maximum displacement dependent only upon the swivel time (T_{swivel}). As the hydraulic torque, which is dependent on the swivel angle, is smaller than the load torque, the speed falls until both torques are equal at X = 0.7. A further increase of the hydraulic torque then causes the drive to accelerate again until the original speed is achieved. The hydraulic torque must now be reduced until it is equal to the load torque as otherwise the speed would continue to increase.

Due to the influence of the servo valve the unit swivels with a time delay. This is dependent on the natural frequency of the valve.

$$t_{delay} = \frac{1}{2 \bullet \pi \bullet f_e}$$

The speed variation is thus greater and the steady state is only achieved after a short time delay.

Assuming that the electronic closed loop control is of optimum design (with a gain of $K_{Rn} \geq 100$ in the speed control loop), the speed variation Δn_2 can be calculated as follows:

- without the time delay of the servo valve:

$$\Delta n_2 = \frac{3X^2 \bullet \Delta p \bullet V_{g2max}}{4 \bullet \pi^2 \bullet I_g} \bullet t_{swiv} in rpm$$

- with time delay of the servo valve:

$$\Delta n_2 = \frac{3X^2 \bullet \Delta p \bullet V_{g2max}}{4 \bullet \pi^2 \bullet I_g} \bullet t_{swiv} \bullet$$
$$\bullet \left(1 + \frac{2 \bullet t_{delay}}{X \bullet t_{swivel}}\right) in rpm$$

where:

f_e = Natural frequency in Hz,
I_g = Reflected moment of inertia in kgm^2,
M_{T2} = Torque of secondary unit in Nm,
$M_{T2\,max}$ = Max. torque of secondary unit in Nm,
Δn_2 = Speed deviation of secondary unit in rpm,
Δp = Pressure differential in bar,
t_{swiv} = Control time in s,
t_{delay} = Delay time in s,
$V_{g2\,max}$ = Max. stroke of secondary unit in cm^3,
X = Torque relationship.

As can be seen from the formula for Δn_2, the greater the reflected moment of inertia, the smaller the fall in speed, a fact assisted by the large natural moment of inertia of the electric motor. However, the crucial value is the swivel time T_{swivel} which determines the rate of torque build-up. Although the electric motor is capable of increasing the torque within the air gap of the motor within 15 - 20 msecs, this has no influence on the dynamic response of the unit due to its high natural moment of inertia.

Step functions, such as we have shown here in **Fig. 25** for the load change, are frequently used in simulation calculations as they can be clearly defined. In practice, however, they do not exist, as the acceleration must then be infinitely great. The load torque actually rises along a ramp as illustrated in **Fig. 26**. As the rise in torque occurs more quickly than the unit can swivel out, a fall in speed is inevitable, but this is much less than is shown in **Fig. 25**.

Speed variation without the time delay of the servo valve:

$$\Delta n_2 = \frac{3X^2 \bullet \Delta p \bullet V_{g2max}}{4 \bullet \pi^2 \bullet J_g} \bullet t_{swiv} \bullet \left(1 - \frac{t_1}{X \bullet t_{swiv}}\right) \text{ in rpm}$$

Speed variation with time delay of the servo valve:

$$\Delta n_2 = \frac{3X^2 \bullet \Delta p \bullet V_{g2max}}{4 \bullet \pi^2 \bullet J_g} \bullet t_{swiv} \bullet \left(1 - \frac{t_1}{X \bullet t_{swiv}} + \frac{2 \bullet t_{delay}}{X \bullet t_{swiv}}\right) \text{ in rpm}$$

	1	2	3	4	
t_{swiv}	60	60	60	60	ms
t_1	–	–	40	40	ms
t_{delay}	–	3,5	–	3,5	ms
$J_g = 10 \bullet J_r$	1	1	1	1	kgm²
Δp	250	250	250	250	bar
$M_{d2\,max}$	995	995	995	995	Nm
X	0,7	0,7	0,7	0,7	
$V_{g2\,max}$	250	250	250	250	cm³
Δn_2	140	163	7	30	min⁻¹

Table 2: Calculation example for Δn_2 from Fig. 25 and Fig. 26

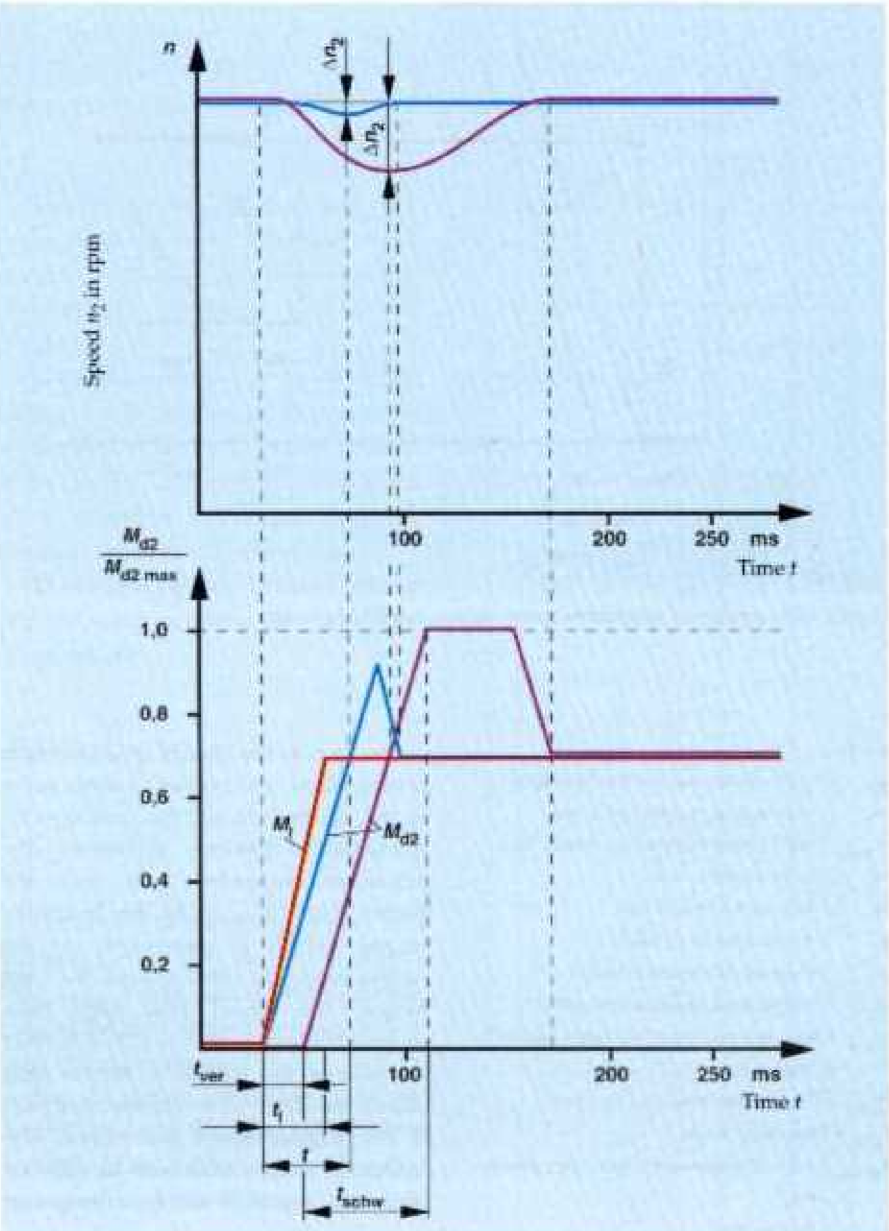


Fig. 26: Speed deviation with load torque ramp

where
I_s = Reflected moment of inertia in kgm²,
I_2 = Natural moment of inertia of secondary unit in kgm²,
Δn_2 = Speed deviation of secondary unit in rpm,
Δp = Pressure differential in bar,
t_1 = Ramp time of load torque in s,
t_{stell} = Control time in s,
t_{ver} = Delay time in s,
$V_{g2\,max}$ = Max. stroke in cm³,
X = Torque relationship.

Utilising the above formula for speed variation a numerical example has been calculated and the results shown in **Table 2**. In this case an axial piston unit type A4VSO250DS1 has been installed and a torque relationship of $X = 70\%$ considered. Columns 1 and 2 cover a load step without and with time delay. Columns 3 and 4 show a load torque ramp without and with the time delay of the servo valve.

The drive shown includes a reflected equivalent mass of 10 times the moment of inertia of the drive unit. For reasons of stability this is the value that should be used.
The time delay is:

$$t_{delay} = \frac{1}{2 \bullet \pi \bullet f_e} = \frac{1}{2 \bullet \pi \bullet 45} = 3,5 \text{ ms}$$

where
f_e = Natural frequency in Hz,
t_{delay} = Delay time in s.

The calculation also shows the influence of the natural frequency of the servo valve referred to the time delay. It becomes obvious that an increase in natural frequency by means of better design has a greater influence on speed accuracy than any improvement in the mechanical area.

The dynamic characteristic of an axial piston unit on the secondary control is determined to a great extent by the dynamic response of the swivel angle positioning device, as defined by control time $T_{stellzeit}$. The dynamics of the overall control loop and the regulator transfer characteristics are also important. These theoretical processes as shown in **Fig. 26** were put into practice using a development test rig with a stepped load change (**Fig. 27**).

To this purpose two axial piston units type A4VS with a displacement of 250 cm³ were used, tensioned against each other (**Fig. 28**).

Secondary unit (2) is speed controlled. Loading of the system can be infinitely varied with the torque controlled unit (4). With this circuit the primary unit needs only to cover the hydraulic-mechanical and volumetric losses of all hydraulic components and the power losses in mechanical section 3. The primary power in this circuit constitutes less than one third of the secondary unit power.

A pressure difference of Δp = 245 was selected for this test, and the control time of unit (4) determined from 0 to $\alpha_{2\,max}$ with 47 msecs.

Fig. 27 shows a torque jump from 570 to 0 Nm at a speed of 1000 rpm. The torque generated by the torque controlled unit (4) acts like a motor on the speed controlled secondary unit (2). In order to sustain the pre-set speed of 1000 rpm the secondary unit (2) must work as a hydraulic pump. This way it works against the motor-driven torque of unit (4).

With a torque jump to M_{T2} = 0 Nm from unit (4) there is no driving moment, so therefore the speed controlled secondary unit (2) must change over to motor operation in order to maintain the pre-set speed of 1000 rpm. During this torque jump a speed deviation Δn_2 = -150 rpm will occur, which will, however, be brought under control without overshooting (T = 1.7 secs).

This test was then carried out with an additional reflected moment of inertia of 1.9 kgm², the torque control time being 250 msecs.

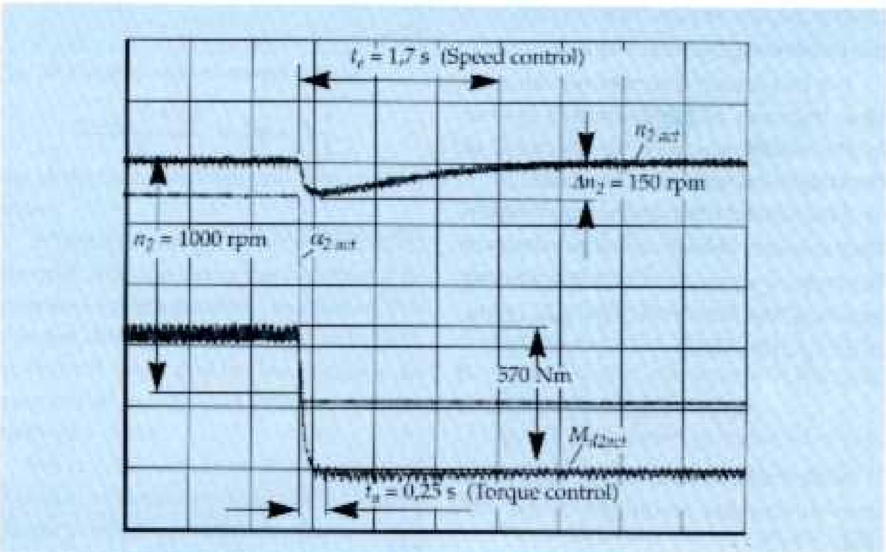


Fig. 27: Speed response of secondary unit A4VSO71DS1

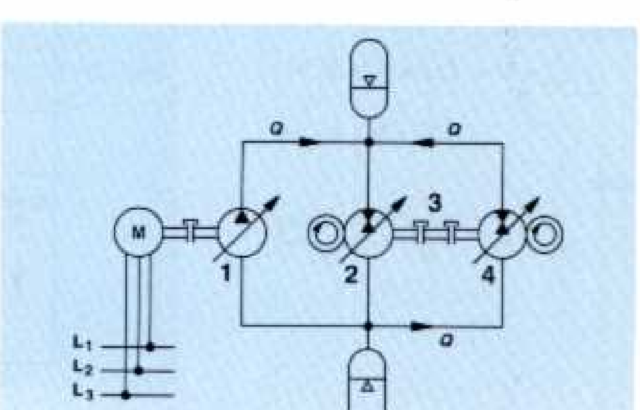


Fig. 28: Basic diagram of test circuit

Fig. 29 shows the reaction of the test equipment to a speed command value jump from 400 rpm to 1050 rpm for a load of 500 Nm.

The increase in speed effects a reduction of torque to 300 Nm during a length of time 400 msecs (T = 400 msecs).

This reduction in torque is caused by the fact that during the acceleration process with the secondary unit (2) working as a generator, the torque must be reduced in order for the acceleration torque to be sustained. This torque difference is recorded by the measurement shaft. The torque of the load unit (4) is not affected by this process.

After a time of 1.4 secs the required speed will be attained, again without overshooting.

Due to the special importance of dynamic response with secondary control, the user needs the parameters which determine the system characteristics in the project engineering stage.

For this reason a correction value for time response of the controlled system is defined, the so-called time factor T_r of the controlled system in seconds.

This time factor indicates different transmission behaviour depending on the type of application and is found by equating the kinetic energy E_{kin} of the rotating drive shaft

$$E_{kin} = \frac{1}{2} J_g \bullet \omega_2^2$$

with the hydraulic energy E_{hyd} to be used during one revolution of the secondary unit.

$$E_{hyd} = \int P_{hyd} \bullet dt$$

t_r characterises the dynamic response of the controlled system and secondary unit.

$$\int P_{hyd} \bullet dt = \int \Delta p \bullet Q_2 \bullet dt =$$

$$= \int \Delta p \bullet V_{g2} \bullet n_2 \bullet dt$$

with

$$\omega_2 = \dot{\varphi}_2 = \frac{d\varphi_2}{dt} = \frac{2 \bullet \pi \bullet n_2}{60}$$

gives

$$\int P_{hyd} \bullet dt = \frac{1}{20 \bullet \pi} \int \Delta p \bullet V_{g2} \bullet \omega_2 \bullet \ dt$$

If we assume that Δp and V_{g2} are constant and $V_{g2} = V_{g2}$ max, it follows that

$$\int P_{hyd} \bullet dt = \frac{1}{20 \bullet \pi} \Delta p \bullet V_{g2max} \bullet \int \ d\varphi_2$$

Referred to one revolution we obtain:

$$\int_0^{2\pi} d\varphi_2 = 2 \bullet \pi$$

From this it follows that:

$$\int_0^{2\pi} P_{hyd} \bullet dt = \frac{\Delta p \bullet V_{g2max}}{10}$$

Thus:

$$\frac{1}{2} J_g \bullet \omega_2^2 = \frac{\Delta p \bullet V_{g2max}}{10}$$

For a length of time

$$t_r = \frac{1}{f} = \frac{2 \bullet \pi}{\omega_2}$$

$$\omega_2 = \frac{2\pi}{t}$$

the time factor of the controlled system will be

$$t_r = 2 \bullet \pi \sqrt{\frac{5 \bullet J_g}{\Delta p \bullet V_{g2max}}}$$

where

E_{hyd}	=	Hydraulic energy in Nm,
E_{kin}	=	Kinetic energy of rotating drive in Nm,
f	=	Frequency in 1/s,
J_g	=	Reflected moment of inertia in kgm^2,
P_{hyd}	=	Hydraulic power in watts,
Δp	=	Pressure differential in bar ,
Q_2	=	Flow in cm^3/s,
t	=	Time in s,
t_r	=	Time factor of control area in s,
V_{g2}	=	Stroke in cm^3,
ω_2	=	Angular velocity 1/s.

With the swivel time values of the control system T_{swivel} and time factor of the controlled system T_r the characteristic for the transmission behaviour of the secondary unit can be determined. These apply to a complete series of axial piston units, and are virtually independent of the signal processing in the control circuit.

The controlled system is considered to be friction-free. It has integral transmission behaviour with the integration constant

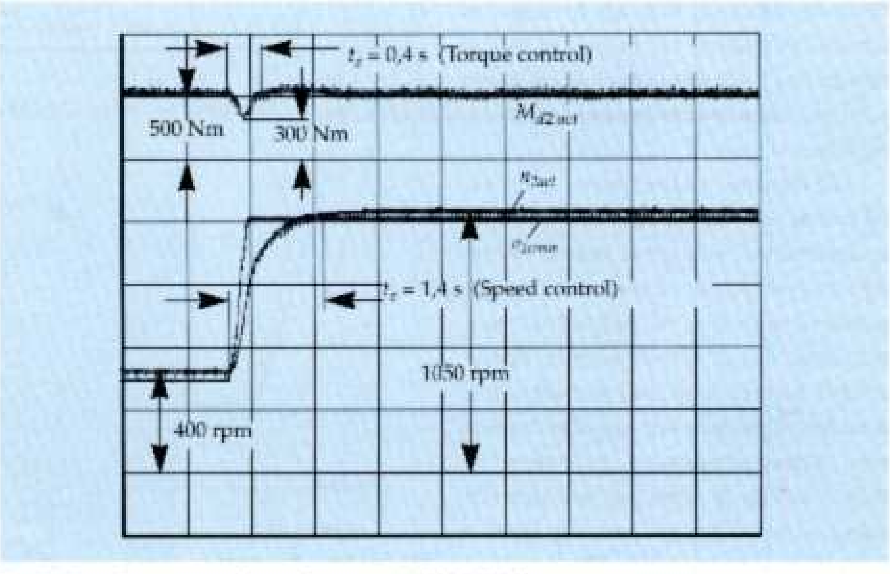


Fig. 29: Speed response of secondary unit A4VSO71DS1

$$K_a = \frac{1}{20 \cdot \pi \cdot J_g}$$

This integration constant depends solely on the reflected moment of inertia on the controlled axis. This instance of an integrated controlled system is the least favourable option with regard to system stability.

For the given output torque M_{T2} the resulting rotational acceleration $\dot{n}_2$ will be greater, the faster the response of the control circuit and the smaller the swivel time T_{swivel}.

The time factor of the controlled system is proportional to the moment of

$$t_r = 2 \cdot \pi \sqrt{\frac{5 \cdot J_g}{\Delta p \cdot V_{g2max}}}$$

inertia and is a parameter of the drive which eliminates the influence of the size factor i.e. the dynamic response of a complete series of units may be determined from this single stability diagram (**Fig. 30**).

The area below the curves represents the stability zones. Above the curves instability can be expected.

The user is thus in possession of all the parameters which determine the system characteristics in the project engineering stage. If the practically achievable and pre-determined control time T_{swivel} is greater than the curve for the pre-determined limiting positioning time, the secondary speed control of a given series operating under conditions of impressed pressure can only achieve the pre-selected command speed after a period of oscillation. The greater the difference between the required and practical values of control time, the worse the situation will become. If the practically achievable control time is less than the limiting control time, the speed control will reach the set command speed under light running conditions without overshoot, and will have a certain amount of dynamic reserve. The value of this reserve will depend on the difference between the two control times.

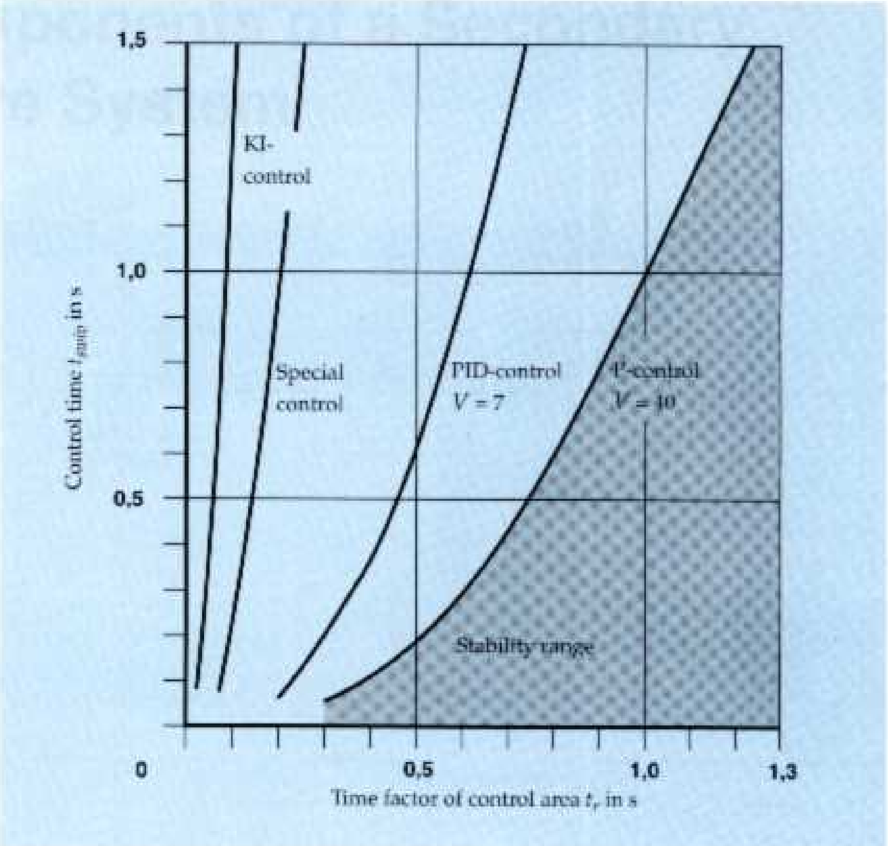


Fig. 30: Stability curves control setting

When designing a system, it must be ensured that the drive lies below the illustrated limiting curves. In view of this it is not difficult to see that a reduction in control time and/or an increase in moment of inertia will increase the stability of a drive.

We can also see from the curves that if a better regulator with PID transfer characteristics is used instead of a P regulator, the limiting control time can be considerably offset. This means that with the same time factor T_r, and given that the pressure difference and displacement remain constant, the control time T_{swivel} can be greater, or the same control time can be used with a lower moment of inertia.

In conclusion it can be said that, with reducing moment of inertia the dynamics of the speed control loop must be increased.

In practice, this can often occur where, for example, the reflected inertia is reduced due to the square of the gear reduction coming into play.

As all problems are reduced when the control times are reduced, in conclusion we can say:

No matter how short, control time will always be too long.

Electronic Components of a Secondary Controlled Drive System

Introduction

Unlike conventional hydrostatic drives, with secondary control the actuator influences the swivel angle and hence the displacement. The speed actual value from the tachometer is compared with the command value and this then controls the servo valve of the control system via the controller. Any variation in speed will result in a change in displacement and thus also a change in torque by means of the actuator of the secondary unit. The controller output value will be approximately equivalent to the acceleration minus the load torque.

As long as the basic natural frequency is well above that of the drive, the system will consist of two integrators in series, normally renowned as being extremely difficult to stabilise.

The design of the control system as in **Fig. 31** is characterised by an integral time delay function (I-T4). The positional feedback of the swivel angle actual value for forming a subordinate swivel angle circuit has enabled the integral characteristic to be converted into proportional characteristic with time-delay (P-T5).

As the swivel angle position can be maintained, this markedly improves the control characteristic of the secondary unit. It may be useful in extreme cases to add a second differentiator for stabilisation purposes. This could be for example the 2nd derivation of the speed

$$\frac{d^2n}{dt^2}$$

the "jerk" or actual value signal acceleration.

This double differentiation of speed is, however, seldom used in practice, as it has no significant improvement on the control characteristic. Not only the position but also the speed dy/dt of the positioning cylinder is fed back in these cases. This is identical to the acceleration in speed i.e. the 2nd derivation of the speed.

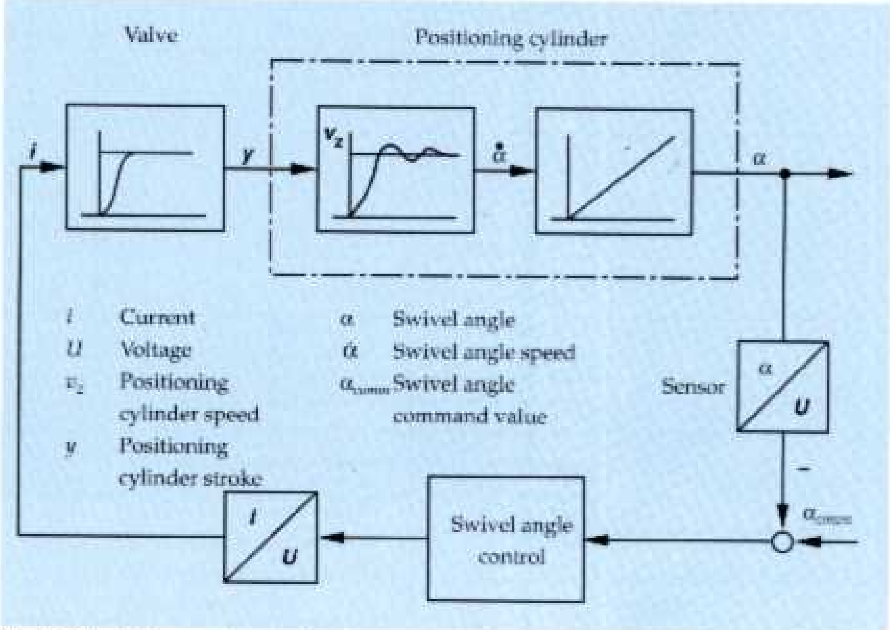


Fig. 31: Block diagram of swivel angle control circuit

Speed control

Referred to the swivel angle, the speed has a proportional time delay characteristic (P-T1) (**Fig. 32**).

If the secondary unit lies at the lower end of the speed range and is driven with low friction, it has an almost integral characteristic.

Thus for a constant speed under stable conditions, relatively small changes in disturbance torque Z will cause acceleration or deceleration to occur. At a constant swivel angle, this will lead to relatively large speed variations, even to the extent of stopping the unit or overspeed before levelling out. The swivel angle must therefore be continually corrected in order to achieve a constant speed.

The main application area of secondary control is in the speed control circuit with free torque value i.e. the torque is set that is required to maintain the preset speed.

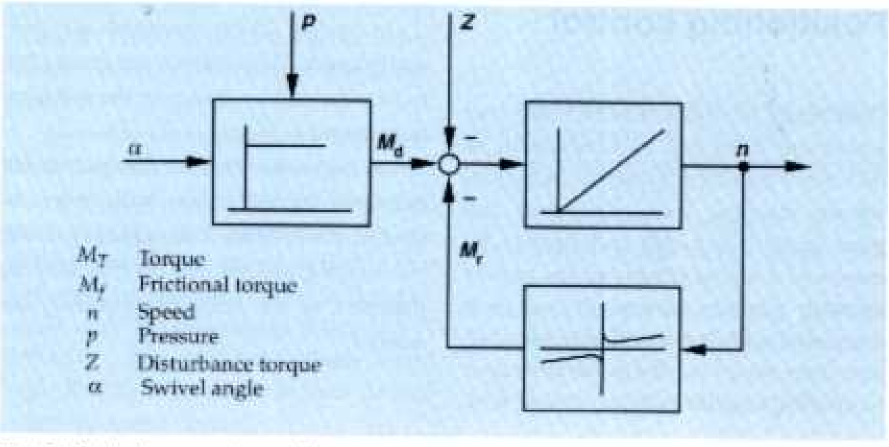


Fig. 32: Block diagram of speed characteristic

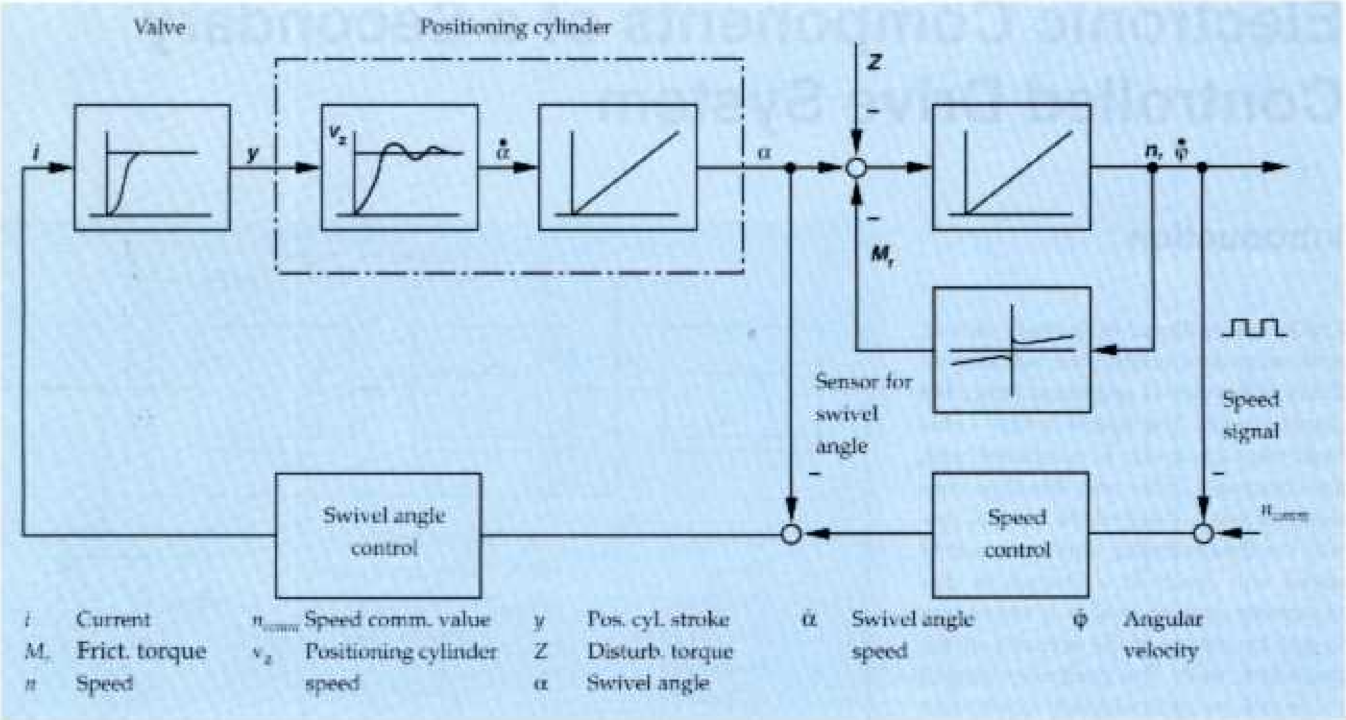


Fig. 33: Structure of speed control circuit

The block diagram is designed as in **Fig. 33**.

Under the influence of load and frictional torques, at constant speed the secondary unit requires a swivel angle that is variable across the zero position.

A sustained speed deviation must therefore be reckoned with when using a speed controller with proportional characteristics. This deviation will depend on the size of the countertorques and the controller amplification.

This control deviation is compensated for by installing a speed controller with proportional integral characteristics (PI controller).

Positioning control

Positioning control is required of drives which are used e.g. in automation as drives for handling systems, industrial robots, machine tools, driverless (remote control) vehicles and similar. By means of a digital controller the pre-set angular position command value is compared with the actual angular position determined via the pulse generator to give the angular position differential.

The advantage of a digital control lies in the high resolution attainable when determining the angular position. This is limited only by the pulse rate of the generator per revolution

- 2500 pulses per revolution give a resolution of 0.144° and hence, referred to one revolution, an accuracy of 0.04%

and by the resolution of the hardware counter module used

- 32 bits give a total of 2^{32} (= 4 294 967 296) pulses. For a processing frequency of 500 Hz and a generator of 2500 pulses per revolution this means a maximum speed of 12000 rpm.

In reality 24-bit counters are normally used. This enables the resolution to be extended by means of the software as required.

For generators of 2 500 pulses for example, 10 000 pulses will be evaluated at the counter. The accuracy of the positioning process can in this way be matched to the relevant operating sequence.

The positioning circuit is designed as shown in **Fig. 34**.

It comprises a micro processing control system with a control algorithm specially tailored to secondary control, where the digital controller transmits directly the command value for the analogue swivel angle.

Utilising the almost integral characteristic speed, with its real integral characteristic for angular position as described earlier, a quasi double integral characteristic is achieved in the positioning control loop. For this reason speed feedback is required for positioning control, achievable by means of a special algorithm in the digital positioning control.

As with speed control there will be a residual control deviation caused by the influence of friction and load torque.

A PI positioning control would cause oscillation due to the angular position integral characteristic and is therefore unsuitable. The residual control deviation is thus compensated for by means of special algorithms matched to the requirements of secondary control.

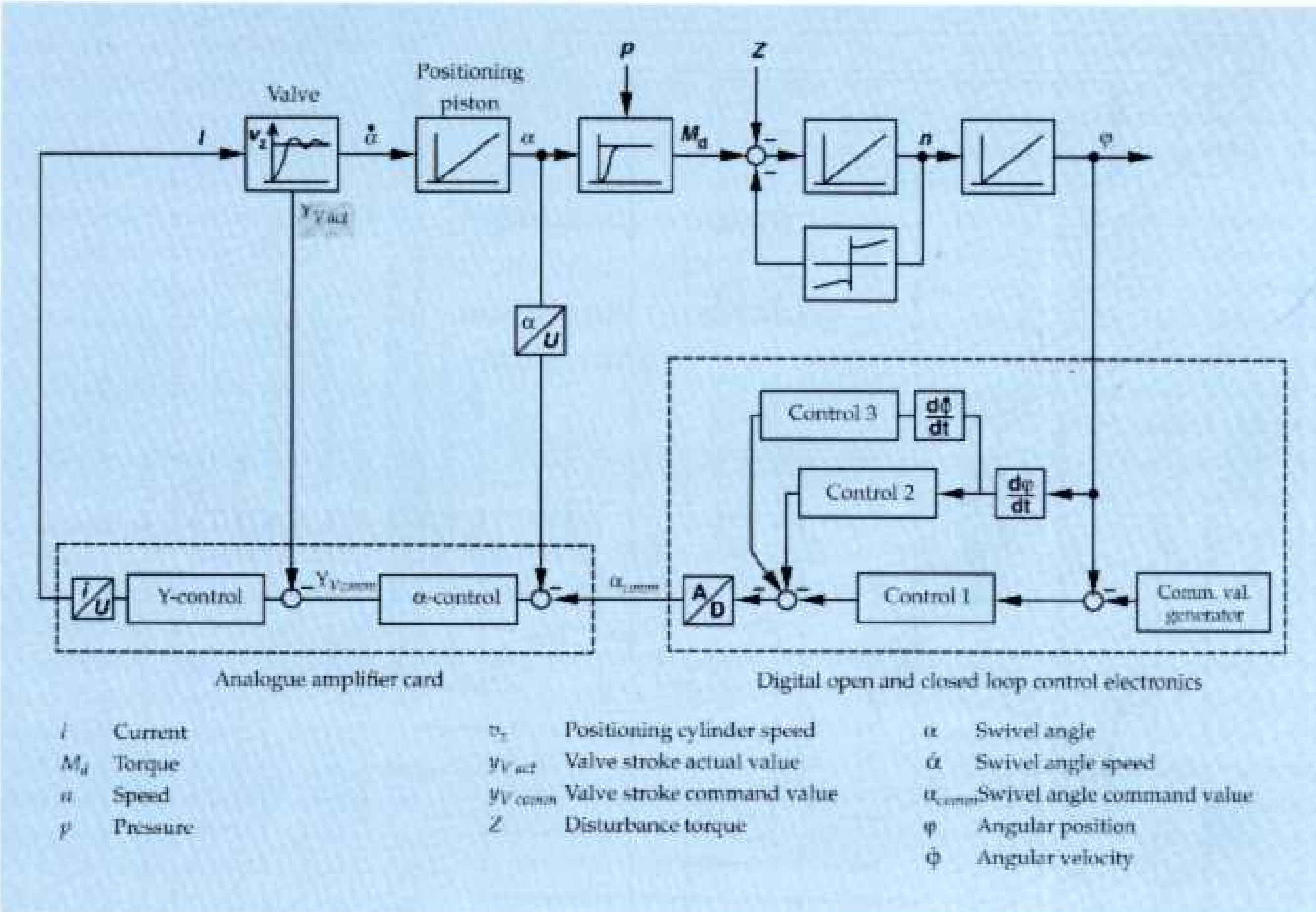


Fig. 34: Positioning circuit with secondary control for proportional valves with positional feedback

Analogue standard controller for series A4VS

In August 1988 the control and monitoring electronics for speed control with secondary controlled axial piston units, the VT 12000, was introduced (**Fig. 35**). It contains a ramp for the speed command value, an analogue speed control with PID characteristics, a likewise subordinate analogue swivel angle control and various monitoring functions. The amplification stage for the servo valve and voltage stabilisation are also integrated into the unit.

At the differential amplifier input (**Fig. 36**) the speed command value will be ± 10 volts. With potentiometer R 201 the speed command value can be reduced to $<n_{max}$. The slew rate of the speed command value can be pre-set from 20 msecs to 5 msecs by means of an attachable ramp. This is set at potentiometer R 203, where the ranges of the speed ramp are selected via jumpers 3 and 4.

The PID speed control, which has parameters that may be changed when the machine is at a standstill, processes the speed difference between command and actual values. The output signal of the following PD swivel angle control delivers the input signal for the servo valve output stage. The output current of the VT12000 controls the servo valve, which positions the swivel angle correctly.

Swivel angle feedback with the standard model is carried out by an inductive positional transducer, whose output signal is generated by a demodulator. The speed actual value delivered by the tachometer is standardised with the resistances R 229 and R 230 to 0 ... ±10 volts and is finely tuned with R 202. This signal is then passed on to the speed control.

Fig. 35: Standard control electronics VT12000

The following functions are assigned to the relay requests K1 to K5:

- K1 switches the speed command value to "ON" or "ZERO".
- K2 switches the speed control to "ON".
- K3 switches the swivel angle command value to "ON".
- K4 switches the speed ramp "OFF" (in operational mode the speed ramp is switched on).
- K5 is used for reversing control.

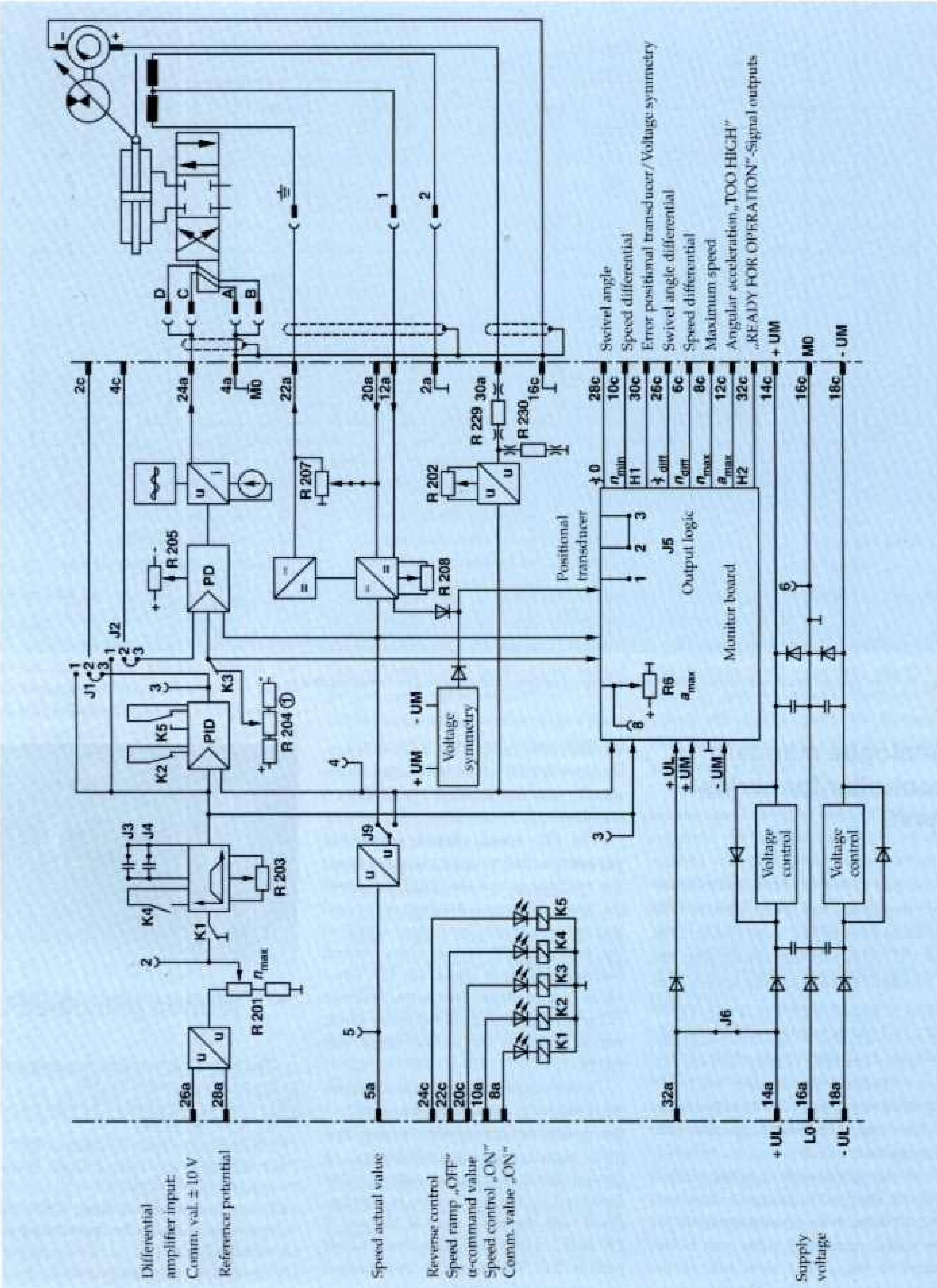


Fig. 36: Block diagram of VT12000 standard control electronics

The monitoring board on the card receives the following signals from the control board (**Fig. 36**):

- actual and command speeds.
  If the speed differential is ≥5% the signal is activated. An error signal will be sent if the speed exceeds the maximum value by 10%.
  An error signal will also be sent if the actual rate of speed increase is faster than the pre-set value.
- swivel angle actual and command values.
  If the swivel angle differential is ≥5% an error signal will be sent.
- Monitoring signal - positional transducer.

In the event of a breakdown in the positional transducer range, caused by a short circuit, cable break or unsymmetry of the stabilising voltage, an error signal will also be sent.

All data is interlinked. It is generated and output in the form of instantaneous switch signals.

If the tachometer with analogue output is replaced by a pulse generator, then the actual speed signal must be matched to ±10 volts by means of a frequency voltage modulator, before it is sent to the VT12000.

Frequency-voltage converter with additional monitoring electronics

The VTS0102 frequency-voltage converter is used together with the VT12000 standard control electronics (**Fig. 37**) and a digital tachometer wherever great demands are made on speed accuracy and operational safety.

On receipt of two pulse sequences the input signals from the evaluation switching operation are divided into frequency and directional signals. By means of a jumper the input frequency may be doubled or increased fourfold. The converter component converts the frequency into a proportional analogue voltage, the directional signal determining the sign of the analogue voltage.

The pre-set conversion ratio can be finely tuned during commissioning by means of a potentiometer in the front plate.

The sign itself can be inverted by means of a jumper. The f/U converter is fitted with a cable break monitoring device which compares the four input signals of the incremental transducer with each other. If there is a breakdown of one or more lines a memory will be set and an LED will respond simultaneously. The input pulse sequences are individually monitored before conversion to enable recognition of overspeeds.

If there is a fault in one channel the overspeed will be recognised by the other channel. The output signal of this speed monitoring will be added to the $n_{2\,max}$ signal of the VT12000, whose analogue voltage is monitored. On recognition of overspeed a memory will be set once more and an LED activated.

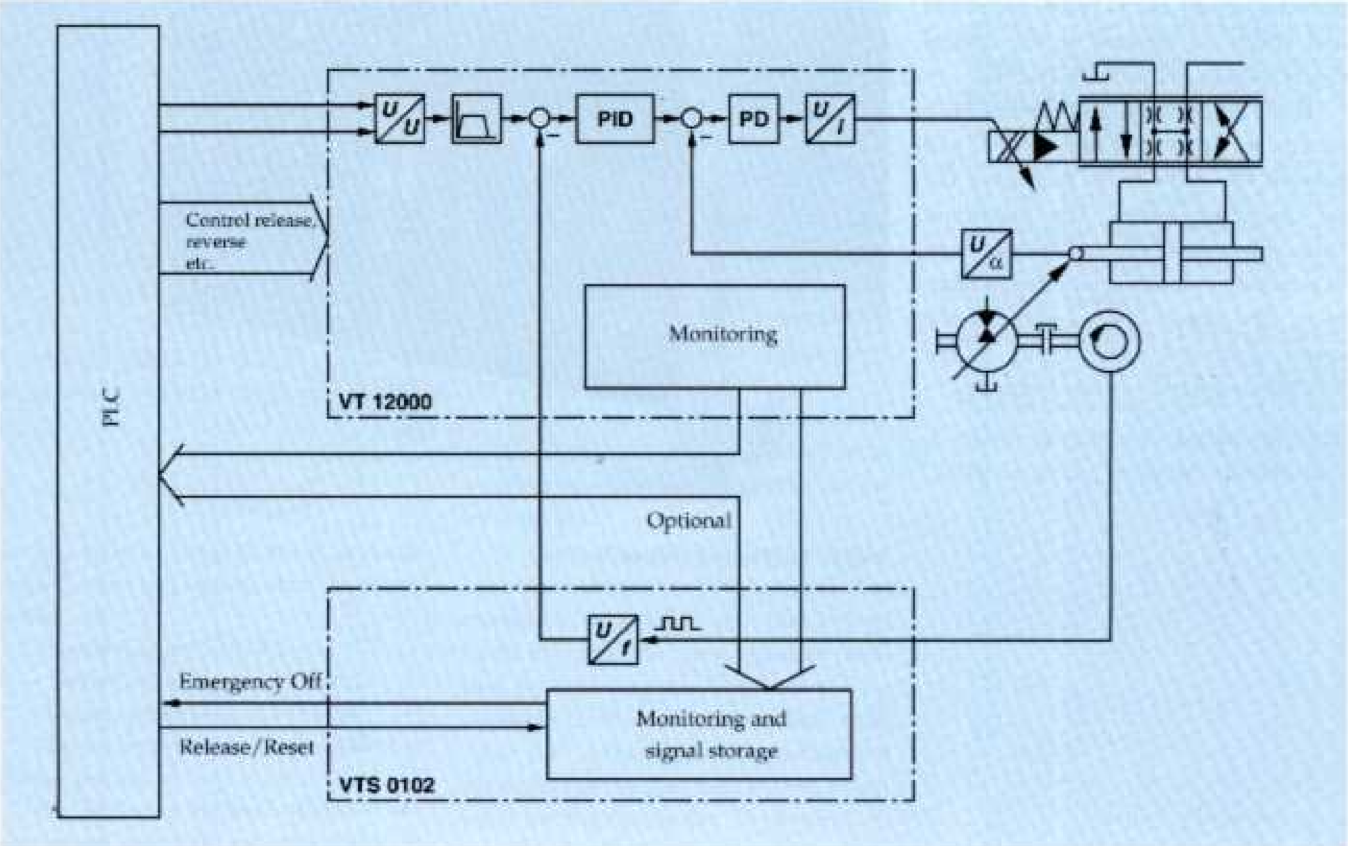


Fig. 37: Signal flow chart of f/U-converter with VT12000

Fig. 38 shows both electronic cards in Eurocard format. The internal monitoring signals of the VT12000 are linked to those of the f/U converter. The VT12000 error messages and the f/U converter monitoring signals are intermediately stored on the electronic card. The error states will remain stored until reset, either by activating a key or via a 24 V signal.

Torque control

The chapter "Special Characteristics of Secondary Control" discusses a type of torque control (page 71) by means of the swivel angle. This torque control card (**Fig. 39**) can, however, as the name says, also be used for torque control if there is a torque gauge. This card works again in conjunction with the VT12000 control and monitoring electronics (**Fig. 40**).

The torque actual value output by the gauge is sent to the PID control on the card. The active correcting function operates as an error value correction and effects an increase in the control characteristics of the control device.

If the monitoring and control electronics are active, the limit switch will prevent the drive from exceeding the set maximum speed. The maximum value is set separately for each direction of rotation via internal or external trimmers. The speed control acts on the speed command value input of the VT12000 standard control electronics.

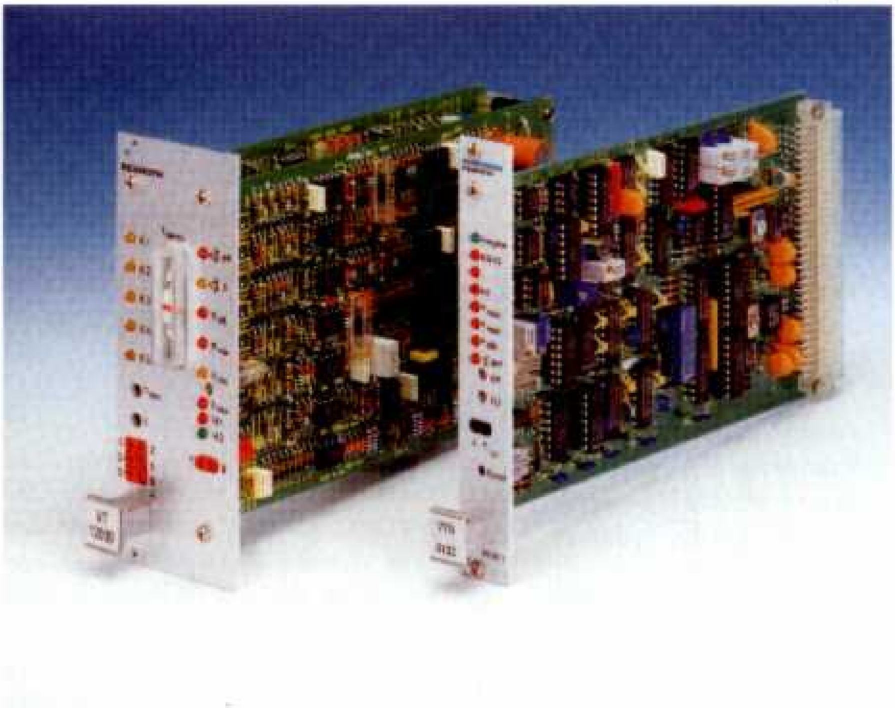


Fig. 38: VT12000 standard control electronics with VTS0102 f/U-converter in Eurocard format

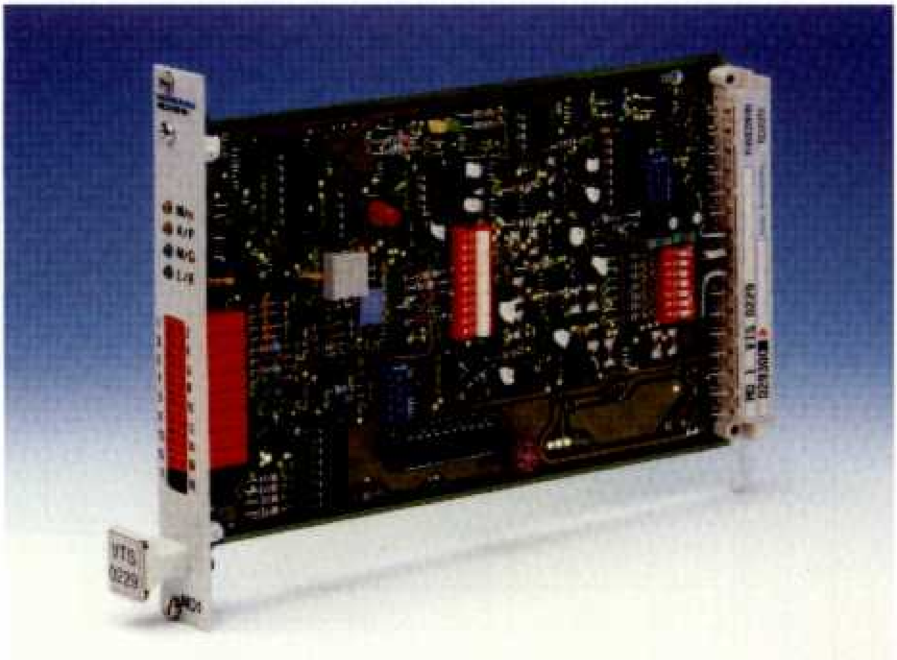


Fig. 39: MD1 torque control card

Power Control

Another important component in the use of secondary control is the LB1 power control card for the power, speed or pressure-dependent limitation of the swivel angle in four quadrant operation (**Fig. 40**). It is used whenever overloading of the powered machine on the primary side is to be avoided i.e. when the primary side power and the installed accumulator capacity are lower than the corner power of the secondary unit.

This operating mode occurs frequently in practice with mobile machinery, whenever several actuators connected in parallel are driven from a primary station. Even with a single actuator, overloading of the powered machine may occur for example during the acceleration phase.

The power of a secondary controlled unit is the product of pressure, displacement and speed

. $P_{hyd} \sim \Delta p \bullet V_{g2} \bullet n_2$.

On reaching the pre-set maximum value, the power of a secondary unit can be limited or even reduced by lowering the speed command value. This process affects the control circuit and may, under certain conditions, lead to instability.

More stable conditions can be expected when the speed command value remains constant and the swivel angle is influenced by a pure control operation, which is seldom unstable.

This maximum permissible swivel angle α_{per} is calculated as follows from the pre-set maximum power P_{limit} at an impressed pressure Δp_{act} and speed n_{2act}:

$$|\alpha_{per}| = K \bullet \frac{P_{limit}}{|\Delta p_{act} \bullet n_{2act}|}.$$

The standardising factor K is determined from the system data at corner power, i.e.:

- Speed n_{max} = 10 volts,
- Swivel angle α_{max} = 10 volts,
- Corner power P_{cor} = 10 volts.

Maximum power is given relative to corner power.

Fig. 41 shows the block diagram of a secondary unit with power control. In practice power control is usually used only in motor operation, as in generator operation i.e. when using a diesel motor, the braking energy has to be converted into heat by means of a pressure relief valve. In this instance only the swivel angle direction will be influenced depending on the direction of rotation. With a deceleration process the total braking torque is available as with positioning. If, when operating a bi-directional pressure controlled primary unit (mooring), the braking power is to be passed on to the mains circuit via the electric motor, power control must be effective in all four quadrants.

When power control is activated the speed control is deactivated, so that when monitoring the control circuit the error message "SPEED VARIATION" may occur. In this case the message "POWER CONTROL ACTIVE" is output. This enables the evaluation logic, for example in a PLC, to suppress the "SPEED VARIATION" error message.

The actual converted power

$$\frac{n_{2act} \bullet \alpha_{2act}}{K} = \frac{P_{hyd}}{\Delta p}$$

at constant operating pressure can be output to a main system as a display for feedback via a second functional group of the power control card. Only values $n_{2\,actual}$ and $\alpha_{2\,actual}$ are pre-set. As $n_{2\,actual}$ input is independent of $n_{2actual}$ for power control, power generation of a torque or swivel angle controlled load unit can be calculated by multiplication and added to the maximum power of a speed controlled drive unit to give extra power to the primary unit.

We are familiar with such operating modes from test rigs and also from winding operations in the paper industry (**Fig. 42**).

If a winding machine has a speed controlled secondary unit coupled mechanically to a load unit, the power from the primary unit together with that generated by the load unit will be available to the speed controlled unit. After the primary power has been set the power of the load unit will be calculated by the multiplier. Depending on whether the load unit is operating as a generator or as a motor, the power limit of the speed controlled secondary unit will either increase or decrease.

Apart from the power display function, the multiplier functional group can be used to control a diesel motor as drive of the primary unit, depending on system status of the secondary unit, in such a way that sufficient power is always available without the diesel motor having to run constantly at maximum speed. This is a way of reducing power loss in the complete system.

The speed command value of the diesel motor is dependent on various factors:

- a switched input with invertible logic, which fixes the basic speed of the diesel motor,
- a pressure switch input, whose response effects maximum output current. It monitors the operating pressure, responding if the pressure drops to below the minimum value.
- the actual power level attained, determined by the product of $n_{2\,actual}$ and $\alpha_{2\,actual}$. If the product of swivel angle and speed increases, the diesel speed will also increase. If it decreases, the diesel speed will also decrease. This function only occurs during the acceleration phase i.e. in motor operation.
- additional influence of output current by the actual speed and actual swivel angle values. Here again the actual swivel angle is effective only in motor operation.
- additional influence of output current via an analogue input from a pressure pick-up, to enable the diesel motor speed to be increased when the operating pressure is decreasing.

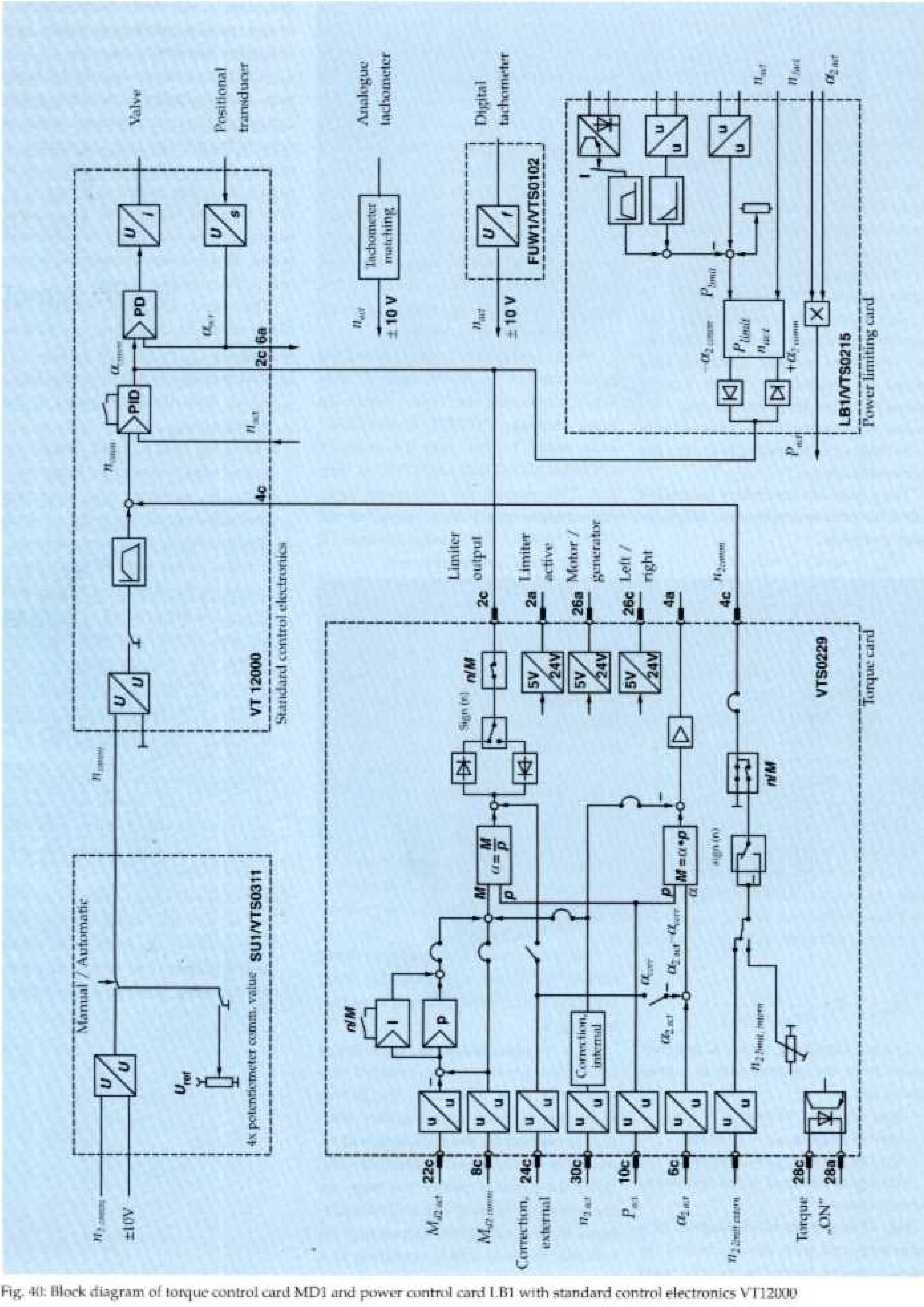


Fig. 40: Block diagram of torque control card MD1 and power control card LB1 with standard control electronics VT12000

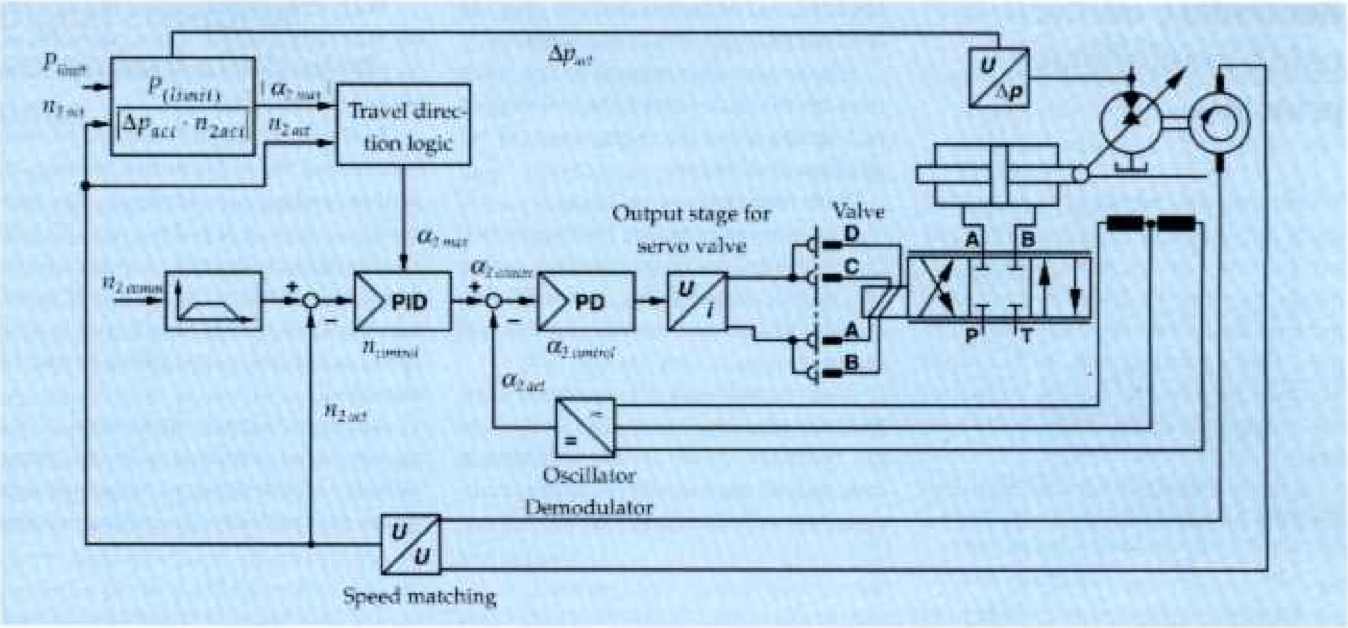


Fig. 41: Block diagram of secondary unit with power control

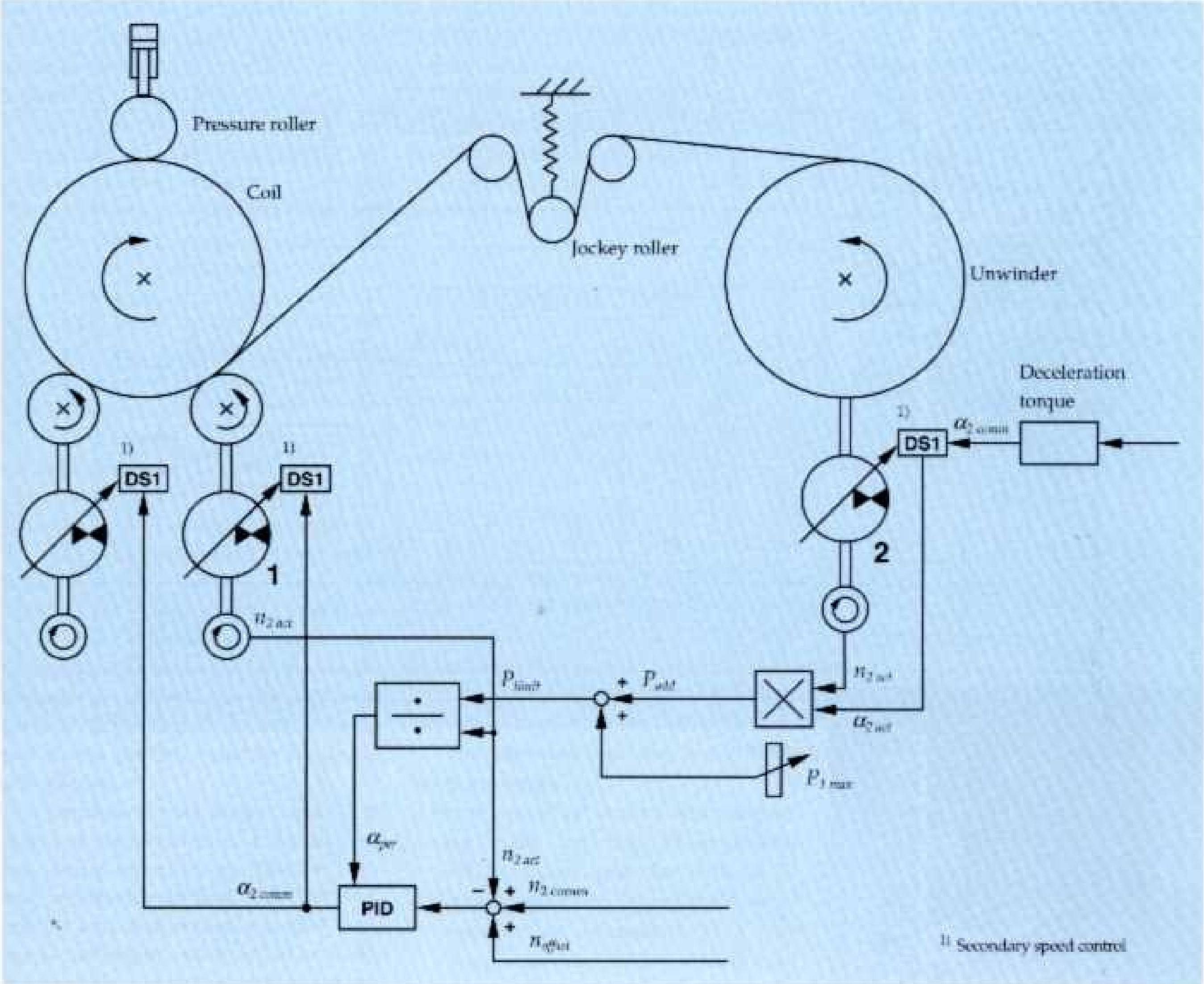


Fig. 42: Secondary control with power control

Secondary control using master/slave principle

Where several secondary controlled units are coupled together either by non-positive or positive locking, it makes no sense to drive the two units independently with separate speed controls. This would result in the I-sections in closed loop control having a separating or opposite-acting effect on the controls.

If the non-positive locking malfunctions the master/slave principle will be applied, as demonstrated in **Fig. 43** with the example of a paper winder.

The paper roll is wound onto a roller (reel-spool) (2), whereby the roll tension and speed must be kept constant regardless of diameter.

Both these values can be set.

The winder is driven by way of two controlled rollers (1) at constant power i.e. if the diameter is increased the speed will decrease and the torque will increase.

The master axis (3) is speed controlled in the usual way. The swivel angle feedback value of this machine is transmitted as the swivel angle command value to the slave axis (4).

With this winding operation we can see that only a single speed control is effective with two subordinate swivel angle control circuits.

In the event of an error and of malfunction of the non-positive locking, or during setting up without the reel-spool, control of the slave axis (7) will be effected temporarily via a speed control with limitation (8). This will maintain the slave axis at an adjustable limit speed command value, which cannot be exceeded.

Another common application of the master/slave principle is in the paper industry where different torque settings (9) are required for the winding process.

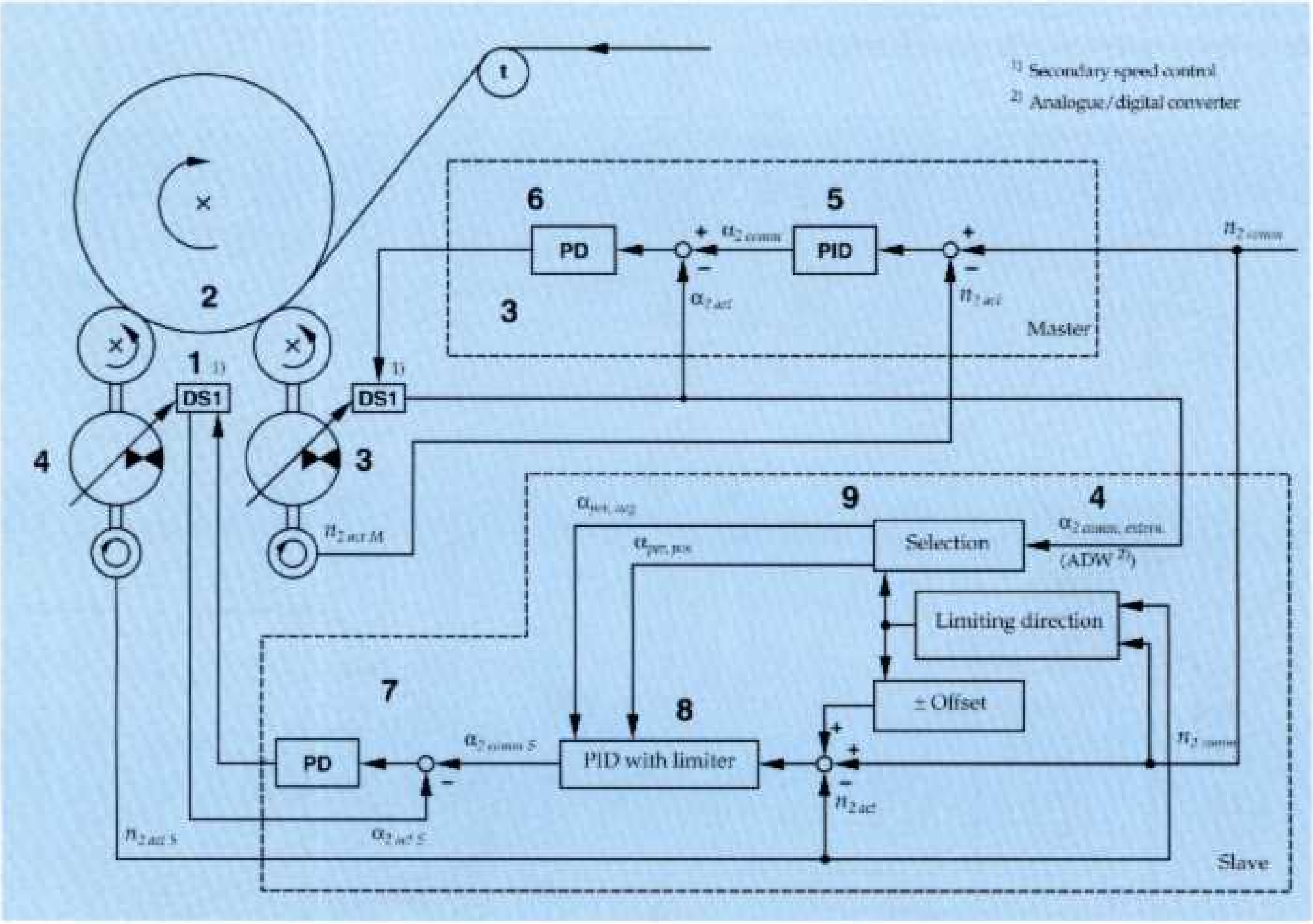


Fig. 43: Secondary control with master/slave principle

Digital control of secondary controlled units

Development in automation technology is characterised today by the great advances made in the integration of semiconductors, leading to favourably-priced digital processors and memories. In 1971 the first microprocessors to be combined with semiconductor memories and input/output components, producing reasonably priced micro processing units, appeared on the market, taking it by storm. With the aid of microprocessors, which far exceed the capabilities of the earlier process computers, decentralised automation systems can be built up. At the same time the microprocessors are being increasingly used with digital controls and freely programmable control systems.

Secondary control must, however, be usable with digital control if it is to follow the trend towards automation.

As well as being able to replace one or more analogue controls, digital controls are capable of carrying out additional functions such as delayed control of command values, independent switching to different control and output values, adaptive control parameters dependent on the operating point, and also additional monitoring of limiting values.

Other new functions are:

Communication with other digital controls, mutual redundancy, automatic error detection and diagnostics and the selection of various control algorithms, in particular adaptive control algorithms.

Adaptive or self-adjusting control is of particular importance. With the help of an adaption algorithm, this uses the information contained in the measured process output signals to match a control to the process between two sampling points.

Complete closed loop control circuits can be achieved in a digital control. Some examples are cascade control, multivalue control with coupling control and disturbance variable compensation operations, that can be easily modified or completely changed either during commissioning or later by configuration of the software. In addition a large range of numbers may be used for the control parameters and scanning time.

The characteristics of digital closed loop control with process computers or microprocessors are as follows:

- Open and closed loop control algorithms are realised as software.
- Time quantised signals are produced.
- The signals are amplitude-quantised via finite word lengths in the A/D converter, the central unit and the D/A converter.
- Process analysis and control synthesis can be carried out by the computer itself.

As the open and closed loop control algorithms in the software are highly flexible, we are no longer restricted to standard components with P, I and D characteristics, as is the case with analogue control systems. Advanced algorithms from mathematical process models may also be used.

Another particular advantage is the fact that on-line digital controls permit process identification, control design and simulation processes to be undertaken.

MCS Digital control card

The MCS digital control card (**Fig. 44**) was developed for the control of axial piston units type A10VS that are fitted with a positioning device by way of a proportional valve. The card contains all functional groups necessary for determination of valve piston and swivel angle positions as well as for analogue speed feedback.

The block diagram for the MCS digital control card is shown in **Fig. 45**.

The following functions in the hardware are digital:

- Processing of all control and parameter data for the Philips/Valvo 93C100 micro controller with 16/32 bit processor.- RS 232 or RS 485 serial interface for connection to a personal computer (PC) or remote control box BB3.
- Release/error reset input and start/stop input, opto decoupled (24 V).
- Switching transistor for control of a check valve connected to an opto decoupled output ("CONTROL ACTIVE").
- Power-up reset.
- Watchdog circuit for monitoring the controller.
- 8 diagnostic LEDs for error and status messages.
- Optional card with 16 opto decoupled 24 V inputs/outputs, PLC connectable.

The following are analogue:

- Two differential or current inputs for command values or other input signals such as n_{comm}, α_{comm}, M_{Tcomm}, directed via a multiplexer to the A/D converter.
- Voltage distributor differential input for analogue tachometer and tuning amplifier for precise speed matching.
- Oscillator demodulator module for evaluation of the swivel angle inductive positional transducers and valve control.
- Reading the analogue voltage values for n_{comm}, n_{act}, α_{act} and γ_{act} by means of a parallel converting 12 bit/4 channel A/D converter.
- Output of control value via 14 bit D/A converter and a 0 ... ± 10 V tuning amplifier.
- Clocked output stage with current feedback for valve.

Fig. 44: MCS digital control card

- Monitoring of oscillator demodulator module, release input and watchdog circuit of the controller. When the monitoring system responds the switching transistor is blocked and a 24 V error message is output opto decoupled.

Fig. 46 shows the system arrangement of the controller card.

The standard software for the closed loop digital control comprises the following functional blocks:

- PID control with subordinate PD swivel angle control and PD valve spool positioning control.
- Speed command value ramp with pre-set acceleration and deceleration.
- Safety functions by means of software monitoring of:
  - maximum speed,
  - speed differential,
  - acceleration,
  - swivel angle differential and
  - valve positioning differential.
- Direct switching of check valve via hardware or software monitoring.
- Simple operation with Start / Stop signal by means of sequence program for input / output sequence operations.
- Three control parameter sets permit speed-dependent switching.
- More exact influence of control circuit and output of status messages possible by use of plug board for PLC communication (for each one 16 opto decoupled inputs and outputs).
- Additional analogue input for special functions such as external setting of α_{max}, power limitation and torque.

A commercial PC or remote control box may be connected via a serial interface for setting the control parameters and for the display of status values. There is a console available for the PC for input of parameters and display of process data. Parameters are protected against unauthorised modification by means of a code word.

By setting control modes the control circuits may be individually activated, matched and optimised during commissioning.

Matching of the swivel angle positional transducer is also possible to a certain extent.

Other software components include open or closed loop torque control with compensation of the friction characteristic, load limitation and the master/slave principle.

The hardware described is designed specially for use with the A10VSO series.

When using an A4VS secondary unit the same software is used with the hardware, but with different system interfaces.

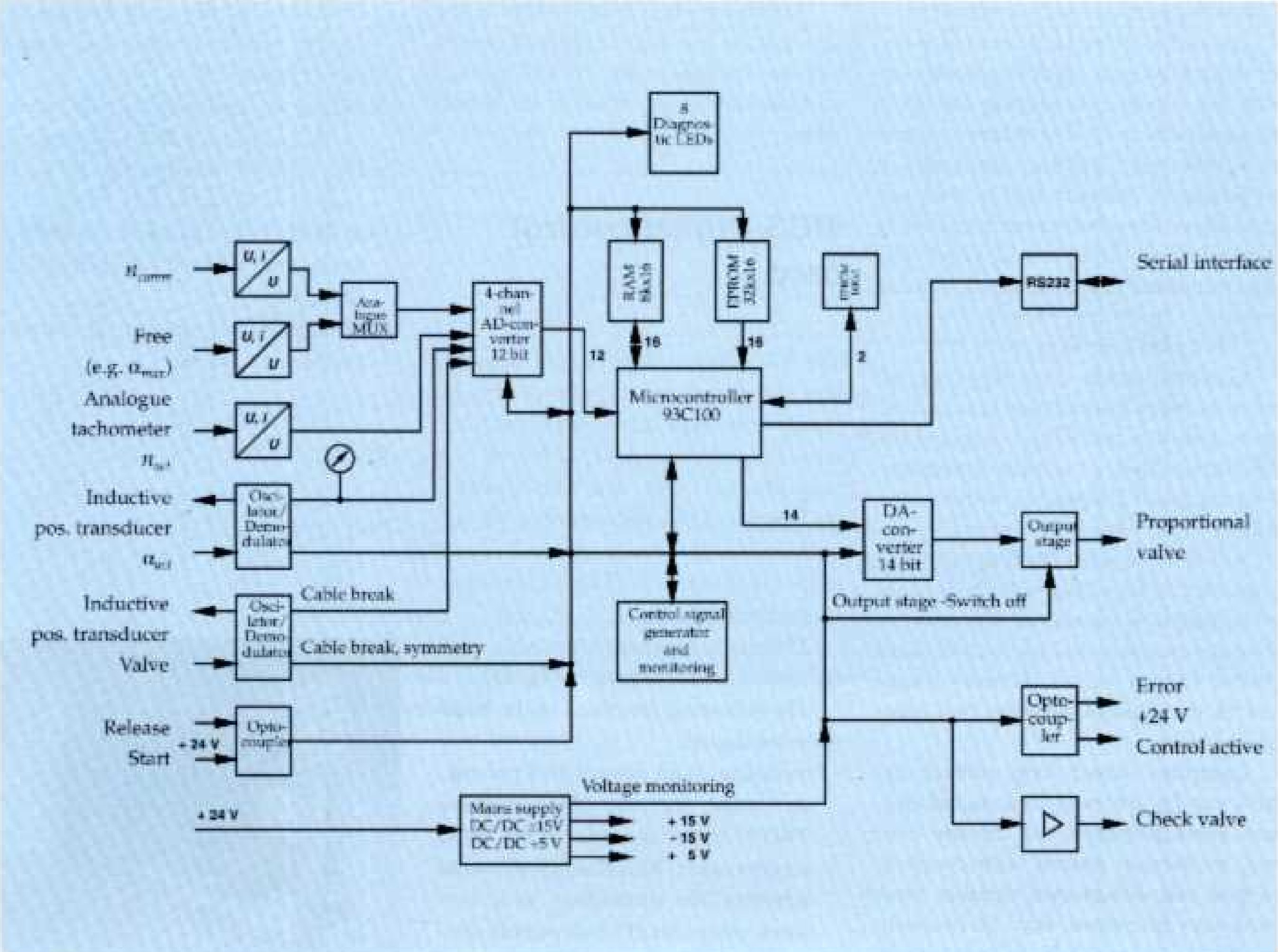


Fig. 45: Block diagram of MCS digital control card

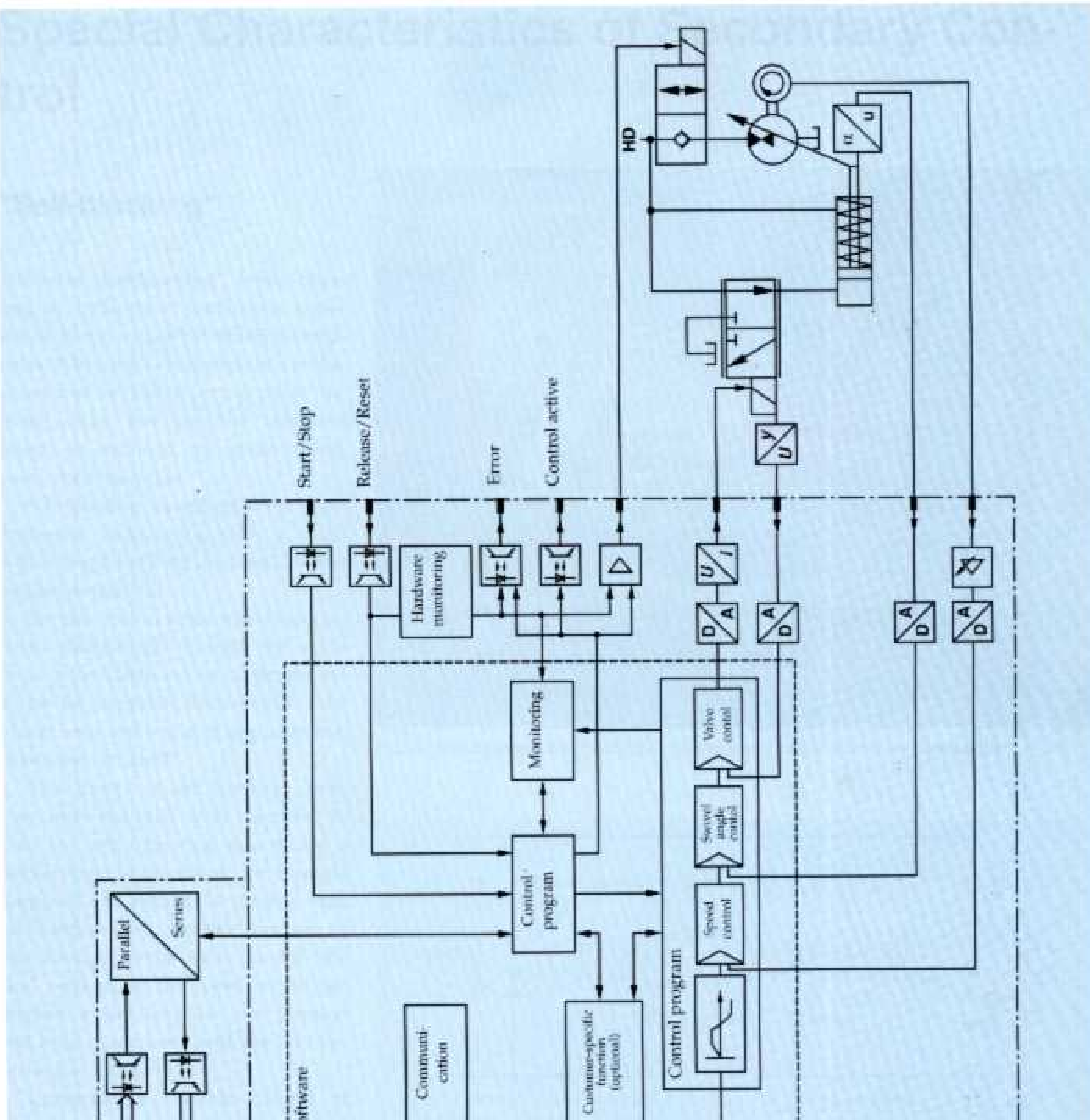


Fig. 46: System arrangement of MCS digital control card

Special Characteristics of Secondary Control

"Self-blocking"

The term "Self-blocking", when referring to hydrostatic units, was introduced in the forties by hydraulic engineers. When the swivel angle of a motor is reduced beyond a certain point, the motor does not generate sufficient torque to maintain the given speed, even when unloaded.

Self-blocking means therefore - uncontrolled deceleration with a small swivel angle until the hydraulic motor reaches a standstill.

For this reason some manufacturers have mechanically limited the minimum permissible swivel angle, for example for hydraulic motors with axial piston units and variable displacement, to between 4° and 7°.

This theory of self-blocking, however, does not hold with scientists. If, with the unit operating as a motor, a swivel angle of approx. 4° for example decreases and eventually reaches zero, no continuously variable control of swivel angle could occur through the zero position, as the speed would not increase again until the zero position had been crossed and with the unit operating as a generator.

Continuously variable control of swivel angle through the zero position is, however, imperative with secondary control. Without it no secondary control would be possible.

In 1980 a series of tests were started to prove the linearity, which fulfilled all expectations with regard to linear operation during swivel angle control through zero. The term "Self-blocking", when used in this context, therefore had to be redefined.

In 1993 the Dresden Technical University was given a contract to carry out a series of tests to examine the proportional relationships between torque and swivel angle, taking into consideration pressure differential, speed, fluid temperature and long-term behaviour.

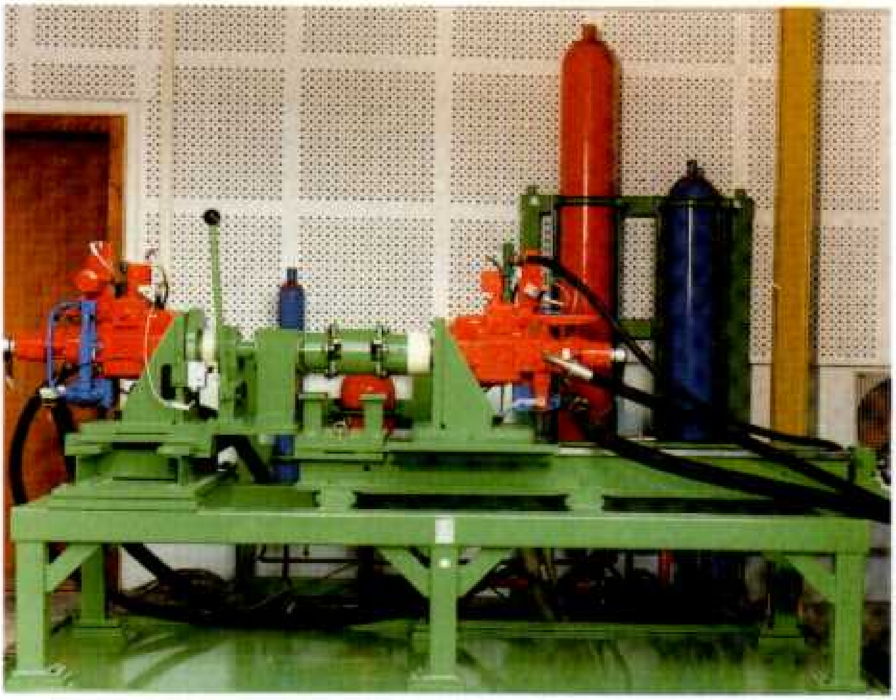


Fig. 47: Development test stand; back tensioning principle with axial piston units

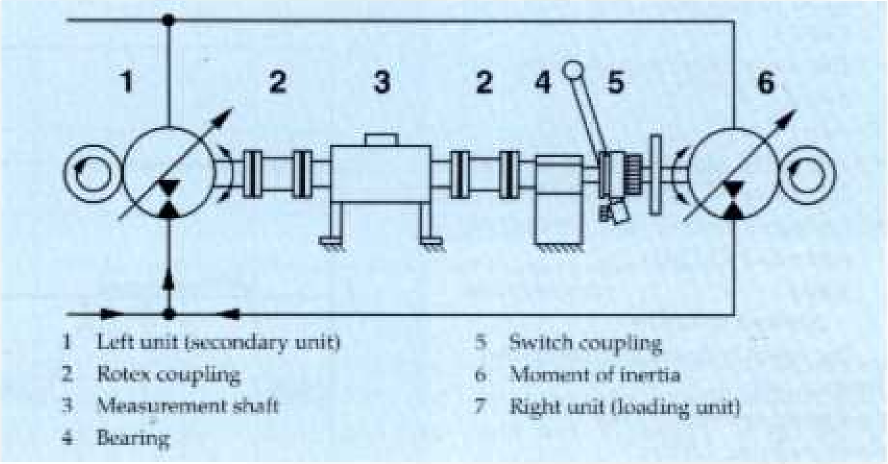


Fig. 48: Basic circuit of test equipment

A two-axis test rig **(Fig. 47)** was used for the tests. This test rig consisted of two A4VSO axial piston units tensioned against each other.

The secondary unit with the built-on DC tacho-generator (30 volts at 1000 rpm) has a displacement of 71 cm³, the loading unit having a displacement of 40 cm³.

The two-axis test rig is connected to a central oil supply, to which three other secondary controlled units are also connected.

The basic circuit for the test equipment is shown in **Fig 48**. The central oil supply is not shown.

The two axial piston units are interconnected by means of Rotex and dog clutches. The left hand secondary unit is speed controlled (closed loop) the right hand one being torque controlled (open loop), thus tensioning the transmission line. The load torque generated by the torque controlled unit is measured by means of the measurement shaft situated between the Rotex couplings.

The tensioning power is infinitely adjustable in all four quadrants by means of the speed input and load torque. The unit on the left operates as a motor, the right as a generator. With the associated power flow
mechanical —> hydraulic —> mechanical, only the power loss from the central oil supply needs to be generated.

The series of tests was carried out with the following components and technical data:

Loading unit

- Axial piston unit type A4VSO40 with built-on proportional valve STW0063
- Digital control electronics MCS, type VT0235

  The swivel angle is closed loop controlled.

Secondary unit

- Axial piston unit type A4VSO71 with built-on servo valve 4WS2EM10
- Analogue control and monitoring electronics VT12000

  Speed compensation $2040 rpm \equiv 10 volts$

  The speed is closed loop controlled.

Technical data

Operating pressure: 275 bar
Boost pressure: 7.5 bar
Fluid temperature: 50° C

Fig. 49 shows the swivel angles over centre or torques of both units at constant speed.

where:

$\alpha_{act\ load}$ Swivel angle actual value of loading unit

$\alpha_{act\ sec}$ Swivel angle actual value of secondary unit

n_{act} Speed actual value of tensioned axis

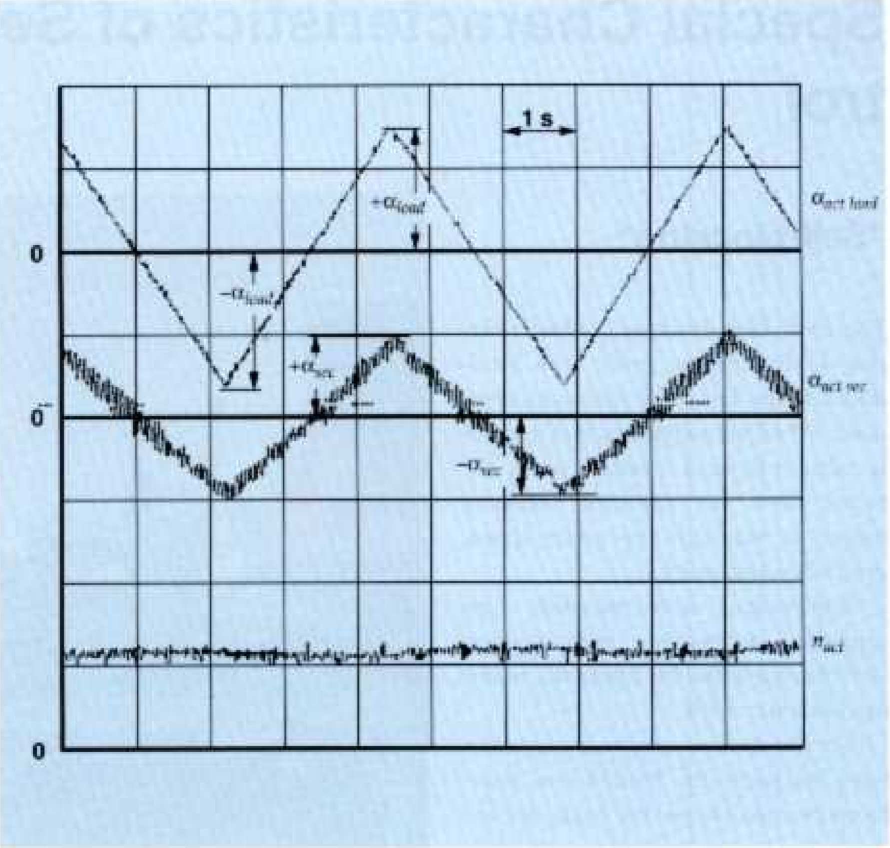


Fig. 49: Centre crossover point at constant speed

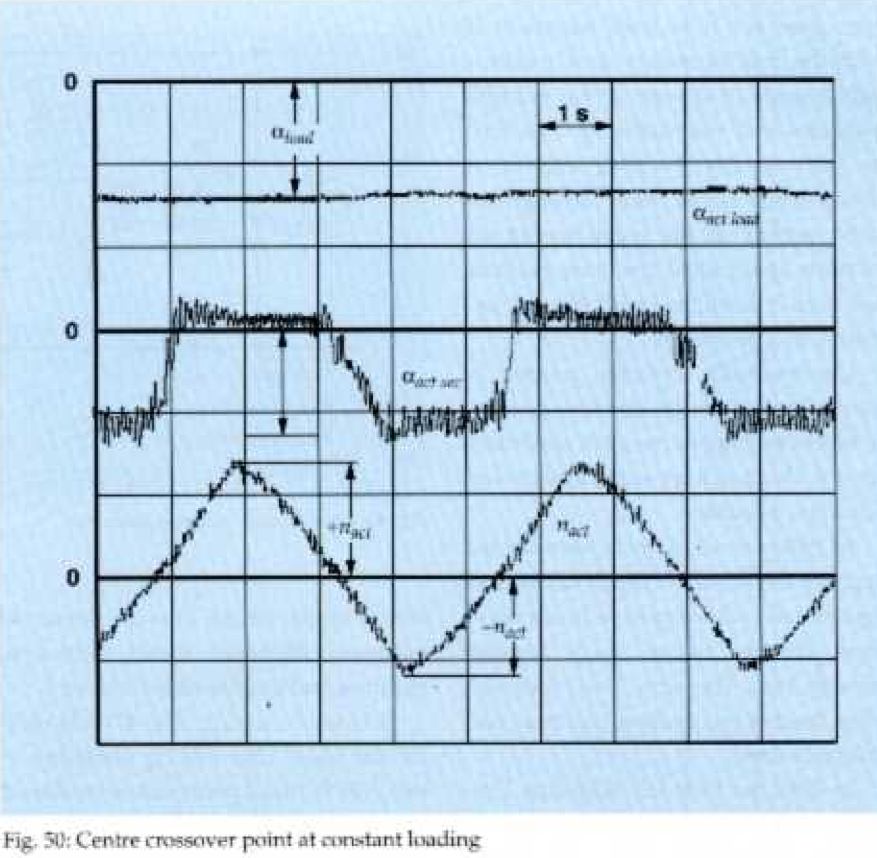


Fig. 50: Centre crossover point at constant loading

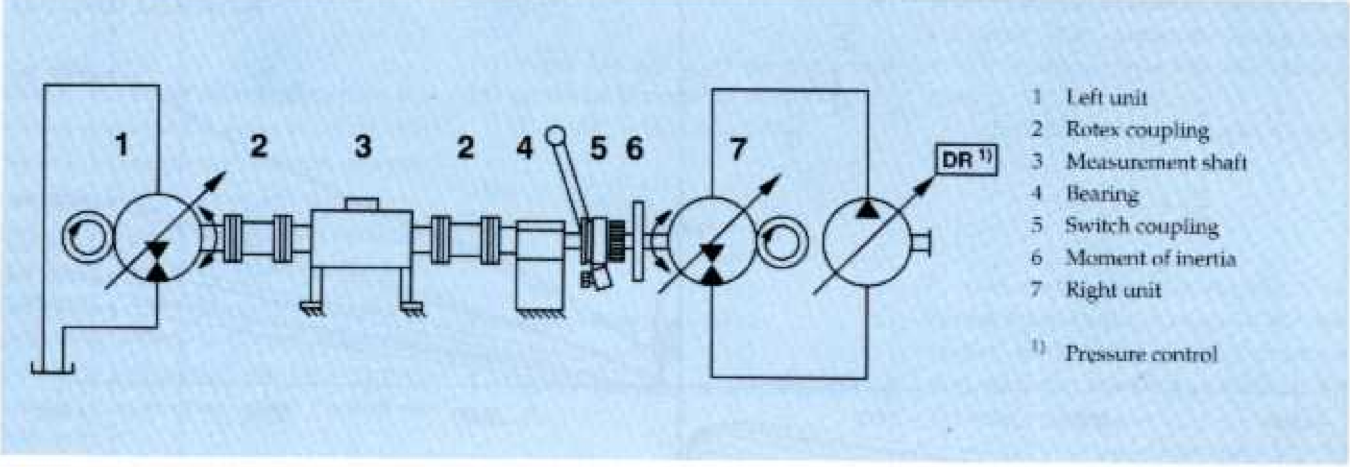


Fig. 51: Test set up; swivel angle against speed

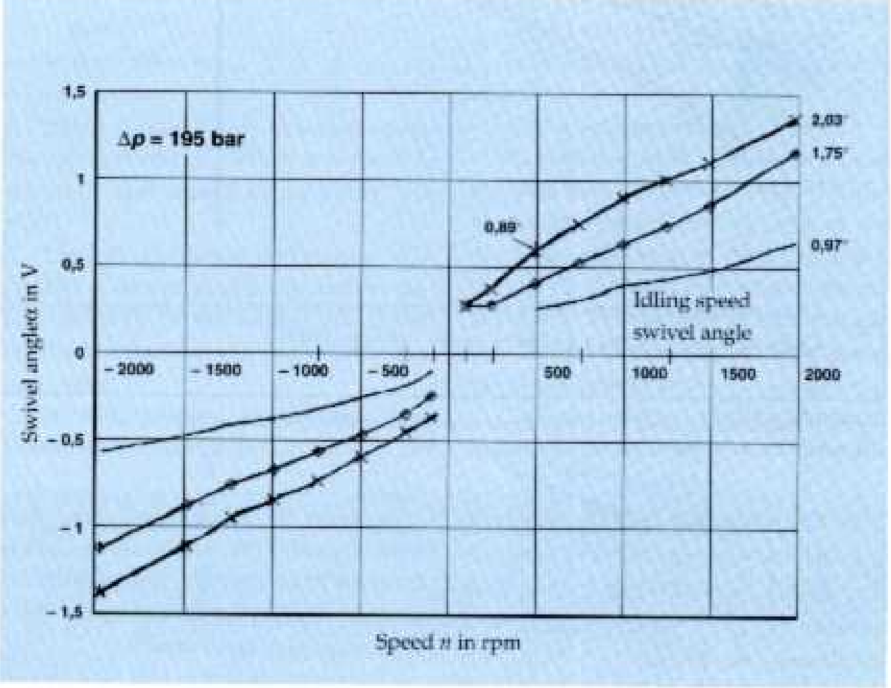


Fig. 52: Measurement results; swivel angle α in volts against speed n in rpm

The speed of the axis remains constant at 210 rpm, the swivel angle of the loading unit being controlled in a triangular function with a frequency of 0.2 Hertz in the range of +4.4 to -4.65°. During this phase there will be continual changeover between motor and generator operation.

The secondary unit completes the load change. Here too there will be continual changeover between motor and generator operation.

The swivelling actions of both units are linear and there will be no discontinuous action.

The "shaky" progression of the swivel angle actual value of the secondary unit indicates a dynamic equilibrium during the speed control process. The displacement of the loading unit is 44% lower than that of the secondary unit, a fact that must be considered when converting the swivel angle into a torque value.

As can be seen in **Fig. 50**, at constant generator loading and with the loading unit operating as a pump, speed reversal will occur, a process which can for example occur with upwards and downwards movement of a constant load on a crane. The swivel angle of the loading unit here will be 4.5°. The speed change of the secondary controlled axis within the range of +280 to -270 rpm with a frequency of 0.2 Hertz will result in a swivel angle change of the secondary unit from 3.8 to 0.4° ± 0.15°. As expected this transition shows no discontinuous action. The noticeable dynamic equilibrium prevents static friction from occurring.

Here is an interesting question in this context :

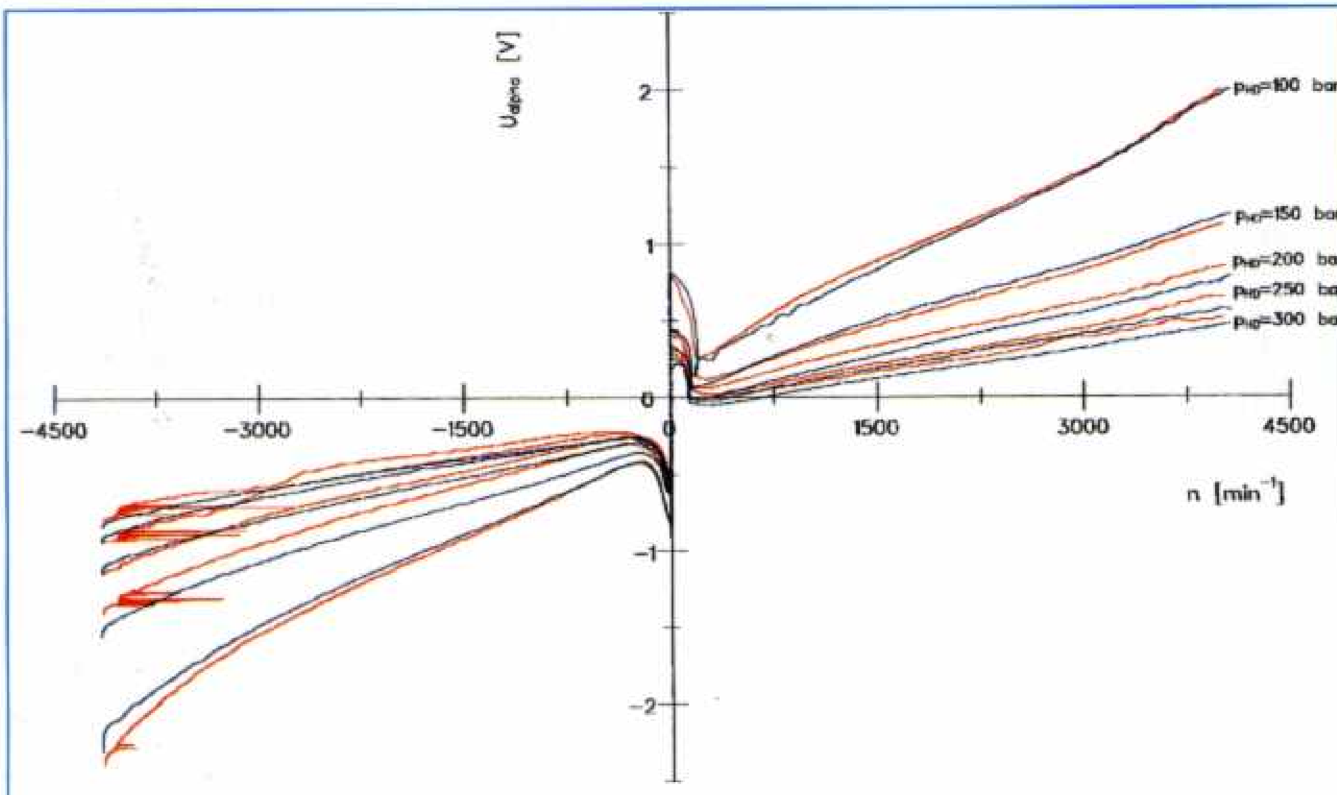
What will be the minimum swivel angle for a given operating pressure with respect to speed in a state of equilibrium? These measurements were carried out again on a tensioning test rig, using in this case two axial piston units, each with a displacement of 250 cm³ (**Fig. 51**). The unit on the right is speed controlled in closed circuit operation.

The operating pressure is generated by a pressure controlled axial piston unit with a pressure of 200 bar, the boost pressure being 4.5 bar.

The unit on the left will be merely "dragged along" at a swivel angle of 0°. An additional moment of inertia of 1.9 kgm², the three couplings and torque measurement shaft are also driven.

Fig. 52 shows the swivel angle progression in volts against the speed in a state of equilibrium ($10.3 volts \hat{=} 15°$).

In the top curve, which is recorded at a drain oil temperature of 28°C of the unit on the right, the swivel angle shows an almost linear increase from



Characteristic of right and left A4VSO71 unit at various operating pressures with clockwise and anti-clockwise rotation.
Command value settings for speed range: –10 V to +10 V

Fig. 53: Measurement results; swivel angle α in volts against speed *n* in rpm

0.89° and 500 rpm to 2.03° at 2000 rpm. It is inversely proportional to temperature, at 43°C and 2000 rpm having been reduced to 1.75°, corresponding to a displacement of 28 cm^3.

If the unit on the left is disconnected via the dog clutch, presenting a reduced moment of inertia to the drive, the swivel angle will be reduced at 2000 rpm to 0.97°. The unit will then still have a displacement of 16 cm^3. The flow requirement at idling speed will also be correspondingly small.

To complete the picture it should be mentioned that an increase in operating pressure to the levels commonly used today (280 - 300 bar) will cause a further reduction of the swivel angle.

It must be emphasised here that, with respect to the "self-blocking" of hydrostatic axial or radial piston units, regardless of type, the torque on the drive shaft must be reconsidered.

The curve shown in **Fig. 52** excludes the speed range below 250 rpm. Yet the knowledge of this area is important for start-up. Using a tensioning test rig with units of 71 cm^3 displacement, **Fig. 53** shows a record taking into account the lower speed range with start-up from zero speed at various operating pressures. If a speed command value is pre-set by means of a time ramp, an initial breakaway torque will be generated at a pre-set operating pressure and an increasing swivel angle to overcome the static friction. As the coefficient of friction is considerably smaller when movement takes place and the torque requirement is thus lower, after breakaway the swivel angle will reduce once more, to enable it to increase in a linear fashion from approx. 50 rpm upwards (Stribeck curve). These measurements were carried out both with increasing and decreasing speed. There is clear evidence of hysteresis, brought about by the frictional torque being lower when the axial piston unit is decelerating than that when the unit is accelerating. A reduced swivel angle will thus be necessary in order to maintain the lower speed.

The measurements were carried out at an operating temperature of 44°C in closed circuit operation (p_{boost} = 15 bar), the pre-set command value for the speed range being ± 10 volts.

Torque control

Due to the linear relationship between swivel angle and torque it seems reasonable to use the swivel angle at a preset operating pressure for torque control with secondary controlled axial piston units. However, knowing the relationships between swivel angle, operating pressure, torque, fluid temperature and speed (Stribeck curve), use of a correction curve cannot be avoided.

As the usual torque measurement shafts only attain a sufficiently long service life in static operation, torque control is being increasingly applied with electrical and hydraulic loading units, which dispenses with the need for this measurement device.

Torque control with electrical machines

For decades now the DC shunt generator dynamometer has been used as a loading unit in test environments for power ranges of up to 1000 kW. This is due to its stable operating behaviour and the fact that it is not prone to oscillations.

Even power converter-fed rotary generators which have recently grown in popularity render possible near-realistic simulation on the test rig, of a road traffic situation for example.

Both machines are designed for external cooling, the cooled air being directed via the rotor surface, through channels in the stack and also between the poles.

As the electrical motor is pendulous and torque measurement is carried out on the balance principle, this motor leaves something to be desired with respect to response sensitivity and reaction to mechanical influences.

With torque control the torque is calculated from the size and speed of the machine.

Electrical motors equipped for torque control have a built-on digital speed tachometer at the through shaft end.

Torque M_T acting at the shaft of a DC generator is made up as follows:

$$M_T = M_E - M_0 - M_A \pm M_b,$$

where
M_b = acceleration torque in Nm,
M_E = electrical torque in the bearing clearance in Nm,
M_A = additional loss torque in Nm,
M_0 = idling torque in Nm.

The air gap torque is calculated as follows:

$$M_E = \frac{60 \bullet E}{2 \bullet \pi \bullet n} \bullet I_A,$$

where
E = "primary voltage" or power in volts of electric motor,
I_A = armature current in A,
n = speed in rpm.

Idling torque M_0 is determined by the idling current I_0, where $I_0 = f(n,E)$ is recorded and stored in an idling test run.

The low additional loss torque M_A= f(n,IA) is determined in the factory before delivery of the motor and stored for the nominal current in the computer as a curve.

In this process the current heat losses in the armature circuit are not taken into consideration. These losses influence the armature tensioning, which is not necessary for the calculation.

Acceleration torque of the rotating mass,

$$M_b = J \bullet \frac{d\omega}{dt}$$

with J as moment of inertia, is accounted for in the run-up of the unloaded machine by a zero compensation, without needing to know the absolute value.

All technical data determined by the shaft torque such as armature current I_A, speed n and primary voltage E can therefore be measured i.e.

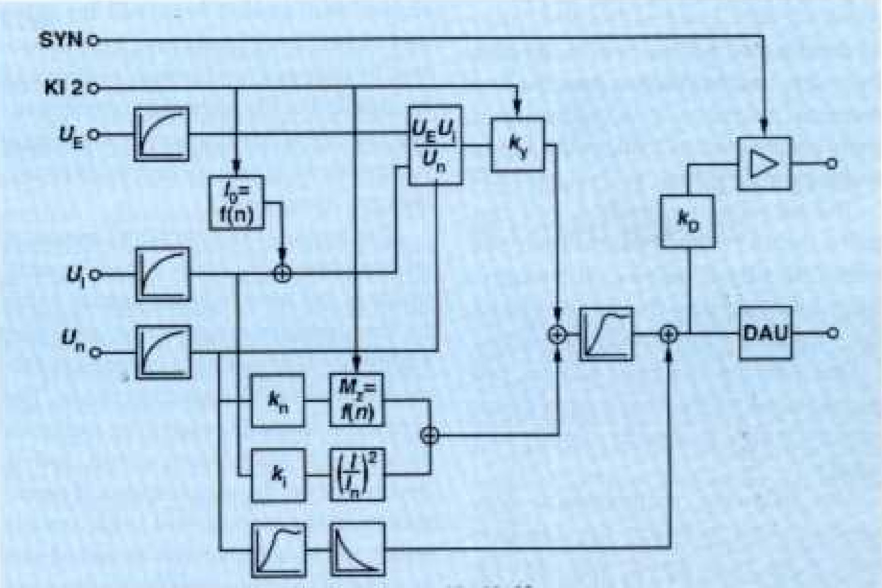
-I_A at a shunt resistor,

-n at the digital speed tachometer and

-E at a voltage bridge.

By referring back to a micro processor with associated memories all losses within the total operating area of the machine are picked up. Aligning operations for individual loss torques are not necessary. This simplifies commissioning, further compensation being possible at any time.

The electronics industry stipulates that with this method of measuring val-



Syn: Synchronisation
KL2: Measurement range 2
DAU: Digital/analogue converter
U_E, U_i, U_n: Analogue input voltages
k_n, k_y, k_i, k_D: Standardisation factors

Fig. 54: Functional principle of torque calculator

ues I_A, n and E the shaft torque output can deviate by <1%.

The torque calculator, which is equipped with a 16 bit micro processor, operates as follows:

The analogue input voltages U_E, U_i and U_n are sent to the computer via an analogue-digital converter.

Using these values the micro processor determines the electrical torque taking into account the idling current (**Fig. 54**). The values for M_A and M_b are then determined and added to this value.

This resulting torque value corresponds to the mechanical torque at the shaft. This torque is determined at intervals of 1.5 msec via a digital-analogue converter. The 9 volts corresponds to the nominal torque. A mean value is calculated and output as a digital display.

The torque value determined by the torque calculator fulfils the requirements for torque control in the areas of dynamic response and statics with sufficient accuracy, without the need for a torque measurement shaft.

However, this is required with a torque closed loop circuit, where the torque actual value has to be measured.

Torque control of secondary controlled units

As the velocity dynamics of hydrostatic units vary and are higher than those of electrical machines, it is necessary to apply torque control without the torque measurement shaft.

The secondary controlled unit cannot be installed as a dynamometer due to the difficulty of passing oil through a rotary joint and also due to the inaccuracies caused by the associated friction.

One solution is torque control, taking into consideration the control piston position which is already output as a voltage.

The following mathematical relationship exists between the displacement of an axial piston unit and the swivel angle, represented by the control piston position:

Displacement of an axial piston unit in swashplate design:

$V_g = 2 \bullet r_Z \bullet A_K \bullet \tan\alpha$.

Displacement of an axial piston unit in bent axis design:

$V_g = 2 \bullet r_h \bullet A_K \bullet \sin\alpha$.

Displacement of a radial piston unit:

$V_g = 2 \bullet e \bullet Z \bullet A_K$.

where:

A_K = piston area in cm^2,
r_Z = geometrical radius of cylinder drum in cm,
r_h = geometrical radius of drive flange in cm,
V_g = displacement in cm^3,
Z = no. of pistons,
α = swivel angle in °
e = eccentricity in cm,
η_{mh} = mechanical-hydraulic efficiency.

From the factors displacement and pressure differential we arrive at the torque

$$M = \frac{\Delta p \bullet V_g}{20 \bullet \pi} \bullet \eta_{mh}$$

or, simplified, to

$$M_T \sim \Delta p \bullet f(\alpha).$$

Torque conversion by means of operating pressure and displacement or swivel angle or eccentricity is the same for all hydrostatic units. In hydrostatics, we are returning to the various methods of altering the displacement with torque control from which other drive systems are excluded.

The demands of the automotive industry and related branches for open and closed loop speed and torque control in dynamic endurance testing can be fulfilled with secondary control using the same loading units for torque control as for speed control without additional components.

The torque is calculated by means of an analogue calculation circuit by multiplying the swivel angle actual value by the pressure actual value and then adding or subtracting a correction factor, depending on operating mode. The correction factor is created by means of an analogue calculation circuit and is dependent on the actual speed. It corresponds to the extra swivel angle around which the swashplate has to swivel out in order to compensate for mechanical leakage of the unit.

In order to obtain as accurate a simulation of the correction curve as possible by means of a calculation circuit, the characteristic of each secondary unit must be measured individually and the circuit designed accordingly.

The analogue calculation circuit and all necessary additional functions are contained on the MD1/VTS0229 torque controller card. This card is used with the VT12000 analogue control and monitoring electronics necessary for speed control.

The MD1 torque controller card operates as follows (**Fig. 55**):

The torque command value $M_{T command}$ is entered as a voltage (0 to ± 10 volts), linked with the speed actual value n_{actual} and passed on to the calculating circuit. Using the pressure actual value Δp the torque calculator assigns and outputs a swivel angle command value $\alpha_{command}$.

The correction factor (**Fig. 53**) is created dependent on the size of machine selected and its respective speed and pressure.

Correction value $U_a = f(n)$ is identical to the idling torque M_0 and is determined on the test rig before the unit is delivered.

Therefore the correction value reproduces a voltage (torque) dependent on unit size, which also has to be generated when operating as a motor in order to compensate for inherent losses from the unit.

This means that, for example, if a zero torque is required at the output shaft, the unit will operate as a motor with a speed-dependent correction value.

In motor operation the correction value is added to the pre-set torque command value and is converted into a swivel angle, taking into consideration the speed, which is then passed on to the VT12000 standard control electronics via a limiting circuit.

Determination of torque considers both motor and generator operation, where the correction value is subtracted from the calculated torque command value.

The signal $\alpha_{command}$ sent from the torque card is adjusted with the swivel angle controller on the VT12000 control and monitoring electronics.

With the analogue calculating circuit $\alpha_{command}$ is built up from the following values:

With motor operation

$$\alpha_{comm} = \frac{M_{T2comm} + M_V}{p_{act}} \bullet C_M.$$

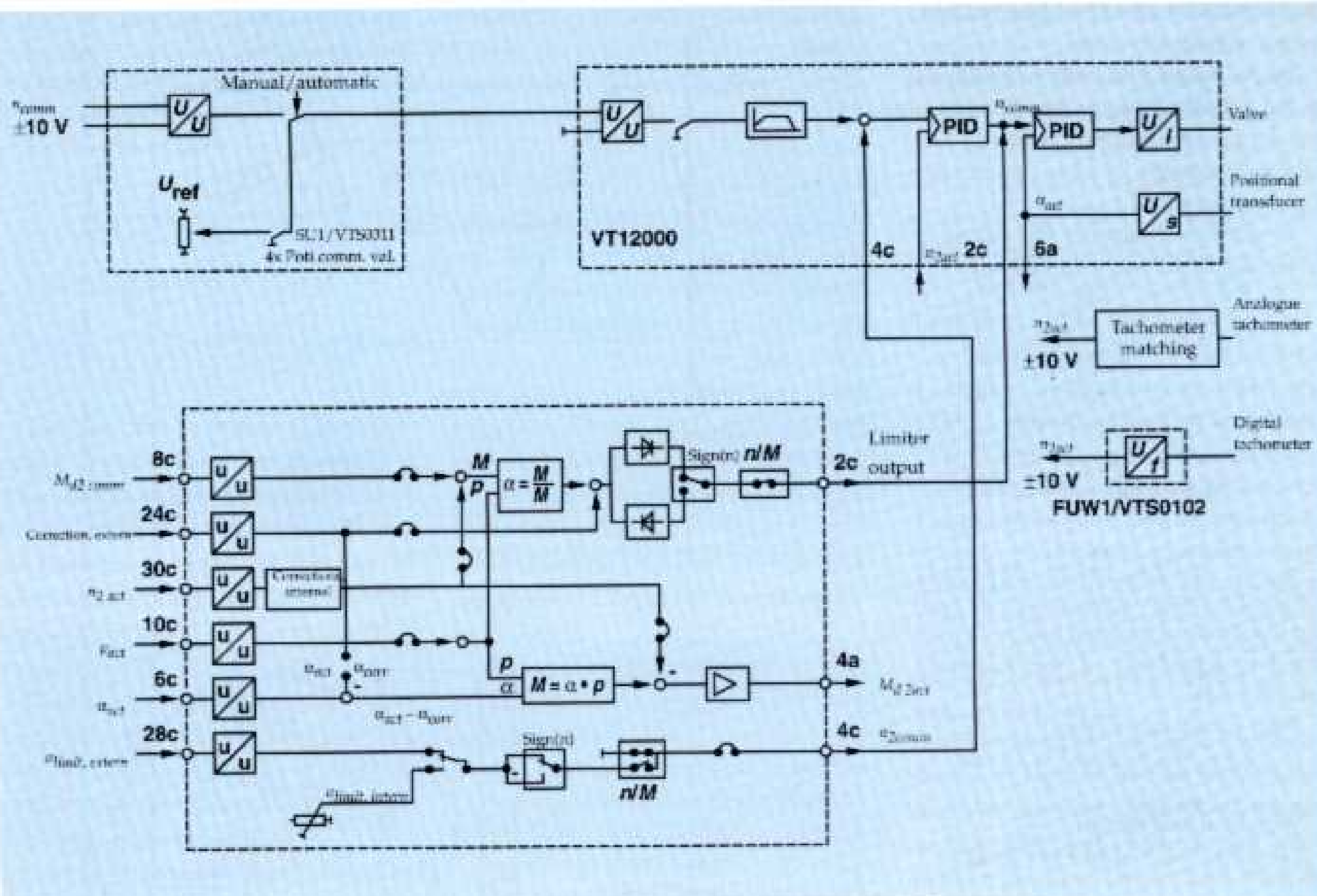


Fig. 55: Torque control card

With generator operation

$$\alpha_{comm} = \frac{M_{T2comm} - M_V}{p_{act}} \bullet C_G.$$

where

C_M = loss factor with motor operation,
C_G = loss factor with generator operation,
M_V = speed-dependent correction value,
p_{act} = pressure actual value in bar,
α_{comm} = swivel angle command value.

A change of torque will result in a speed variation at constant shaft loading. In order to prevent overspeed the torque card contains a speed limiting device with which any required maximum speed can be set.

The speed limiting value $n_{2command}$ is at the input of the PID speed control of the VT12000 standard control electronics. Below this ceiling speed torque control is effective using the method described. If the speed increases to above the limiting value the swivel angle command value will decrease until the speed drops to within the permitted range.

This speed limitation is effective in both directions.

The analogue characteristic can be replaced by a digital corrector card if higher accuracy is required or for extended applications. For a specific model the swivel angle command value is taken from a previously recorded look-up table, and is dependent on speed, pressure and temperature.

The look-up table can be read in and corrected at any time.

With the digital corrector card, $\alpha_{command}$ is produced as follows:

With motor operation

$$\alpha_{comm} = \left(\frac{M_{T2comm}}{p_{act}} \bullet C_M\right) + \alpha_{cor}$$

With generator operation

$$\alpha_{comm} = \left(\frac{M_{T2comm}}{p_{act}} \bullet C_G\right) - \alpha_{cor}$$

α_{cor} is an external swivel angle corrector value that takes into consideration the model of axial piston unit, the speed, pressure and temperature.

Behaviour of the drive at creep speeds

For drives that need to cover a wide speed range, in addition to the characteristics of bearings, guides and disturbance factors, the running of a secondary controlled motor in the lower speed range of <10 rpm is of particular importance.

A limiting value which determines the behaviour of a hydrostatic drive operating at creep speeds is the degree of uniformity of flow δ. It is a measure of the fluctuation of an angular velocity from a reference velocity.

As the speed is directly proportional to the angular velocity the dimensionless degree of uniformity δ is also suitable as a measure of speed fluctuation from a mean speed n_{mean}.

In machine dynamics the degree of uniformity is defined as follows:

$$\delta = \frac{n_{max} - n_{min}}{n_{mean}}.$$

Transferred to flow this means:

$$\delta = \frac{Q_{max} - Q_{min}}{Q_{mean}}.$$

If we analyse piston movement within the piston units it can be seen that the stroke of a piston follows a pure sine law regardless of the kinematic data (**Fig. 56**).

With swashplate design
$Q = 2 \bullet r_Z \bullet A_k \bullet \tan\alpha \bullet \omega \bullet \sin\omega t$.

With bent axis design
$Q = 2 \bullet r_b \bullet A_k \bullet \sin\alpha \bullet \omega \bullet \sin\omega t$.

where
A_K = piston area in cm^2,
n = speed in rpm,
Q = flow in L/min,
r_b = geometrical radius of drive flange in cm,
r_Z = geometrical radius of cylinder drum in cm,
α = swivel angle in °
$\omega t = \psi$ = swivel angle.

If the strokes of the individual pistons on the delivery side are added (vectorial), for an axial piston unit with 9 pistons we achieve the 9-sided polygon as shown in **Fig. 57**.

The degree of uniformity is immediately recognisable if we roll out this polygon onto a flat surface. Its frequency for an odd number of pistons will be

$f = 2 \bullet n \bullet Z$, in other words it will run at twice the speed.

It is evident that the uniformity pattern for an uneven number of pistons progresses with $\pi/(2 \bullet Z)$ and for an even number of pistons with π/Z i.e. the step of even piston numbers is twice as large.

Pressure side

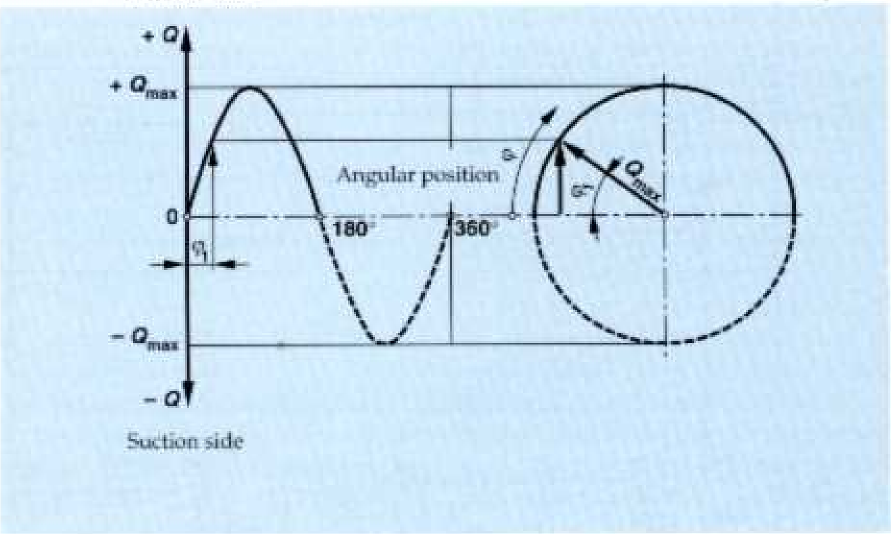


Fig. 56: Piston flow as function of angular position φ

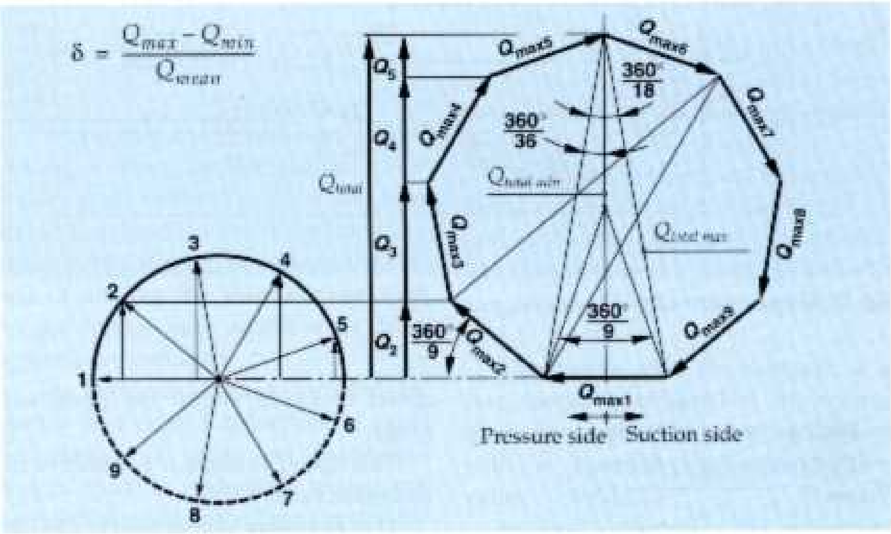


Fig. 57: Vectorial representation of the degree of uniformity for 9 pistons

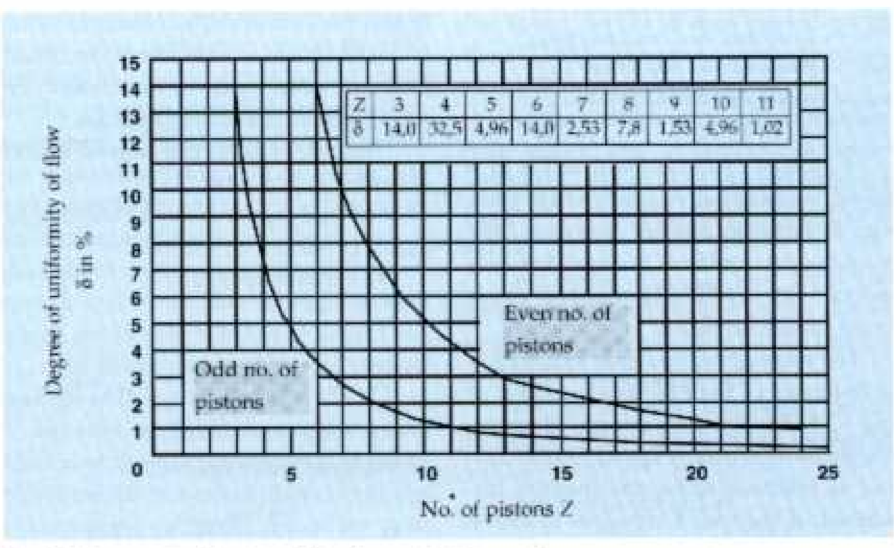


Fig. 58: Degree of uniformity of flow for axial piston units

Fig. 58 shows in graphic form the uniformity of systems with an even and odd number of pistons.

The following equations apply:
for an even number of pistons:

$$\delta = \frac{\pi}{Z} \bullet \tan\frac{90^\circ}{Z} \bullet 100 \text{ in } \%,$$

for an odd number of pistons:

$$\delta = \frac{\pi}{2Z} \bullet \tan\frac{45^\circ}{Z} \bullet 100 \text{ in } \%.$$

It is clear that, with respect to uniformity, a 10-piston drive unit is equally as good as the 5-piston drive unit. Both have an irregularity of approx. 5% and are therefore not used for exacting requirements, 7 or 9-piston drive units being most suitable with an irregularity of only 2.5 or 1.5%.

The tangent values of the reproduced formulae of both systems are in a ratio of 2:1. The uniformity of the units with an odd number of pistons is thus four times more accurate.

Where extremely small irregularities are required it is recommended that two units with an odd number of pistons be coupled together in such a way that one unit follows the other by $\frac{1}{4}$ piston division. Thus a degree of uniformity is obtained that lies between 17 and 19 on the graph for an odd number of pistons, although the number of pistons of this tandem unit is 18, an even number.

This method of reducing the irregularity by coupling together two axial piston units is, however, not always effective. Measurements from the test rig have shown that the actual degree of uniformity is also influenced by reversing operations, increasing if the operating pressure rises and deviating considerably from the calculated value.

Due to the final odd piston number, when rotating an axial piston drive unit, either 3 or 4, or alternatively 4 or 5 main pistons are connected to the pressure side of the control plate. The displacement changes to the same degree per radian. If a constant flow is supplied to the axial piston unit at constant torque when the unit is operating as a motor, this will lead to speed fluctuations at low speeds, this in turn causing pressure fluctuations. The torque can no longer be transferred evenly.

Fig. 59 demonstrates the behaviour of the speed of an axial piston bent axis motor with a fixed displacement of 12 cm^3. Flow to and from the unit is regulated by means of throttles and remains therefore virtually constant. Due to the change in displacement per radian there will be a supply pressure change from 20 to 28 bar. The supply pressure upstream of the throttle will be 50 bar.

As can be seen from the recorder, the speed fluctuation at an average speed of 7.6 rpm will be ± 1.2 rpm. With an average speed of 2.2 rpm the fluctuation will be ±1.3 rpm.

With secondary controlled drives it is clear that alternative criteria must be applied, as

- the operating pressure either remains constant or is impressed,
- the flow is not throttled,
- speed is regulated and
- the swivel angle and hence the displacement are free values, adapting to the specific load.

Speed and swivel angle behaviour of a secondary controlled unit with a displacement of 28 cm^3 at an impressed pressure of 170 to 200 bar is shown in both recordings (**Fig. 60**). The top curve shows a speed fluctuation of ±0.35 rpm at an average speed of 5.9 rpm, the swivel angle varies between 0.25° and 1.5°. With the lower curve the speed fluctuation is ± 0.4 rpm, based on an average value of 1.6 rpm, the swivel angle fluctuation lying between 0.45° and 0.75°.

It is clear to see that the secondary unit during the positioning process acts in opposition to the torque irregularity, trying to avoid continual acceleration or deceleration of the drive. The system, in other words, endeavours to maintain constant displacement per radian, irrespective of the varying number of pistons under pressure.

The test series leading to the results shown in **Figs. 59** and **60** were carried out with moment of inertia of the secondary unit only, without any additional mass. Speed irregularity with the secondary unit decreases as the moment of inertia increases. The swivel an-

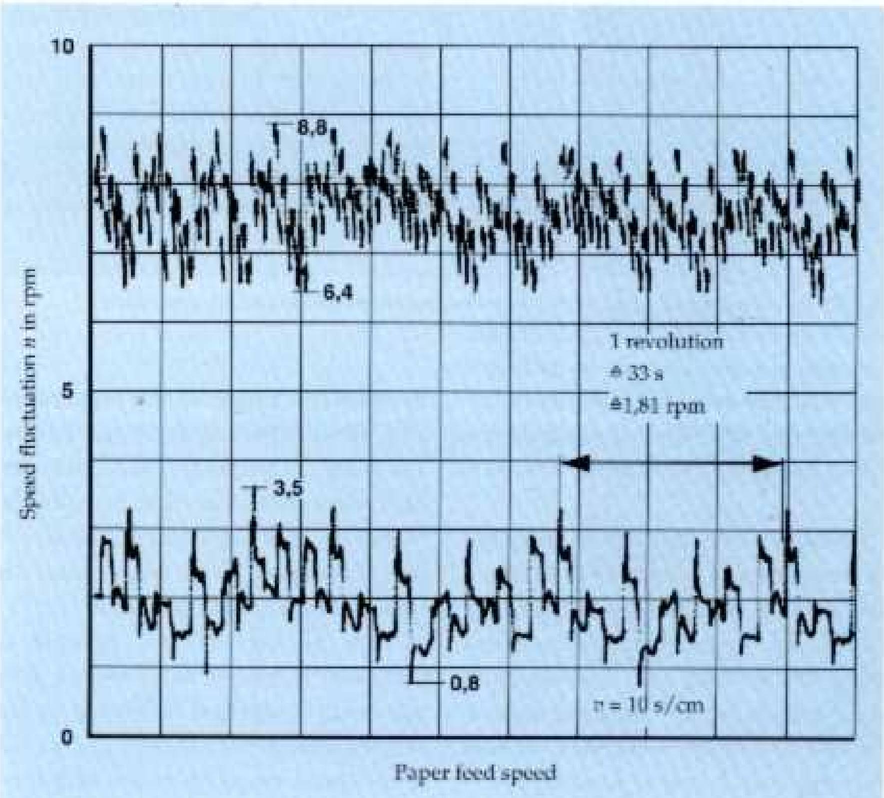


Fig. 59: Speed behaviour of an axial piston bent-axis motor

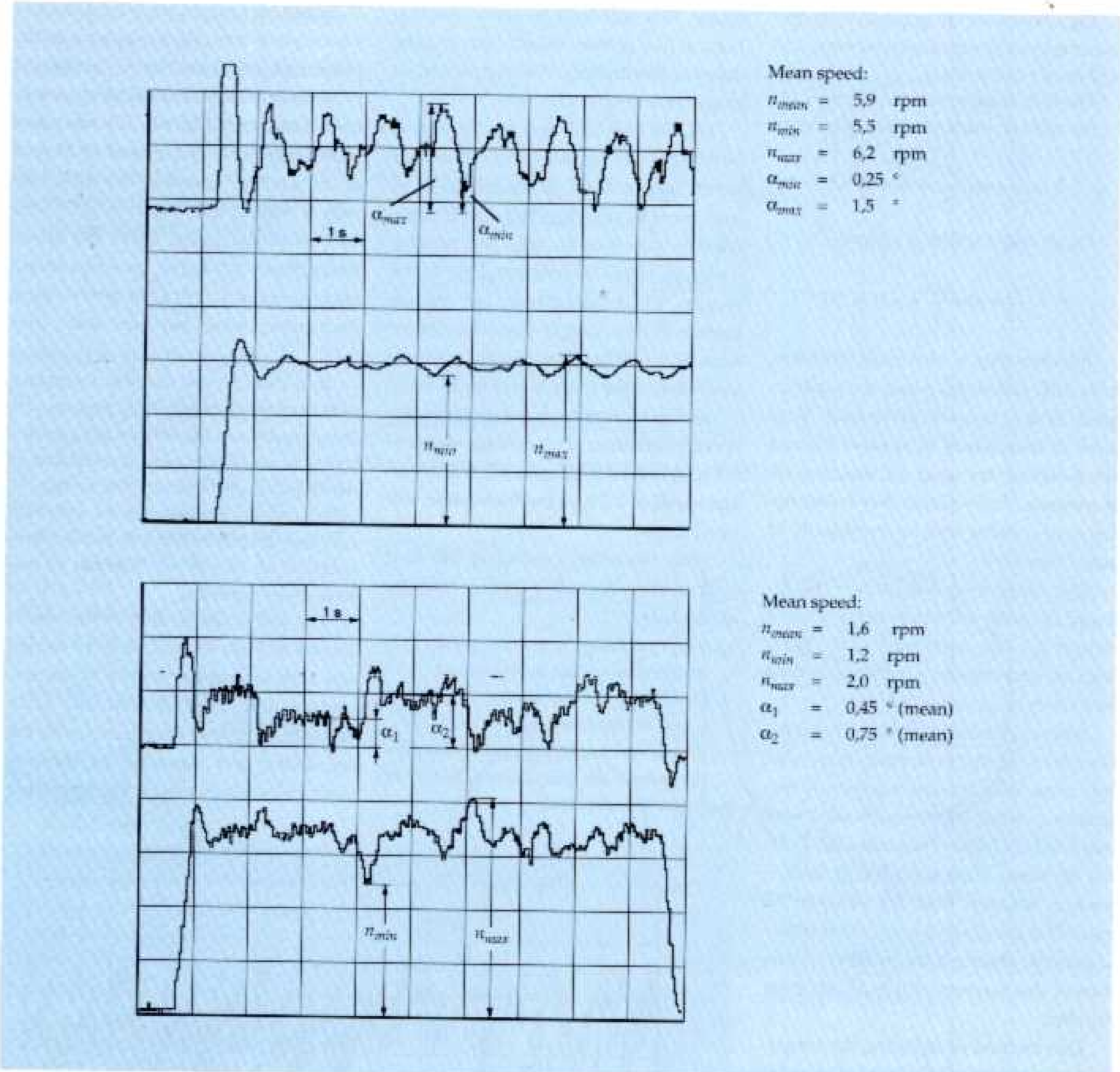


Fig. 60: Speed behaviour of an axial piston swashplate motor, secondary controlled

gle will, however, fluctuate more, although this will have no effect on the speed due to the increasing system inertia.

When a secondary unit is operating at creep speed it is possible to even out the speed to a degree by omitting the closed loop control circuit. As the speed deviation appears with a phase-shift, the control action by the speed controller will be too late to achieve smooth running. With a unit of small displacement and low speed the control characteristic will also deteriorate, because there will no longer be any proportionality between swivel angle and torque. The frictional torque and leakage losses dependent on the swivel angle will thus change to such an extent in the lower part of the speed range that no absolute smooth running state can occur.

This can, however, be improved if the speed signal for adjusting the displacement is changed in relation to the actual value of the swivel angle in the lower speed range by means of a control algorithm, so that the shaft torque of the unit remains constant. The swivel angle command value must in other words be corrected depending on the swivel angle and taking into account the efficiency. As the frequency of the irregularity is double the speed, with a 9-piston unit the correction must be every

$$\frac{360^\circ}{2 \bullet Z} = 20^\circ$$

(Z = no. of pistons).

The swivel angle command value has a positive effect on the accuracy of the drive if positioning is carried out from low speeds.

Natural frequency with secondary controlled units

The natural frequency of a drive is a known criterion for the stability and rigidity of a system. It can be used to calculate the shortest possible acceleration time that a drive system can achieve without oscillating. In a similar way to the spring-mass system the natural frequency of a hydraulic motor can be calculated from spring constant C and moment of inertia J:

$$f_e = \frac{1}{2 \bullet \pi} \bullet \sqrt{\frac{C}{J}} \text{ in } \frac{1}{s}$$

where:

C = spring constant in N / rad,
f_e = natural frequency in sec,
J = moment of inertia in kgm².

The natural frequency of a valve-controlled hydraulic motor (**Fig. 61**) is determined from the following values:

$$C_1 = \left(\frac{V_{g2}}{2 \bullet \pi}\right)^2 \bullet \frac{E_{oil}}{\left(\frac{V_{g2}}{2} + V_{01}\right) \bullet 10^4} \text{ in } \frac{\text{Nm}}{\text{rad}}$$

$$C_2 = \left(\frac{V_{g2}}{2 \bullet \pi}\right)^2 \bullet \frac{E_{oil}}{\left(\frac{V_{g2}}{2} + V_{02}\right) \bullet 10^4} \text{ in } \frac{\text{Nm}}{\text{rad}}$$

$$f_e = \frac{1}{2 \bullet \pi} \sqrt{\frac{C_1 + C_2}{J}} \text{ in } \frac{1}{s}$$

where

E_{oil} = modulus of elasticity of oil in kg / cm•s²
V_{g2} = displacement of secondary unit in cm³,
V_{01} = oil volume side A in cm³, and
V_{02} = oil volume side B in cm³.

The above-defined natural frequency, consisting of an oscillatory spring-mass system, is not possible with secondary control, as the "hydraulic spring" (oil column) is under a constant tension (pressure).

It is therefore not possible to use the natural frequency of a system to describe the control accuracy and dynamic response of a secondary controlled system.

In principle any function of time can be used as an input value of an element or system in mathematical and experimental tests of the behaviour of elements of a closed loop circuit. There are two functions, however, that are of particular importance, namely the step function response and the sine function, as these are mathematically easier to solve and technically more feasible. The response part of the step function response is known as the transient response.

Transient response

Input value x_{input} is changed in steps and response $x_{response}$ of the element or of closed loop circuit is measured. The transient function measured is used to determine the time constants of the element (**Fig. 62**).

Frequency response method

The input value is sinusoidally excited at constant amplitude $\hat{x}_e$. In the transient state with linear systems the output value will also be sinusoidal, amplitude $\hat{x}_a$ and phase φ being dependent on the characteristics of the element (**Fig. 63**). The following equations apply to input and output signals:

$$x_e(t) = \hat{x}_e \sin \omega_t$$

and

$$x_a(t) = \hat{x}_a \sin(\omega_t + \varphi)$$

with

$\omega t = 2 \bullet \pi \bullet f$ as angular frequency.

If this test is carried out for all frequencies between zero and infinity, dependent on the frequency this will result in typical changes in amplitude and phase shift of the output value of the element.

Evaluation of the frequency response can be either in the form of a Bode diagram or locus curve (Nyquist diagram).

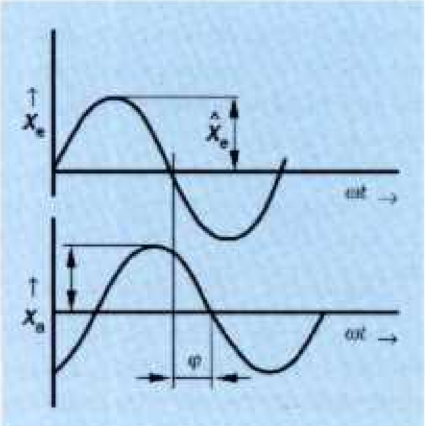


Fig. 63: Measurement of frequency response

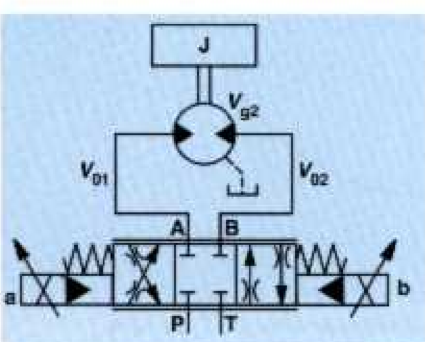


Fig. 61: Valve controlled hydraulic motor

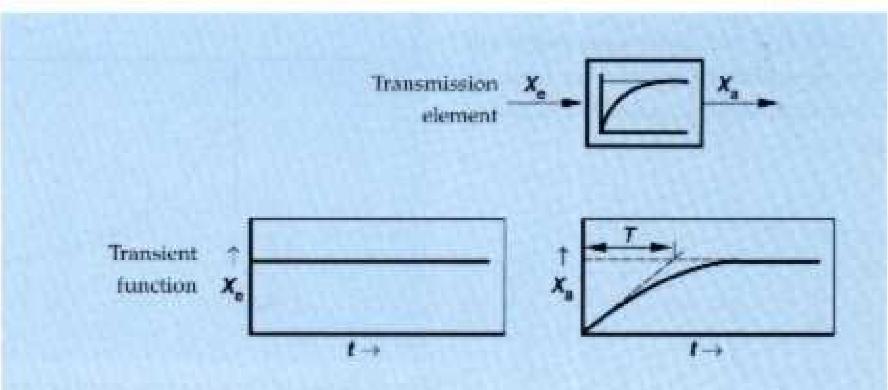


Fig. 62: Measurement of transient function (x_e = 1)

Bode diagram (frequency response characteristic)

Amplitude response

One possibility of representing the test results of frequency response is offered by Bode diagram (**Fig. 64**).

In a double logarithmic co-ordinate system the amplitude relationship $\hat{x}_a/\hat{x}_e$ is plotted as a function of the angular frequency.

Phase response

In a single logarithmic co-ordinate system the phase shift is plotted in the same way as a function of the angular frequency.

The logarithmic representation of the amplitude response

$$|F| = \left|\frac{\hat{x}_a}{\hat{x}_e}\right|$$

and phase response φ with a series circuit consisting of several transmission elements, of which the Bode diagram is a known example, offers the advantage that their behaviour can be determined by geometric addition of the individual amplitude and phase responses (**Fig. 65**).

The time constants are determined in the transient function of systems of the first order at a phase shift of -45° (ω_{input} = corner frequency = $\frac{1}{t}$).

From **Fig. 65** we can see that a system of the first order can attain a maximum phase shift of -90°. This angle corresponds to an amplitude relationship of zero and an angular frequency of

$\omega \rightarrow \infty$.

With P, T and I elements there will be a phase shift φ (negative φ), whereas D elements show a phase lead (positive φ).

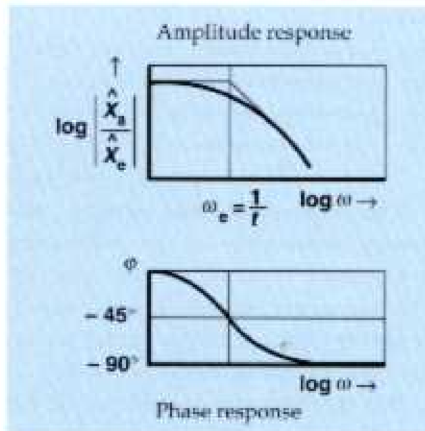


Fig. 64: Results of the frequency response method as Bode diagram for a P-T_1 element

Locus curve

By plotting the amplitude relationship against phase shift over the total frequency range on a complex plane (Gaussian numerical plane) by polar co-ordinates we obtain the locus curve for the test system. The application of the Gaussian numerical plane comes from the fact that the sine or cosine function can also be written as a complex function (Euler's theory).

Fig. 66 shows the locus curve for a system of the first order. The angle between the positive real axis (R_e) and the radial from the zero point to the point in the numerical plane represents the

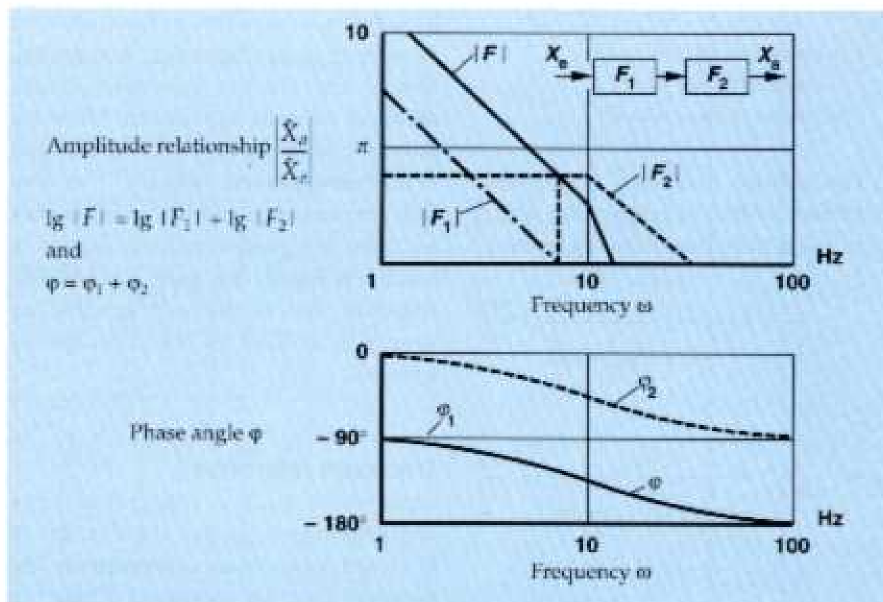


Fig. 65: Series circuit of two frequency responses in the Bode diagram

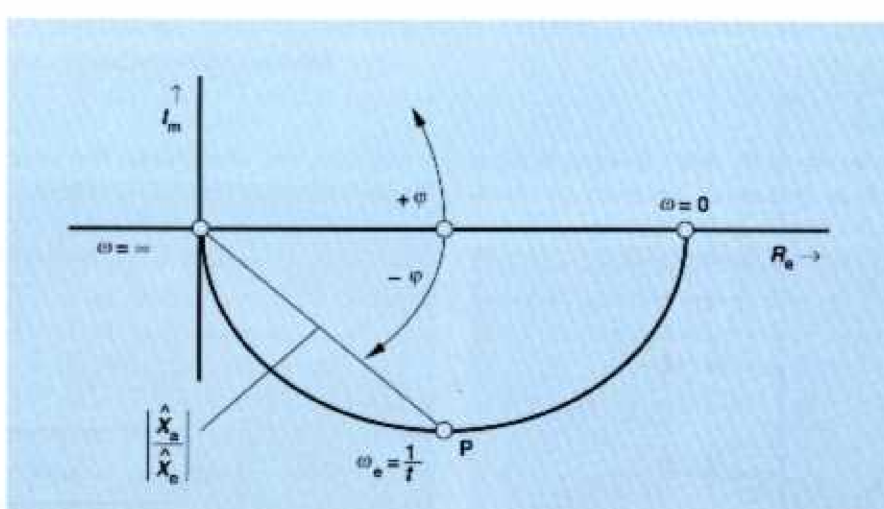


Fig. 66: Result of the frequency response method as locus curve for a P-T_1 element

phase shift, whereas the length of the radial represents the amplitude relationship.

The curve shown in **Fig. 66** describes the behaviour of a P -T_1 element.

From the locus curve we can also determine the time constants of the system at a phase shift of $\varphi = -45°$.

Fig. 67 shows the frequency response characteristic curve of the positioning cylinder of a secondary controlled axial piston unit type A4VSO71DS1, at an impressed pressure of 200 bar, with built-on servo valve with a nominal flow of 20 L/min.

The amplitude response is given in decibels (dB). The amplitude relationship will be

$$dB = 20 \bullet \log \frac{x_a}{x_e}$$

or

$$\frac{x_a}{x_e} = 10^{\left(\frac{dB}{20}\right)}.$$

The Bode diagram thus gives us a clear statement of the dynamic response of a closed loop circuit or of a complete drive system. We can see from this that at a constant input amplitude the output amplitude can change depending on the frequency.

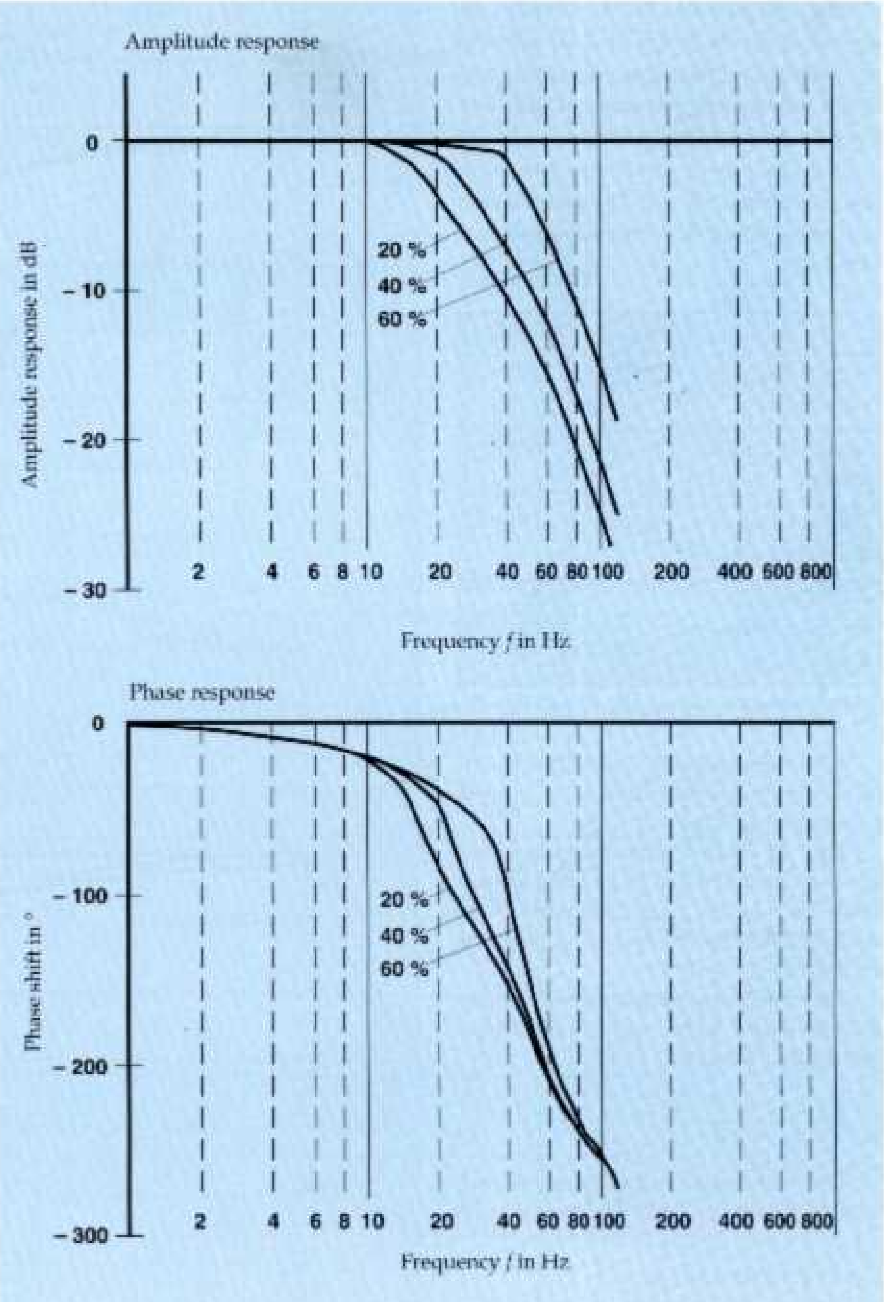


Fig. 67: Frequency response characteristic of the positioning cylinder of a swashplate axial piston pump A4VSO71DS1; Amplitude response top and phase response below

Fig. 68 shows the frequency response of types A4VSO250DS1 without any additional moment of inertia, and at an impressed pressure of 200 bar. Speed amplitude is ± 300 rpm.

The frequency response measurements in **Figs. 69** and **70** were carried out using the following technical data (**Table 3**):

	Fig. 69	Fig. 70
Operating pressure in bar	200	250
Speed amplitude in rpm	± 150	± 500
Natural moment of inertia in kgm²	0,0956	0,0956
Additional moment of inertia in kgm²	~1	~1

Table 3: Technical data to Fig. 69 and Fig. 70

All measurements were carried out in closed circuit operation with a boost pressure of 10.5 bar.

Fig. 71 shows the frequency response characteristic of an axial piston unit in swashplate design, type A10VSO100DS at an operating pressure of 250 bar.

The unit has a natural moment of inertia of 0.0167 kgm² and an additional inertia of 0.37 kgm².

$$\frac{J_{add}}{J_e} = \frac{0,37}{0,0167} = 22,16$$

The command value speed setting was ± 1000 and ± 2000 rpm respectively.

Curve progression 1 to 4:
1 $-\varphi(A_{comm} = 1000$ rpm),
2 $F(A_{comm} = 1000$ rpm),
3 $-\varphi(A_{comm} = 2000$ rpm),
4 $F(A_{comm} = 2000$ rpm).

The combination of a hydrostatic unit and electronics, measurement technology and closed loop control is a feature of secondary control. The successful application of this field of technology requires a comprehensive knowledge of a complete range of parameters and their influence on overall system behaviour.

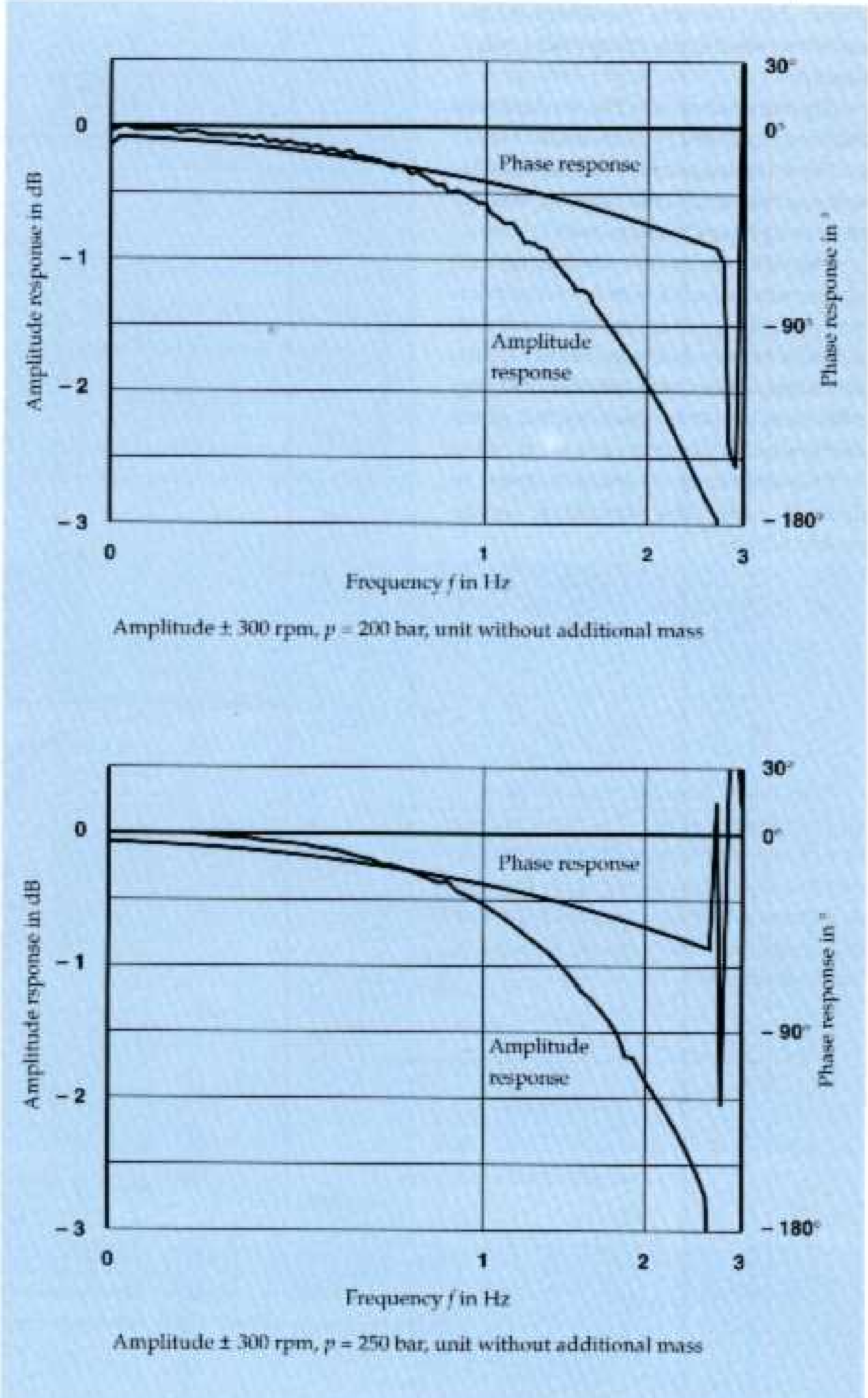


Fig. 68: Frequency response characteristic of a swashplate axial piston pump type A4VSO250DS1

Close observation of the basic physical laws are an important part of closed loop control technology, allowing the optimum choice of equipment and accurate predictions of the precision and dynamic responses that can be expected.

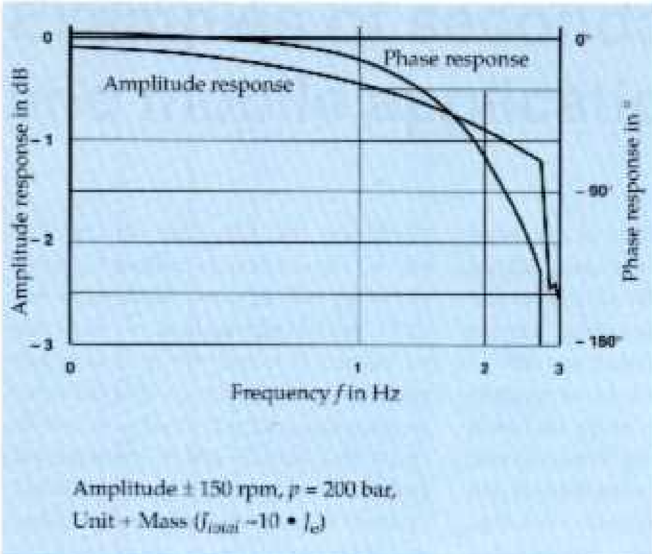


Fig. 69: Frequency response characteristic of a swashplate axial piston pump type A4VSO250DS1

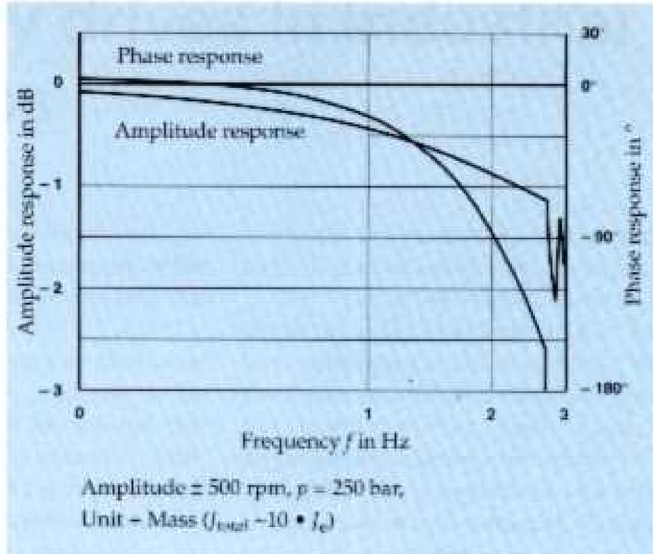


Fig. 70: Frequency response characteristic of a swashplate axial piston pump type A4VSO250DS1

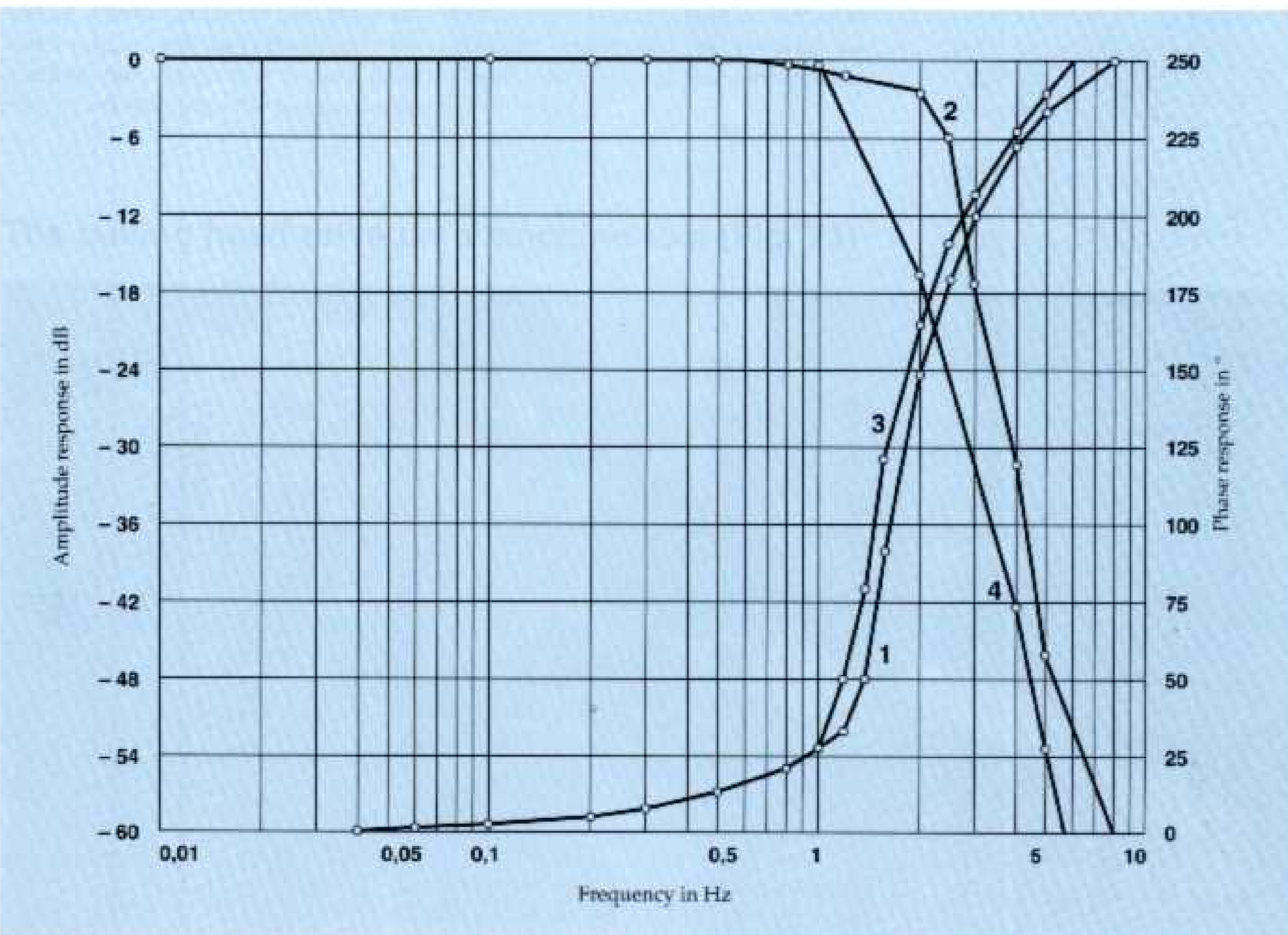


Fig. 71: Frequency response characteristic of a swashplate axial piston pump type A10VSO100DS

Examples of secondary drives in industrial and mobile applications

Back in the early 80s the seemingly complex control characteristics of the secondary drive concept was generally unknown in hydrostatic circles. This was, however, not the only reason for the reluctance to implement secondary control. In spite of its many advantages, development of the digital and adaptive electronics necessary for secondary control was neglected or rather continually postponed. A complete high dynamic response drive system could therefore not yet be offered.

In the initial stages there was no choice but to wait for the right combination i.e. a project engineer who believed in the product and an enthusiastic prospective user. Important factors here were to acknowledge the technical risk involved, to make allowances for it and for the customer to assume some of the responsibility when implementing this system technology.

We are indebted here to those few engineers who were prepared to go ahead in spite of the knowledge that there were no previous references. With the co-operation of these engineers an optimum electronics system was developed for the respective application.

It is therefore not surprising that the first applications of secondary control were in special machines.

This limiting of applications to individual drive concepts that, on technical grounds, could not be covered by closed loop electrical and conventional hydraulic drives, initially had a detrimental effect on component costs.

The use of secondary controlled drives in mass production today has been achieved thanks to a newly defined standard that has come into practice. It still, however, depends mainly on the person and the application.

The following application examples should make it easier for the potential user in his initial implementation of secondary controlled hydrostatic units in the field of high dynamic response drives.

The milling head drive on a machine tool (Fig. 73)

Fig. 72: CNC machining centre

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 70 kW under secondary control (**Fig. 73**).

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. 74**). It may be slewed separately about its vertical axis. This permits the tool to be positioned at any angle and corner position with respect to the workpiece axis.

For physical reasons the tacho-generator for speed control could not be mounted on the through shaft of the hydraulic motor. It was therefore driven by means of a toothed belt on a parallel axis (**Fig. 74**). The torque of 275 Nm necessary to drive the head was produced by an A4V axial swashplate unit of 90 cm^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 and secondary units of approx. 10 metres due to the hydraulic spring being eliminated by virtue of the impressed operating pressure. An additional 10 litre accumulator was installed directly at the drive in order to cater for short term energy peaks required at this point.

Another important factor with this hydrostatic drive is the extremely low weight and compact size of the unit. A D.C. motor of similar power could not have been installed within the same available space.

The main advantage, however, is the high cutting and chip removal capacity which could be achieved even at low speeds, thus considerably reducing production times.

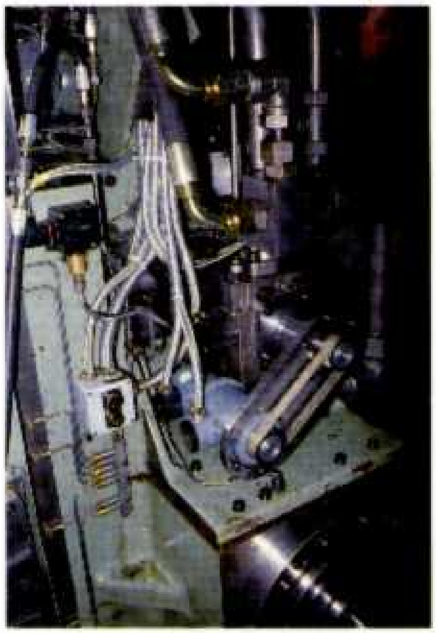
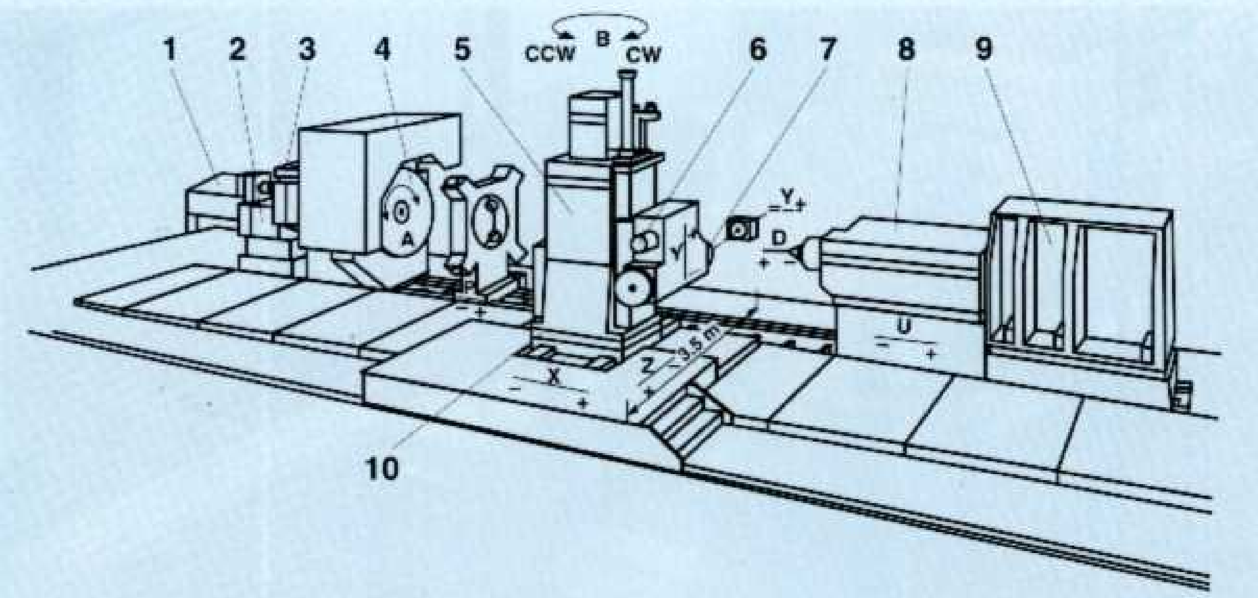


Fig. 74: Drive motor in the multi-tower



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 palett systems
10 Cross slide

Fig. 73: Overall view of the CNC machining centre

A drive for a roughing mill

It has also proved possible to utilise 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 subjected neither to compression nor tension, as otherwise the quality would be reduced. This therefore places high requirements on the dynamics of such drives. **Fig. 75** 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.

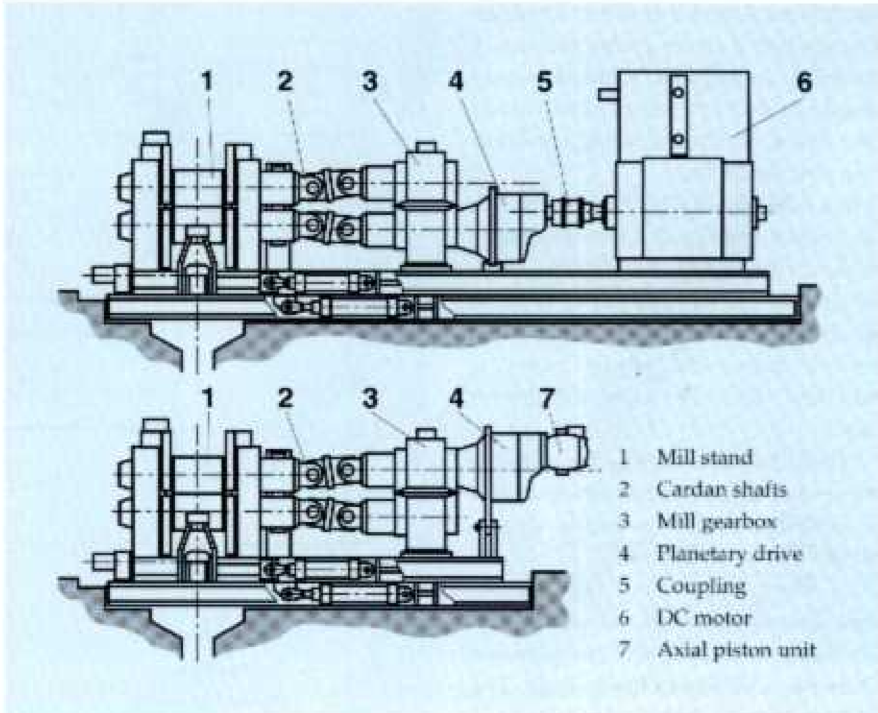


Fig. 75: Roughing mill stand with electrical drive (top) and hydraulic drive (bottom)

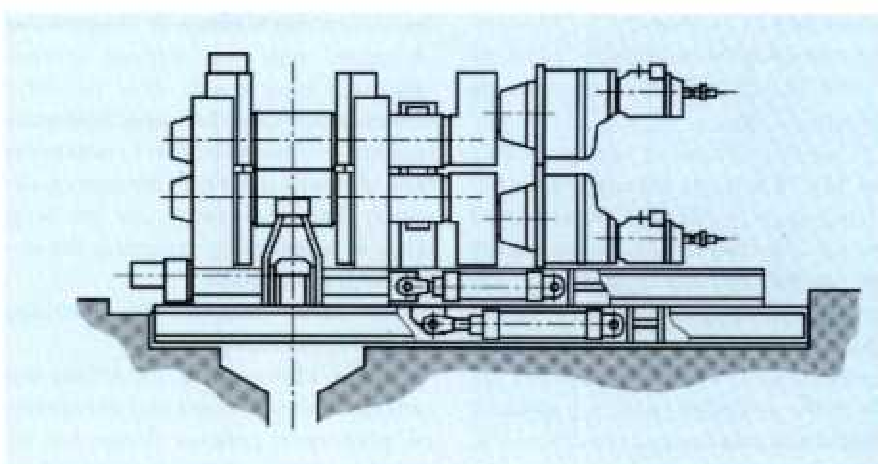


Fig. 76: Mill stand with full hydraulic drive

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 D.C. 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.

It is therefore not surprising that the hydraulic drive was hardly any cheaper, if at all, than its electrical counterpart.

If, however, one utilises 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. 76**).

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 utilise the accepted definitions of dynamic characteristics for speed of D.C. machines following a stepped load change to VDI/VDE guideline 2185.

A measure of the error influences and the quality of rolling, stemming from the overriding change of speed when the torque suddenly changes, is the control envelope $A_1 \bullet t_{settle}$ of the speed control loop (**Fig. 77**).

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

The control envelope is the 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 characterised by the ratio between the amplitude of the first and second control values A_2/A_1, 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 pre-set speed. As the settling time must be less than 200 msec, and if possible considerably smaller, a drive under secondary control is initially at a disadvantage compared with a conventional hydrostatic drive, as the following consideration will make clear.

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. 78** top). Under idle running conditions the pressure differential at the motor is small and the rolls run at the pre-set speed.

When the billet enters the rolls the torque rises almost instantaneously and the pressure in the hydraulic system will also rise proportionally.

As the speed during this process must remain constant, the pump will only need to deliver the compression volume in addition to the leakage. The swivel angle will change accordingly.

This process is very short, being over in 15 to 20 msec. The system can then start to oscillate due to the effect of the hydraulic spring. In practice it must be damped to prevent this i.e. the control operation at the pump must be slowed down, which has an effect on the settling time.

Under conditions of secondary control (**Fig. 78** 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 impressed operating pressure is available and the hydraulic spring is therefore under constant tension. The swivel angle of the secondary unit has adjusted itself to suit idle running conditions e.g. 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 e.g. 12°.

For this operation an A4VSO500DS1 axial piston unit under secondary control requires approximately 60 msecs, with a corresponding effect on the rolling technology.

This apparent disadvantage is easy to overcome, however, by means of a trick circuit (**Fig. 79**). A 2-way cartridge valve and an electrically adjustable pressure reducing valve are built into a control block mounted directly onto the pressure flange of the axial piston unit.

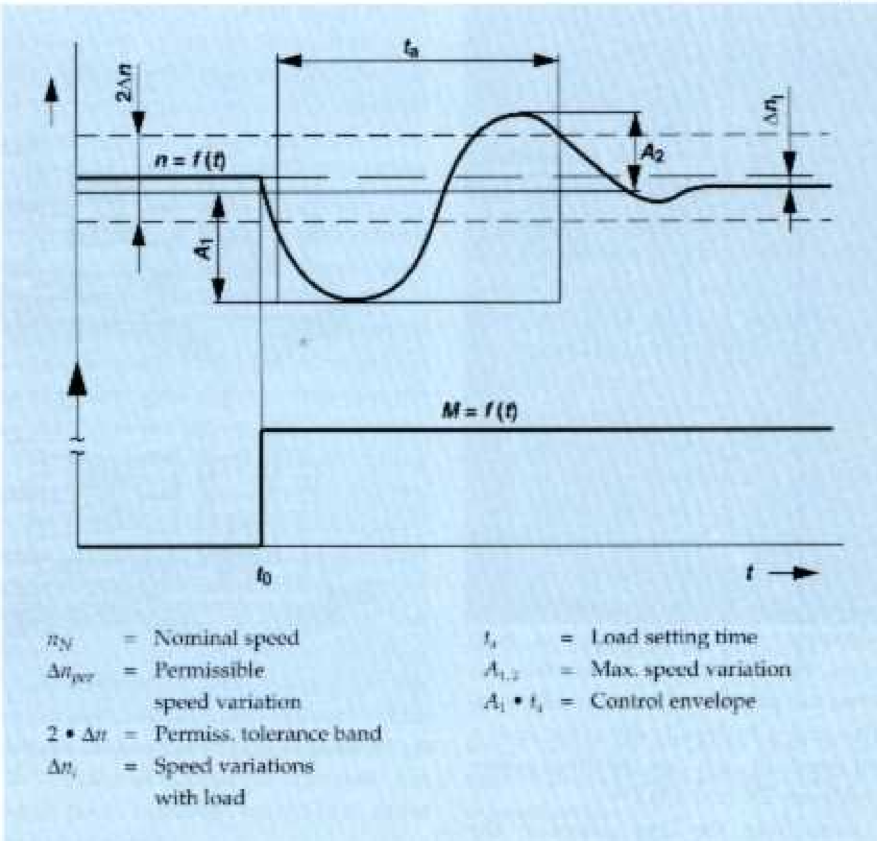


Fig. 77: Control variation with a stepped change in torque

The cartridge valve acts as a "hydraulic isolator" in order to be able to isolate the flow of energy during an emergency situation. The axial piston unit can only work as a generator, returning the energy to the ring main.

For safety reasons this cartridge valve is always installed.

Under idle running conditions the cartridge valve 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 increased to a value close to the steady state conditions which occur when a billet is being rolled, this angle being known beforehand, so that the time loss for the change in swivel angle when a torque shock occurs is compensated for. In this case the valve simply needs to be operated and the displacement of the axial piston unit need only be changed by a small amount.

Any variations from the pre-set value can be determined and can be included in an adaptive open or closed loop control process.

The losses associated with this "trick circuit" when the 2-way cartridge valve is closed during the inlet phase of approx. 500 msec are acceptable.

Another possibility, that of overriding a swivel time which is too long for a roughing mill, specifically with axial piston units having a large displacement, is shown in **Fig. 80**. This uses a tandem unit, both having the same displacement and a control unit that can be by-passed via a valve. Depending on the material to be rolled the control unit is set to a specific swivel angle which then remains constant.

Parallel to the valve both machines operate as motors, the effective direction of the torques being added together. This process stops during the rolling process.

After the billet has left the rollers the drives only need to produce the idling torque at the same speed. The secondary unit will thus reduce the swivel angle to a negative value, because when for example, two 1000 cm^3 axial piston units are used, an idling torque can be produced from approx. 150 cm^3 at an impressed pressure.

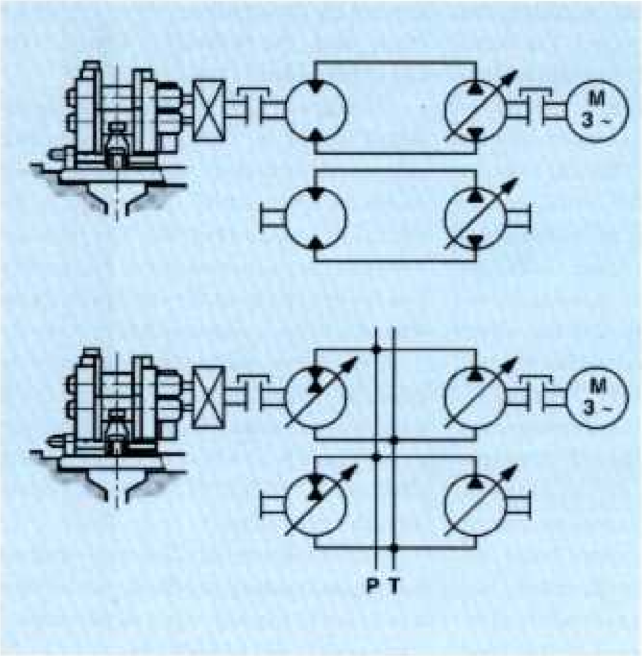


Fig. 78: Conventional drive (top), secondary controlled drive (bottom)

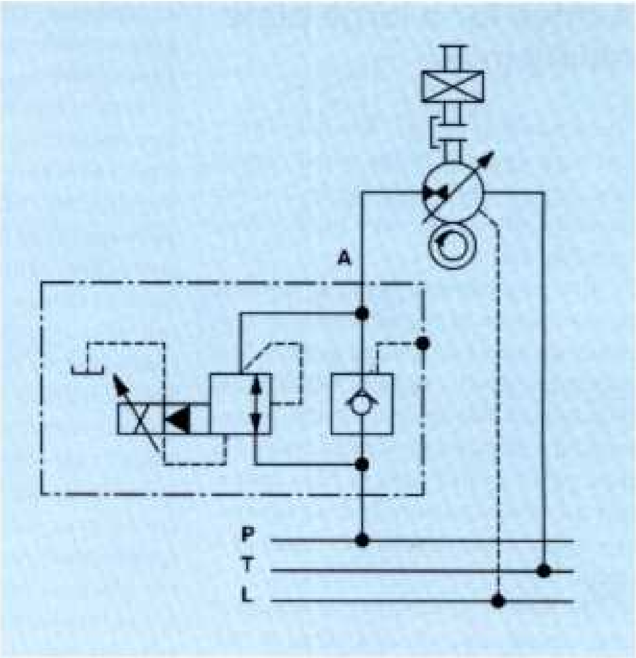


Fig. 79: Auxiliary circuit for a rolling mill drive

Shortly before another billet enters the valve is brought into the crossover position. The control unit then works as a hydraulic pump in opposition to the impressed operating pressure. The idling torque and drive torque for the hydraulic pump are now only required by the secondary unit. The swivel angle of the secondary unit thus assumes a value close to that required by the rolling process. The billet run-in automatically triggers the switching process, and both units then change over to motor operation.

The roughing mills in operation, to which this circuit principle is being applied, have proved the effectiveness of this trick circuit. Although the by-pass valve has to travel the full stroke, which is effected in 8 to 10 msec, torque generation will be within the range of 20 to 40 msec as specified by the manufacturer.

The pressure shock associated with the by-pass process is kept within limits and has no negative effect on the rolling operation.

Fig. 81 shows such a tandem unit consisting of two 1000 cm^3 units, with a circuit in accordance with **Fig. 80**. Behind the gearbox as the first unit is the control unit to be switched, and coupled to this is the secondary controlled variable unit with the electrical tachogenerator. It is obvious that the power density for this size unit cannot be achieved with closed loop controlled electrical units.

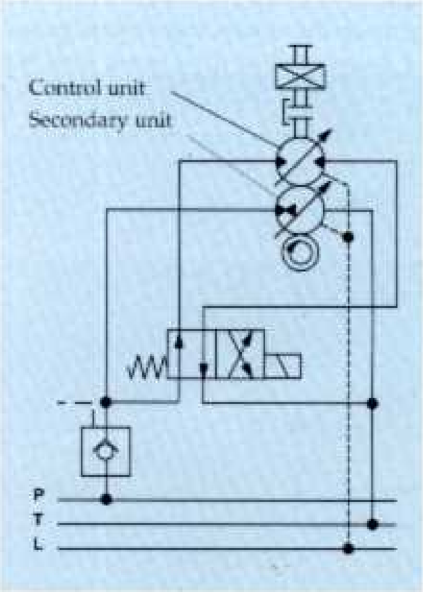


Fig. 80: Auxiliary circuit for a rolling mill drive

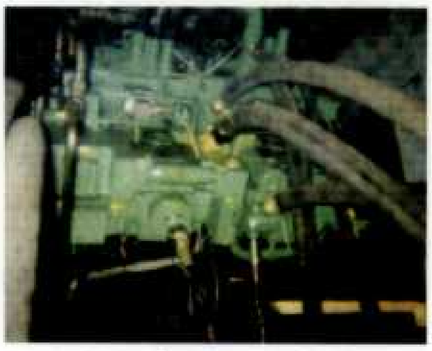


Fig. 81: 1000-cm^3 tandem unit

A drive for a large plate rolling mill

The advantage of high speed accuracy and high response in the speed control loop has also been proved in a roller conveyor drive of a large plate rolling mill (**Fig. 82**).

The requirement was to operate twelve motors of various sizes in synchronism under secondary control. The maximum speed variation, which must be achieved 500 msec after the occurrence of a disturbance torque, was ± 1.5 rpm for all twelve motors. This meant that all twelve motors had to operate absolutely within a tolerance band of 3 rpm.

The motors, without any mechanical inter-connections, drive a total of 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 taking into account the torque shocks to be expected. The computer results were good and the technical risks could be foreseen. It was therefore decided to install standard motors type A4VSO on the mechanical side. In this case the analogue D.C. tacho-generators had to be replaced by incremental generators. However, there was no need 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/VTS0102 was required (**Fig. 83**). All the drives were operated by a single, common command speed value. As the tolerance band did not place an excessive requirement for speed regulation on the secondary control, this was ideally suited to this type of operation, and expensive synchronisation closed loop controls were not required.

This roller operation was originally equipped with electrical drives which could not fulfil the synchronisation requirements. This in turn led to a 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 114 m, distance between primary and secondary units could be up to 80 m. The central oil supply is installed in a cellar (**Fig. 84**). 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 five pressure compensated A4VSO125DR axial piston units. The drive power per power unit is 55 kW and the operating pressure is 160 bar. The system is operated in open circuit.

With a system 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 ,
Max. weight per plate:15 tonnes.

The speed of the units is adjustable between 0.25 and 15 m/min, corresponding to total speeds of between 34 and 2050 rpm.

Fig. 82: Heating oven line with roller conveyors, total length 114 m

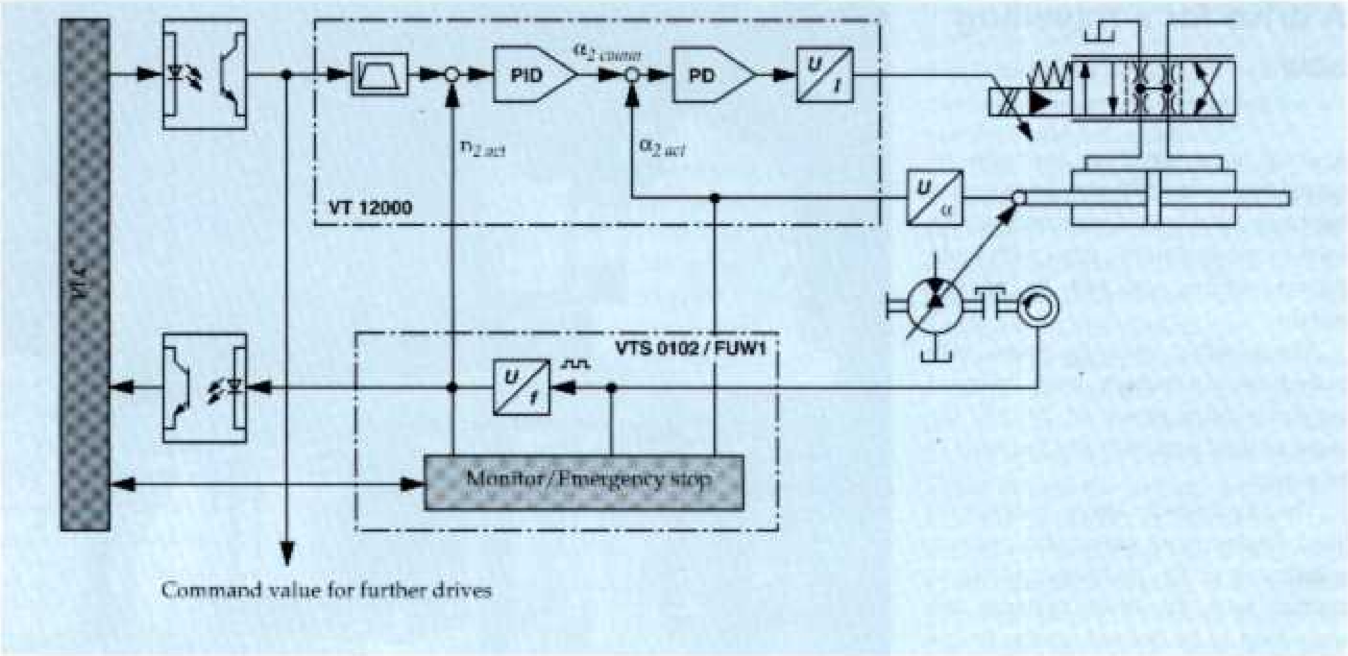


Fig. 83: Combination of standard control electronics VT12000 and f/U converter and monitoring electronics VTS0102

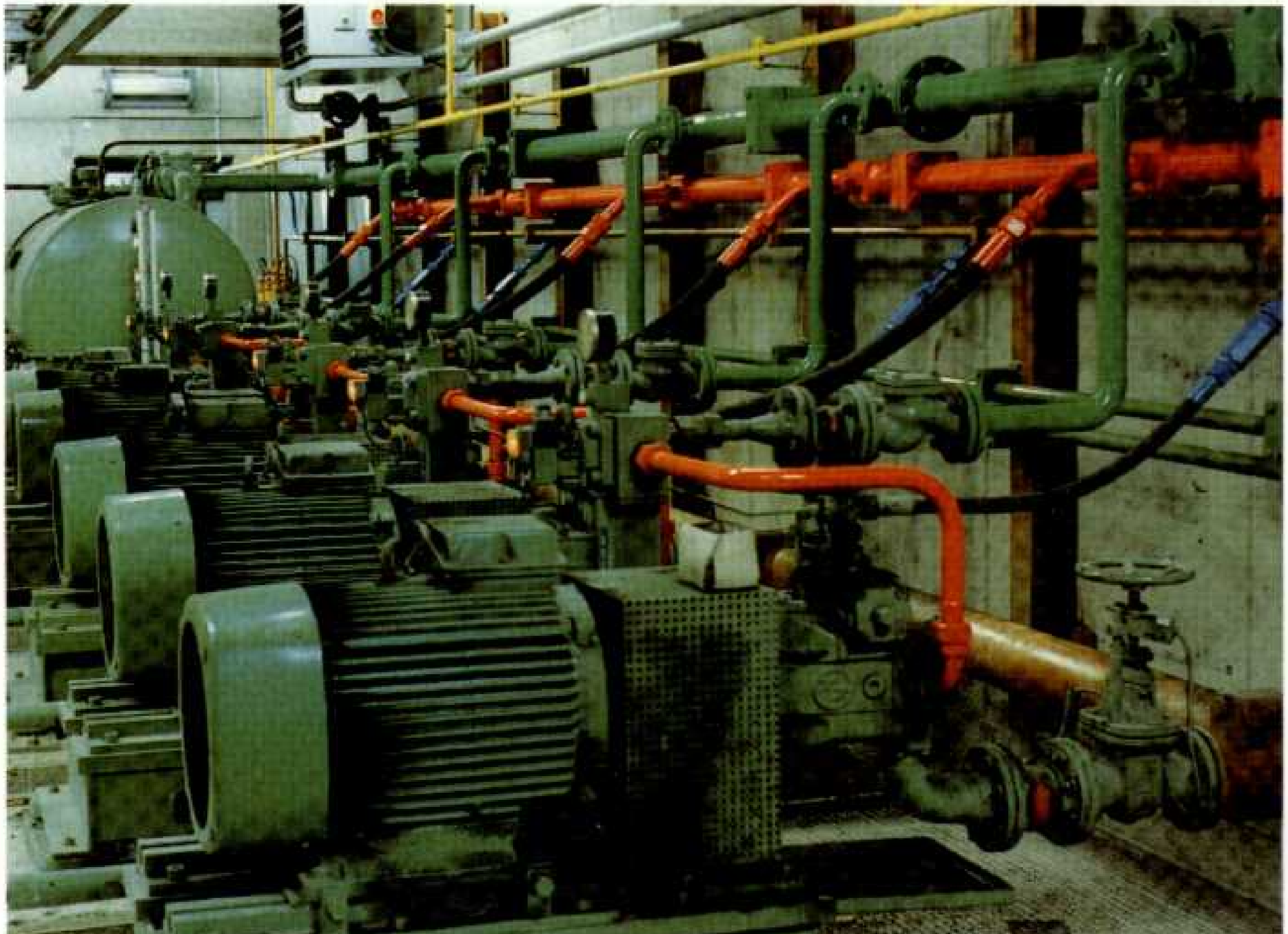


Fig. 84: Central oil supply

A drive for a travelling saw

A drive for a travelling saw was required for a continual non-ferrous casting plant , that would guarantee a high level of speed stability even with interrupted cutting, yet still be of low weight.

The possible variants, speed controlled electric motors, were no good beyond a drive power of 100 kW because of their large size and the mass to be moved.

The hydrostatic drive concept of fixed displacement motor and primary adjustment of the hydraulic pump, as applied with the older systems, left something to be desired, as the torque variations transmitted from saw to motor led to considerable variations in speed.

Although secondary control met initially with some opposition, it was eventually realised that the disadvantages of conventional hydrostatic drives would then be eliminated.

Two years of experience have now proved that the decision in favour of secondary control was the correct one.

Fig. 85 shows the complete saw unit. **Fig. 86** shows the basic circuit diagram. The speed deviations were well within the permissible range. Even with a large radius of the saw blade the speed remains virtually constant over the whole cutting speed range.

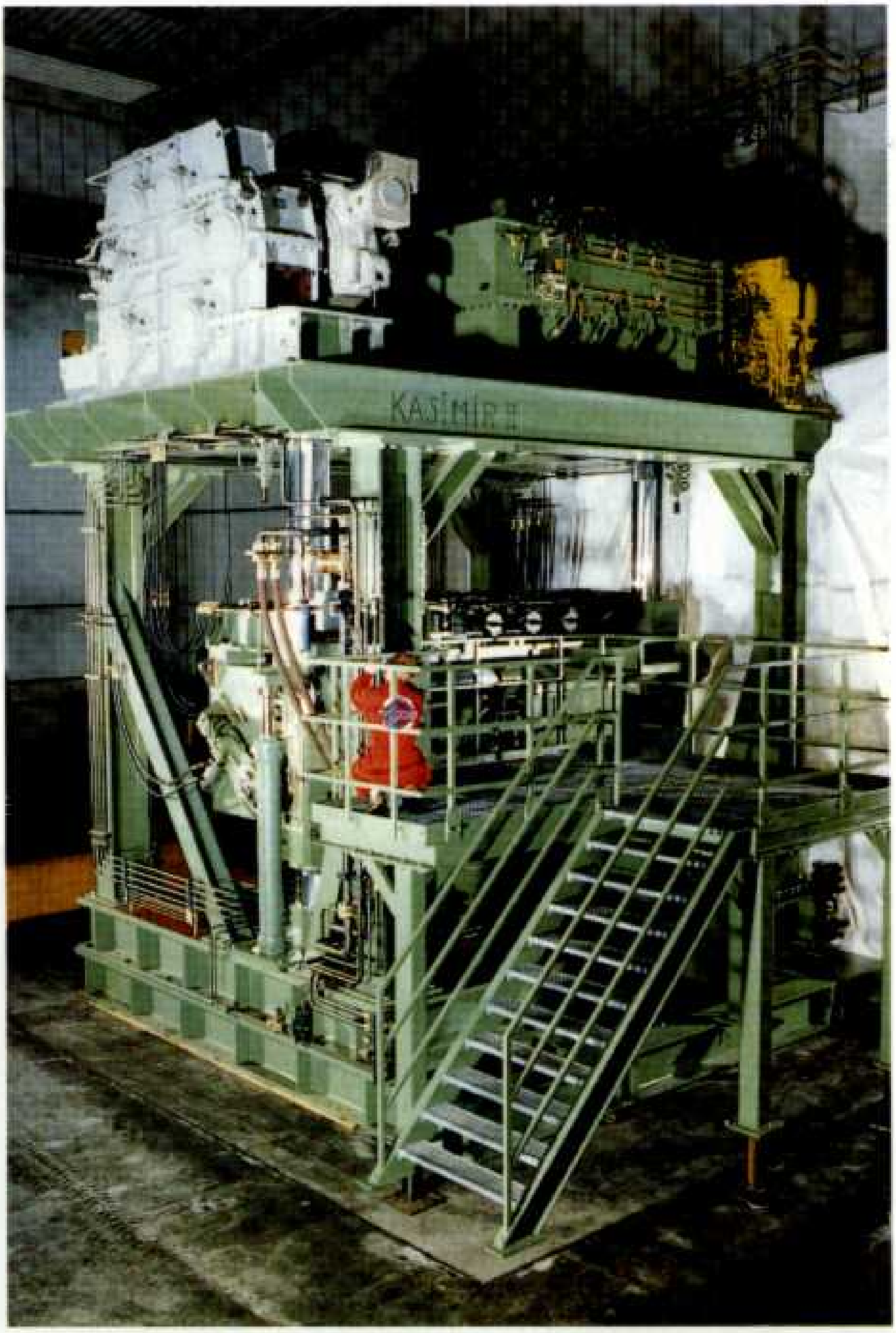


Fig. 85: Overall view of saw unit

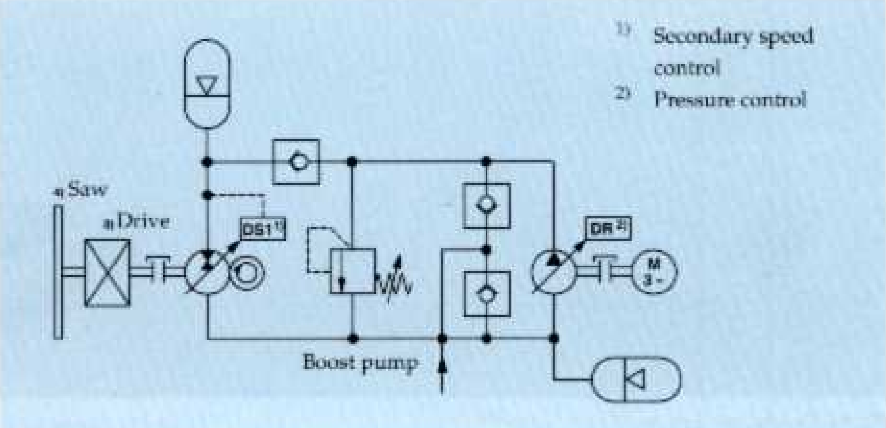


Fig. 86: Basic circuit diagram of the travelling saw

Mobile deep drilling installation

When we see a drilling rig, especially on the northern plains of Germany, we immediately think of oil. Drilling for oil in Europe has increased considerably, as the example of the North Sea demonstrates, and has met with some success. The quantity, however, cannot be compared with that produced by the USA or the Middle East.

What has, however, been considerably more successful in Europe, is the search for natural gas, which is just as much in demand as oil.

The most important gas-producing formations in Germany are concentrated in the North German Basin, as the geological conditions were particlarly favourable here for the formation of natural gas. There is natural gas in Teufen at a depth of 2000 - 5000 m.

The most economical method of drilling into the gas-producing formations, often kilometres away, is to use a moveable, medium-heavy drilling system, designed for use with a maximum lifting load of 240 tonnes at a mast height of 51 m (**Fig. 88**). The power of both 625 kW hydraulic motors is transferred by hydraulic means to the individual actuators. The idea of hydraulic power transfer in drilling platforms is not new. The conventional hydrostatic drive used up to now has certain failings, which have a noticeable detrimental effect on drilling operations. There are, for example, problems when changing over from holding a load to the lowering or lifting movements. The spring effect of the oil column prevents accurate movement, and the power distribution over several oil circuits, all of which had to be designed to withstand maximum power requirement of the actuators, could not be satisfied.

Thus secondary control was also chosen for this case, as the stipulated requirements with respect to dynamic response, positioning accuracy and stepless power distribution can be best fulfilled with this technology.

The high speed response associated with the low moment of inertia permits fast adjustment of disturbance values i.e. the speed will then remain mainly stable during the drilling process and will not be subject to variations. Load control on the drilling line is thus more easily controlled. Braking is by hydraulic means only, the braking energy being passed on to the other actuators. The load can remain suspended as long

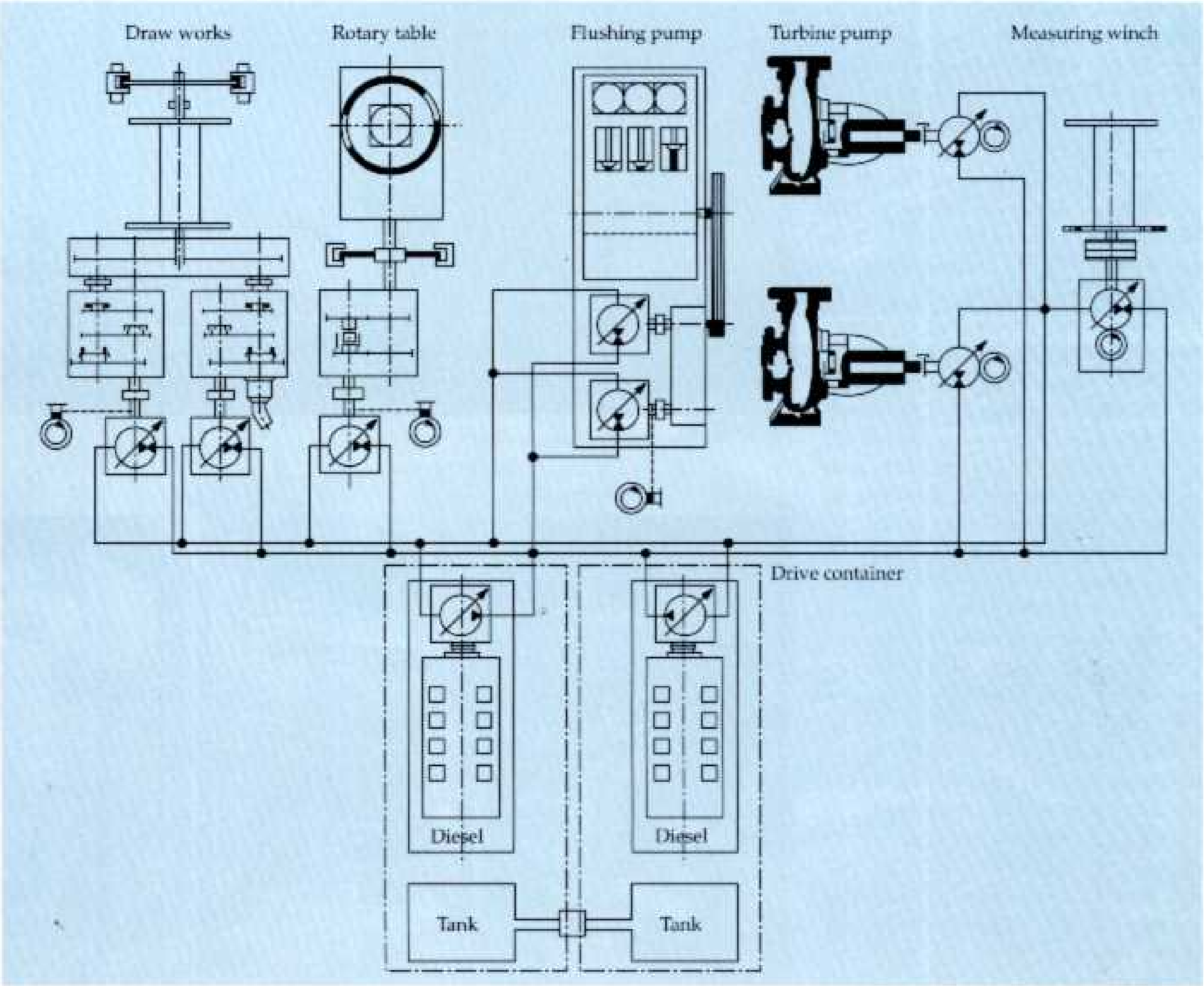


Fig. 87: Basic circuit diagram of the drilling system

as required without the need for another braking system.

This operating mode is not possible with diesel mechanical or diesel electrical drives. These must have an additional brake, in the form of a mechanical lever which has to be operated manually.

Secondary control is the first step towards the further automation of drilling operations.

Fig. 87 shows the basic circuit diagram of the hydraulic system with draw works, rotary table, flushing pump, turbine pump and measuring winch actuators.

The powerful draw works, rotary table and flushing pump are driven by single or tandem 1750 cm^3 displacement axial piston pumps in bent axis design.

Although the use of bent axis design in secondary control is an exception, because the cylinder drum is not shaft driven, but by means of the piston rod, there are no technical reasons why this method should not be used, as the torques act in one direction only and are not reversred.

The only exception is with the draw works, when the empty bucket is lowered. A high dynamic response is then not required. Due to pressure coupling between the primary side and actuator the diesel motor power may be generously distributed as required.

As the total power requirement of the actuators is greater than the available diesel motor power, priorities are set electronically. The speed of the diesel motors can be pre-set according to the power requirement. The actual potential energy produced when the drill pipe is being sunk is passed on to the other actuators. This process also influences the diesel motor speed. Moreover a drive container (**Figs. 89** and **90**) can be switched completely off and the second remaining drive unit operated at reduced speed. In spite of this all the actuators can be run idle, as long as the operating pressure is kept constant.

If the drill pipe is lowered, for example after changing the drilling bit, this will result in a minimum of power being required. The reason for this is, that in this operating mode, the energy recovery increases in proportion to depth, and only the bucket needs to be raised.

Fig. 88: Mobile deep drilling installation

Fig. 89: Drive container

In this case there is only one container in operation, running at low speed. Simultaneously the diesel motor will switch off a series of cylinders and then operate as a compressor against atmospheric pressure.

Due to the mobility of the deep drilling installation all actuators and power units are accommodated in standardised containers. **Fig. 91** shows at the top both drive containers with the silencers for the diesel motors fitted, and underneath the drive for the main winch.

Drilling is carried out in four shifts operating 24 hours a day. There are four men working on each shift together with a shift foreman.

The complete installation is designed explosion-proof.

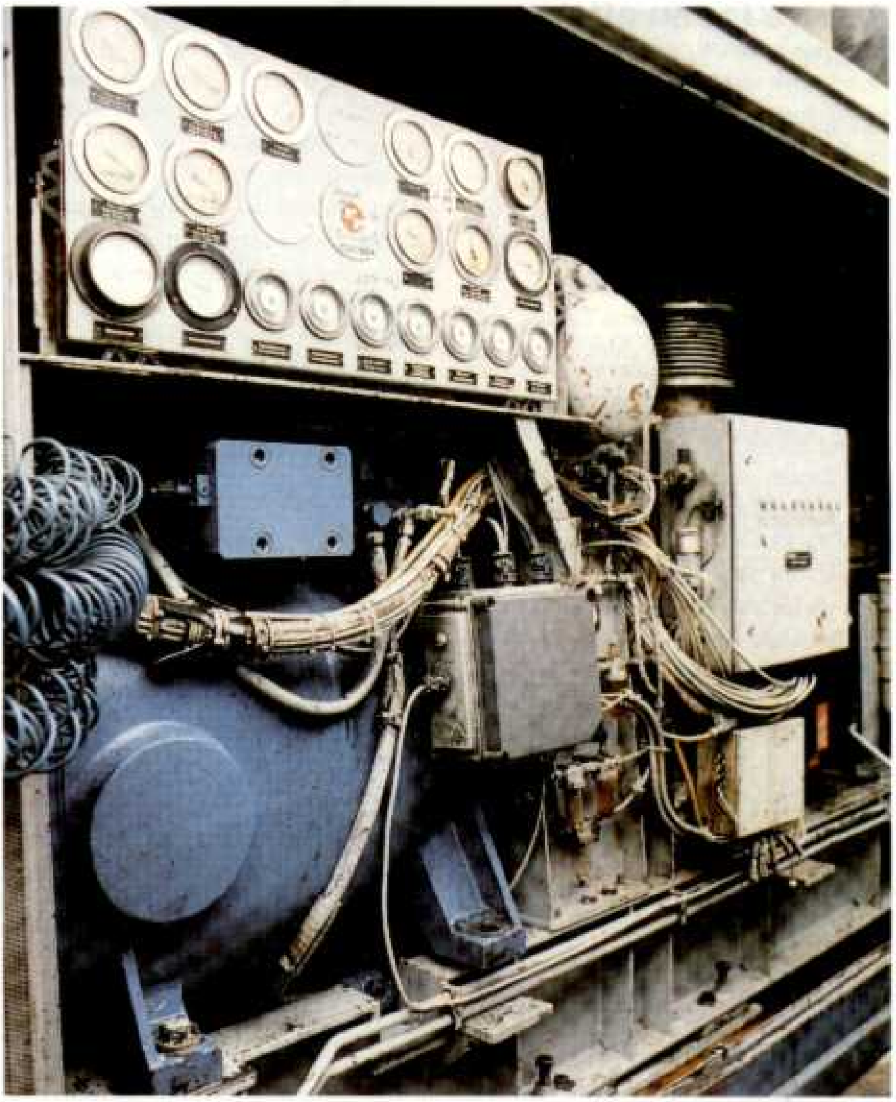
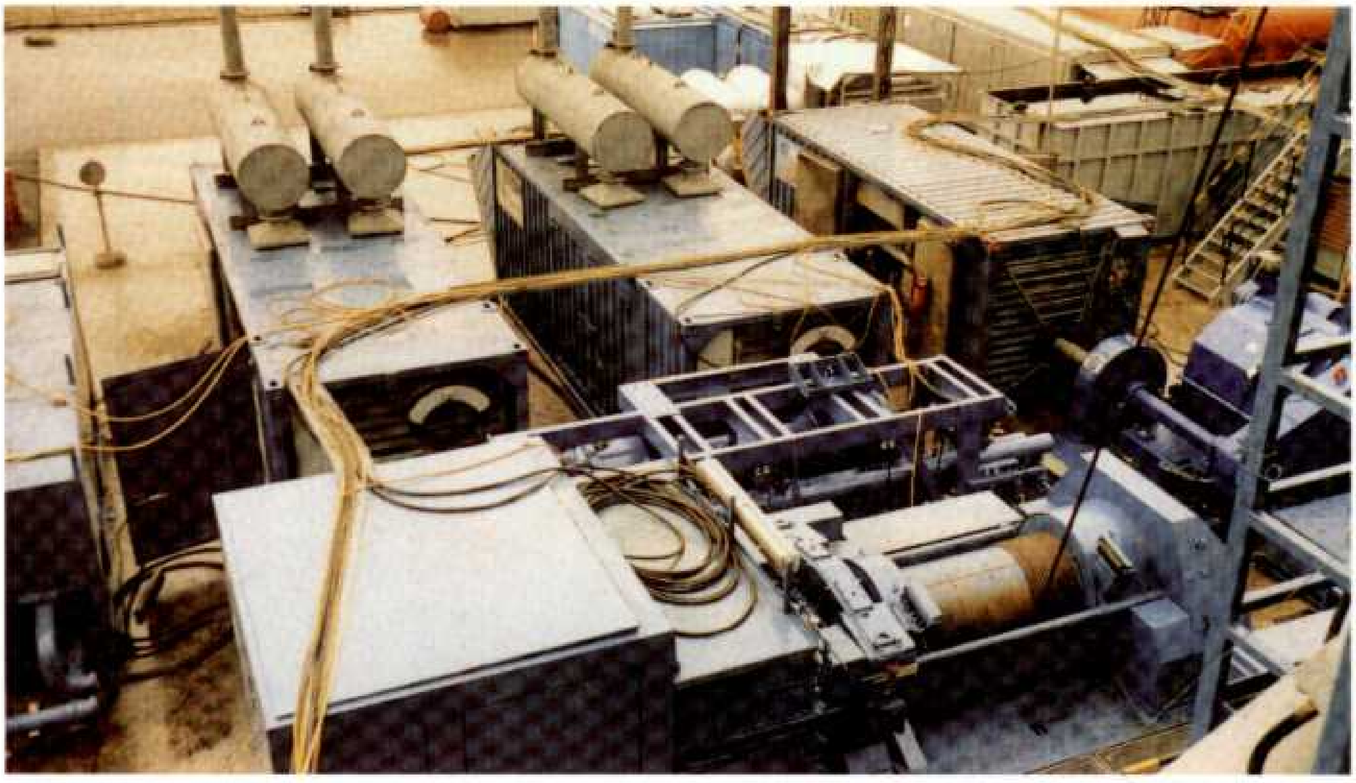


Fig. 90: Drive power unit (right)
Fig. 91: Container with main winch drive



A drive for an offshore crane

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

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

A Dutch company has developed a new generation of off-shore cranes for these tasks, which enable work to be carried out to a depth of 400 metres, even under adverse weather conditions.

Hydraulic drives with secondary control are used to fulfil the high requirements placed on offshore cranes.

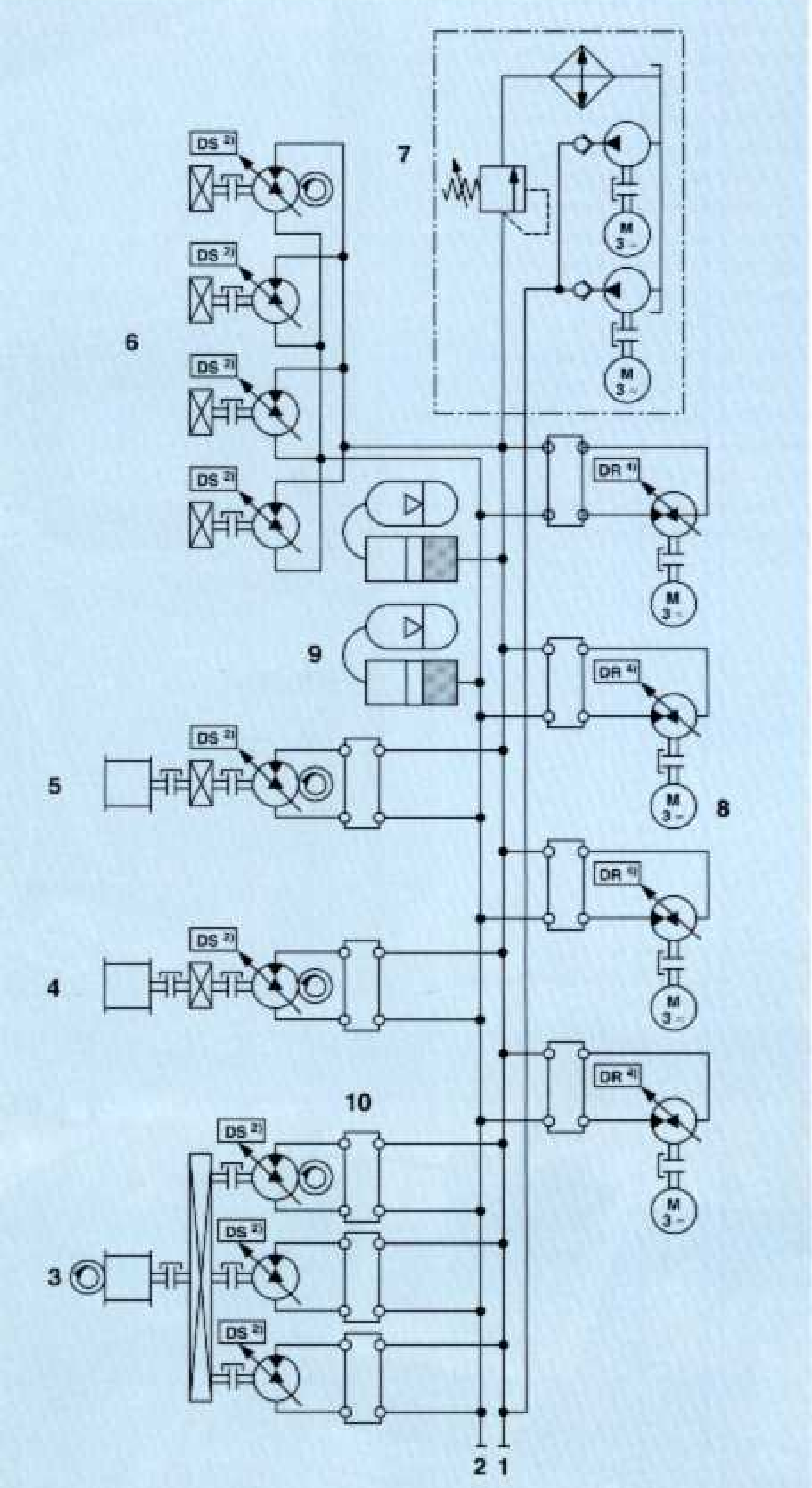
The decisive factors here were:

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

Figs. 93 and **94** show three 700 kW crane installations on a pipe layer and two semi-submersibles. How these and the following installations have not only fulfilled their tasks, but have also demonstrated their reliability under continuous operation, is explained below.

The hydraulic system (**Fig. 92**) 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 and on the speed required. The reason for this is that a secondary unit must receive a flow of oil under these conditions. If the tank is set at a much lower level than



1 Low pressure line
2 High pressure line
3 Main winch
4 Luffing winch
5 Auxiliary winch
6 Slew mechanism
7 Boost pump station
8 High pressure pump station
9 Accumulator station
10 Logic block

2) Secondary speed control
4) Pressure control

Fig. 92: Hydraulic system of an offshore crane

the secondary units, a closed circuit or a pre-filled circuit must be used.

The following actuators were driven:

- the main winch,
- the auxiliary winch,
- the topping winch,
- the slew mechanism and
- the positioning winches.

The hydraulic system remains 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 the winch 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 positioning accuracy mean that the short reaction times and dynamic response of secondary control are essential.

The movement of the ship is measured by means of an accelerometer and converted to suitable movements of the derrick via a microprocessor. 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 operation permits the crane to be operated in winds of up to force 5. Due to the high power required, axial piston units series A2P1000HS, bent axis design with secondary control are installed with a tacho unit built onto the secondary drive.

The same applies here as with mobile deep drilling installations, that there is no technical reason why winching operations cannot be carried out with a torque acting in one direction, in spite of the cylinder drum being driven without shaft by means of the piston rod. Problem-free operation over a period of several years has justified this use.

With the new generation of cranes however, the new A4VS series in swashplate design is used, this having been extended in 1992 by the addition of the NG 1000.

The main advantages are the compact design and low weight, the easy installation of the tacho unit and low capital costs.

Fig. 93: 700-kW crane installation on a semi-submersible

Fig. 94: 3000-kN lattice tower crane on a pipe layer

Lattice tower crane TC 3600

The TC 3600 lattice tower crane was first introduced at the BAUMA exhibition in 1989 (**Fig. 95**).

This crane attracted attention, not only due to its extremely large lifting capacity, but also its quick assembly, the new system being comparable to that of the telescopic crane.

The technical data is as follows:
Max. load torque: 3600 kNm
Max. load capacity: 650 tonnes
Max. roller height: 180 - 190 m
Transport weight: 96 tonnes
Diesel motor power: 390 kW
Max. travel speed: 65 km/hr.

The derrick of the TC 3600 is constructed such that a minimum of three lattice poles for a derrick length of approx. 50 metres can be accommodated on one transporter.

If the load is 30 metres long the TC 3600, including extra parts, can be accommodated on the same number or even fewer vehicles than are required by a large telescopic crane. This is due to the lighter derrick being able to achieve the same lifting capacity with less ballast.

When using a derricking jib a basic derrick system is necessary that is rigid and stable as with the classic lattice tower crane. This is one of the reasons why the TC 3600 can support a load three or even four times heavier than a telescopic crane with derricking jib (**Fig. 96**).

Another special feature of the TC 3600 is the super lifting device. It imparts this new type of fast lifter with a load torque of almost 9600 mt. This high torque is achieved in an extremely short time, in approx. 30 minutes. With this new type of derrick no more space is required as with a telescopic crane which has to transport the derrick separately. The revolving superstructure is also new, consisting of a weight saving latticework with built-on roller revolving connection and the angular-mounted slewing section.

In addition to the many new mechanical features a modern drive system was also chosen, in the form of secondary control (**Fig. 97**).

Fig. 95: Lattice tower crane TC 3600 ready for operation

Fig. 96: Lattice tower crane TC 3600 in operation

By means of a power divider the diesel motor drives five pressure-controlled axial piston pumps of the series A4VSO as well as a boost pump for the closed loop circuit. The hydraulic pumps are grouped together in two groups of three. The actuators, the secondary controlled units for the lifting equipment, the luffing and slewing gear for the derrick and the hydraulic cylinders are all connected to a common pressure line.

This arrangement guarantees good system redundancy, as the actuators can continue operating, albeit at reduced power, if one hydraulic pump breaks down.

The energy recovered when a load is being lowered or when the slew is decelerating (the secondary controlled axes then working as a generator), will be either supplied to the other actuators or will support itself for up to a third of the nominal power on the diesel motor. Another decisive factor in the choice of secondary control was the ability to travel accurately to the millimetre, to position accurately and to hold the load without mechanical brakes.

The hydraulics and open and closed loop controls are closely linked to each other in order to meet the continually increasing demands with regard to easy operation and manufacturers' safety requirements.

The experience gained from the use of mobile harbour cranes and the knowledge of data processing and electronic control technology was put into practice with lattice tower cranes.

The program will send error messages in the event of excess temperature, breakdown of limit switches, overspeeds, cable break, short circuit and over acceleration and will display swivel angle and speed differentials.

There is an integral supporting force analysis and calculation, which places the system in a position to suggest to the crane driver the solution to the given load. On input of the required combination the calculator offers suggestions as to how to achieve this, using data such as length and pressure of derrick, and lengths of main derrick and derrick jib.

The classic overload protection is redundant, supported by the calculator. The limit switches are also redundant. If the switch fails an error message will be sent, but the crane will continue operation. This applies to all error messages. Even a breakdown in one section of the electronics will not necessarily lead to an interruption of operation, as the equipment can revert from one operating mode to a lower one, still retaining the open and closed loop control operations. Emergency lowering equipment is therefore not necessary.

Secondary control also means, however, synchronisation of the winches. At any time it is possible to switch to synchronisation, even if one hook is travelling upwards and the other downwards. By pressing a button both winches will travel in synchronisation yet independently of each other, regardless of how often the winch reeves or how much cable is on both drums.

The information that the crane driver receives is already sorted according to danger class i.e. he receives on his monitor only that data which is relevant to the specific application. He is no longer confronted with a multitude of flashing control lights and he can thus concentrate on the sequence of movements to be carried out.

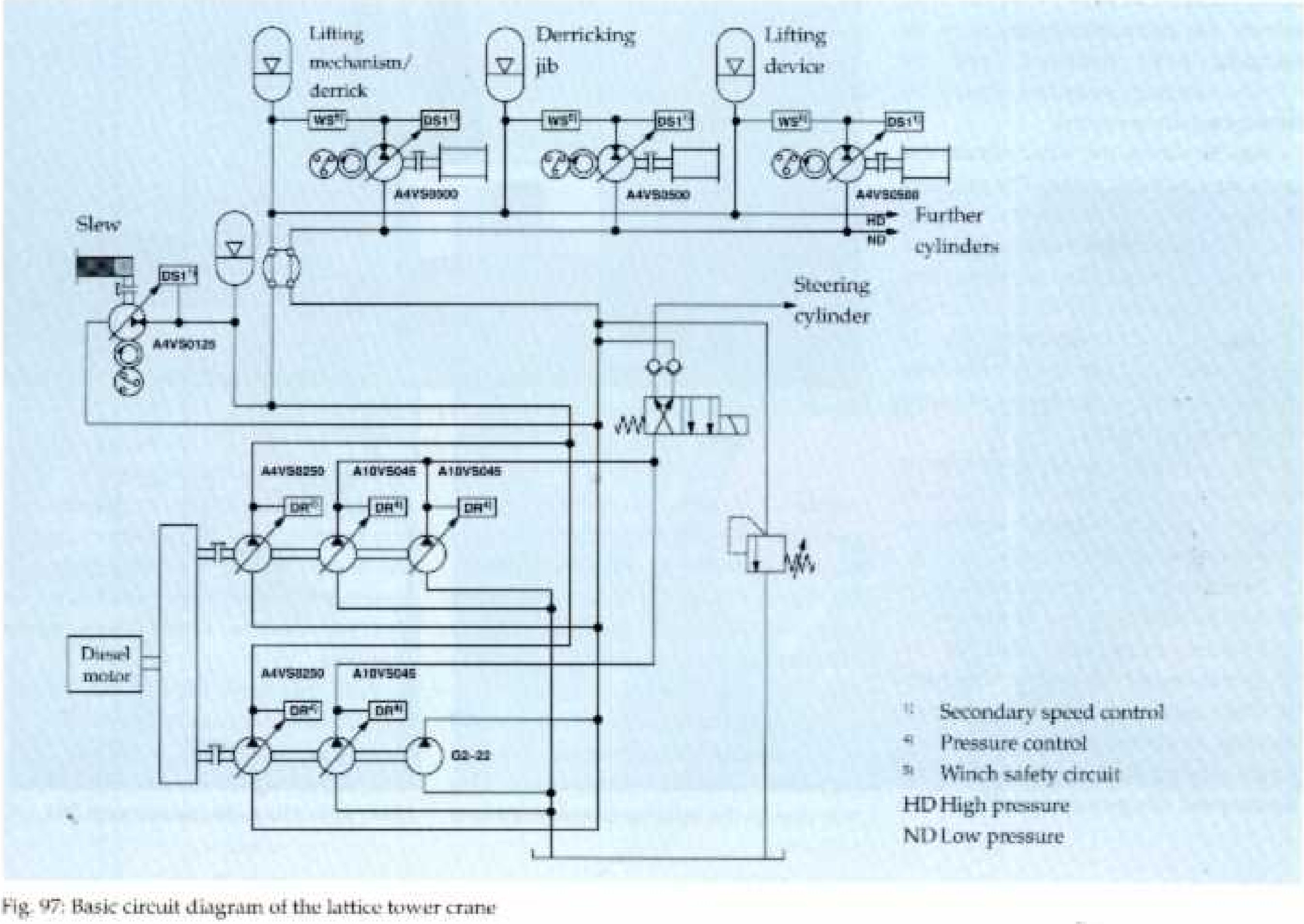


Fig. 97: Basic circuit diagram of the lattice tower crane

A drive for a mobile harbour crane

Fig. 98: Mobile harbour crane type HMK170H

The range of harbour cranes from a reputable manufacturer of mobile machinery was supplied up to now in two versions (**Fig. 98**):

- The electrical version, which consists of diesel motor, generator and electric motors.
  The cylinder for the derricking jib is supplied by an additional hydraulic power unit, which also drives the hydraulic pump.
- The conventional hydraulic variant, which consists of diesel motor, gearbox with mounted control pumps for the hydraulic motors and tilting cylinders.

All rotating drives are operated as closed circuits. Although the hydraulic version is the cheaper option the version with diesel electrical drive was favoured by the manufacturer due to its sensitive control, and this version was more of a market success.

After conversion from the conventional hydraulic version to secondary control the technical problems were solved, the price advantage over the electrical drive remained, and the whole concept of secondary control became generally accepted.

Fig. 99 shows the basic circuit diagram for a harbour crane. The price advantages are clearly on the primary side. With secondary control the gearbox with the four or five mounted control pumps is replaced by an axial piston pump directly connected to the diesel motor and with mechanical pressure control and mounted boost pump (**Fig. 101**).

The pipeline system, consisting merely of a single pressure line but with somewhat larger nominal bore, provides a further saving in costs.

The diesel motor power amounts to 450 kW at 1800 rpm and is transferred to the axial piston pump with 500 cm^3 displacement. In contrast to the diesel electrical drive the diesel speed is matched electronically to the relevant power requirement. Regardless of the diesel speed the generator will run at a constant 1500 rpm. In addition to the generator the lifting and slew devices are also secondary controlled (**Fig. 102**). The lifting device is driven by a tandem unit with 2 x 500 cm^3 displacement, a 125 cm^3 unit with planetary gear being installed on the slew mechanism.

A conventional drive with A6VM units was selected, controlled by a proportional valve, for reasons of cost. This was due to the relative insignificance of the moving gear if the crane needs to be occasionally moved to another location. Oil is supplied by the same hydraulic system with impressed pressure, which also supplies the tilting and support cylinders.

With this drive concept the following technical data can be achieved:
Derricking jib: 42 m,
Min. unloading: 10 metres with 43 t,
Max. unloading: 38 metres with 20 t,

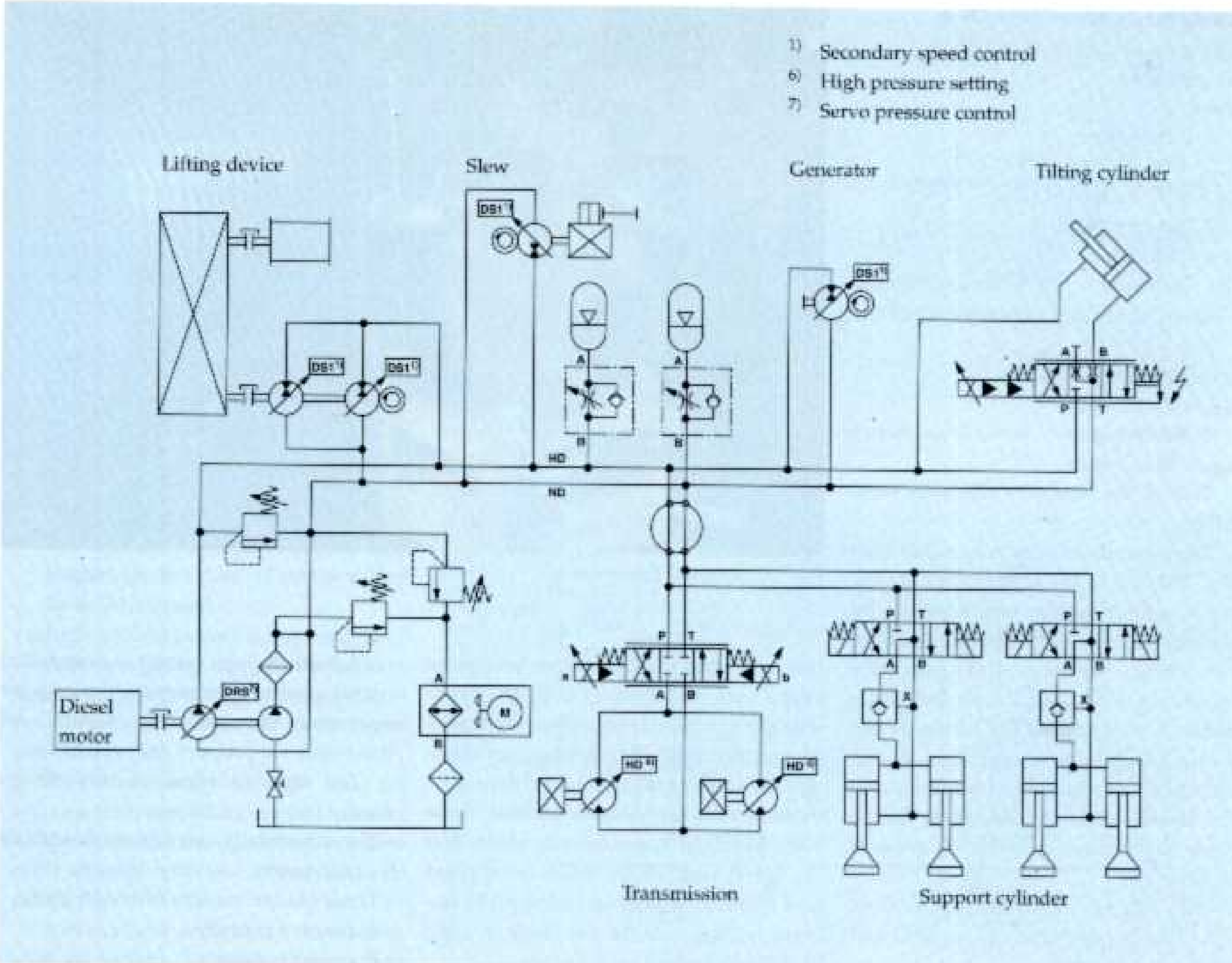


Fig. 99: Basic circuit diagram of mobile harbour crane

Fig. 100: Control electronics

Fig. 101: Power unit

Lifting height above floor: 28 m,
Lifting height below floor: -15 m,
Weight: 220 t.
Speed:
1. Lifting gear:
v_{max} = 30 m/min at 43 t,
v_{max} = 90 m/min at 10 t,
2. Slewing gear:
n_{max} = 1.4 rpm.
This corresponds to a circumferential speed of 180 m / min at the tip of the jib.
3. Tipping gear:
v_{max} = 45 m / min.
4. Moving gear:
v_{max} = 80 m / min.

Two of the four slewable axes were driven.

The complete control electronics (**Fig. 100**) up to the interface programmable control system was supplied by Mannesmann Rexroth. Speed control of the lifting, slewing and generator equipment is digital, the swivel angle feedback is analogue. The lifting device is also fitted with an electrical power control which prevents a pressure drop. The swivel angle can be set so that it limits the torque in order to prevent overloading.

All digital control component groups have programmable control and monitoring functions.


Fig. 102: Slew and lifting drive

A drive for a coke oven feed machine

Exact positioning of coke oven fed machines is a prerequisite for automation as the operating crew no longer travel with the machine, but sit in a central station and merely perform a monitoring function. The operator therefore requires that the maximum permissible deviation from the "oven centre" does not exceed ± 5 mm. Having stated this, the effects 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 or decelerating the service machines, mechanical deformation of the machine frame can occur and cause the chassis to skew with respect 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 pusher rack does not damage the walls of the oven on entering.

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

At the project design stage, three variations for the transmission drive were considered:
- D.C. motors,
- asynchronous motors with a superimposed voltage and
- hydraulic motors.

Hydraulic motors were initially considered only in a conventional drive circuit.

In addition to the transmission there are a number of auxiliary cylinder motions which must be operated when the machine is not travelling.

First the drive problem was solved as shown in simplified form in **Fig. 103.** The transmission drive was split into two parts. The transmission speed was "flow coupled" via the displacement of the two pumps which also catered for the synchronisation. Speed feedback was achieved by means of a tachometer on each side. These were built onto the hydraulic motors or, to prevent slip, onto a non-driven wheel.

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

The disadvantages of this design were:
- Three hydraulic circuits, each with its own pump,
- Three electric motors (through 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 affect the positional accuracy and
- 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. 104.**

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

The advantages of this system compared to **Fig. 103** are:
- Only two pumps with electric motors are required,
- The system can be operated by a single pump (good redundancy),

- The flow requirement 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 valve) and
- Improved oscillation characteristics.

The disadvantages are:

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

After weighing up all the advantages and disadvantages, the results of the investigation leaned towards an electrical drive.

This was exactly the right time for systems with secondary control to make an appearance, although this was a new drive concept with few references to its credit.

After critical analysis of the system with secondary control a decision was made to use the system as shown in **Fig. 105**. The decisive factors were not so much the possibility of energy recovery during the deceleration phase and energy storage, but more the advantages of control and the fulfilment of requirements for high positioning accuracy.

The advantages of secondary control in this instance were:

- One or two pumps / electric motors (better 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,

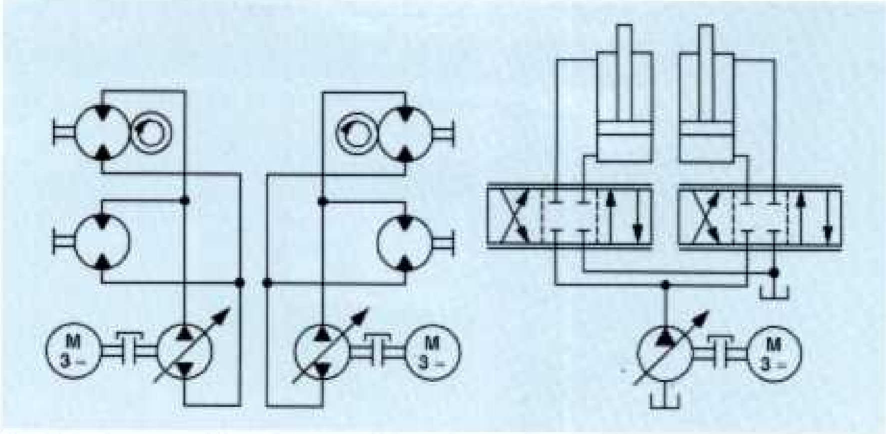


Fig. 103: Conventional drive in closed (left) and open (right) circuit

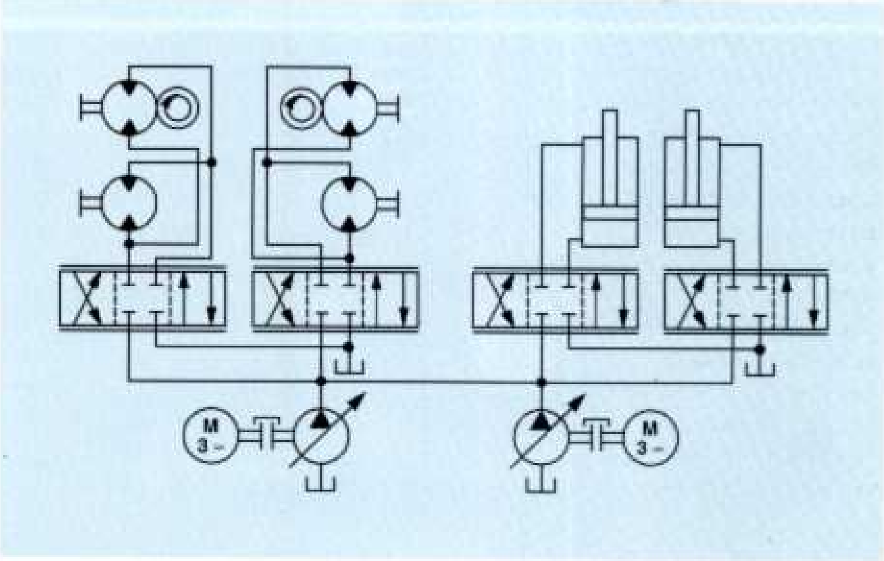


Fig. 104: Conventional drive in open circuit

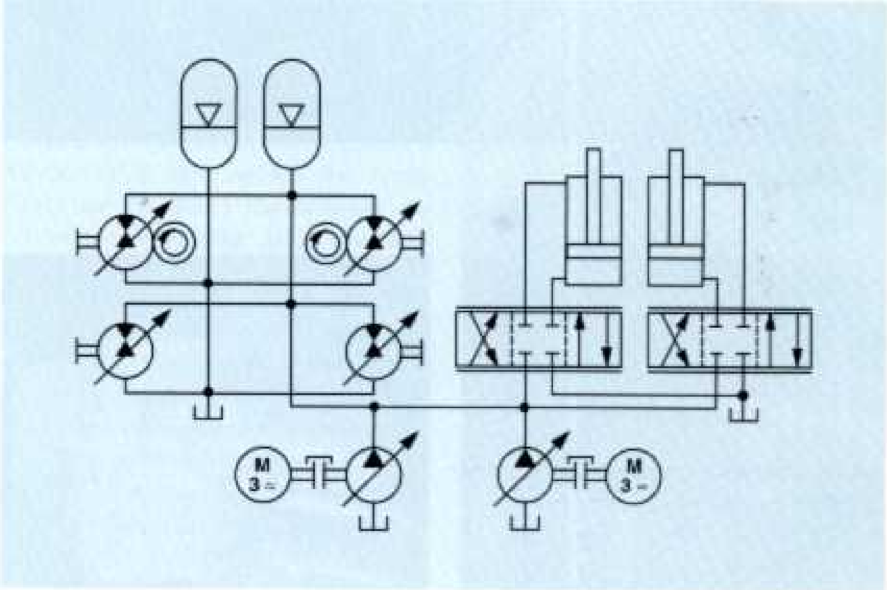


Fig. 105: Secondary control with pressure coupling in open circuit


Fig. 106: Coke ejection machine


Fig. 107: Coke carry-over machine

- The electrical drive power can be reduced,
- The control range is extremely wide and
- The braking energy can be returned to the electrical power line if required.

The transmission drives for the coke ejection machine (**Fig. 106**) and the coke carry over machine (**Fig. 107**) 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 6600 kN for the ejection machine. The highest speed of the machine was 1.5 m/sec. The transmission motors are coupled to the wheels via gearboxes (**Fig. 108**). 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 each other. 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 1 mm is equivalent to an angular rotation of the wheel of 0.09°.

Fig. 109 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 synchronisation control are both performed on an analogue basis. The distance travelled is


Fig. 108: Drive for the coke ejection machine

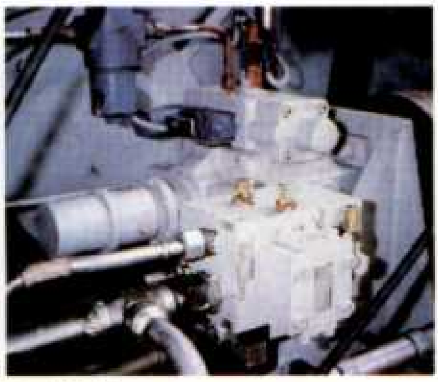


Fig. 109: Drive motor


Fig. 110: Absolute positional tranducer

determined by an incremental impulse generator mounted on a non-driven wheel (**Fig. 110**). The approach and final positioning are 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 motors at above synchronous speed, thus feeding power into the power lines.

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

A drive for a mobile manipulator

In this exercise a mobile manipulator (**Fig. 111**) was required to move a ring weighing up to 100 kN 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 pre-set at ± 2 mm. 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 calculations showed that this could maintain the pre-set tolerances. The overall weight of this vehicle with load is 530 kN. A maximum speed of 2.7 m/sec is required with an acceleration of 0.85 m/sec^2. Although the installed corner power through the actuators taken together amounts to approximately 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. 111: Mobile manipulator in operation

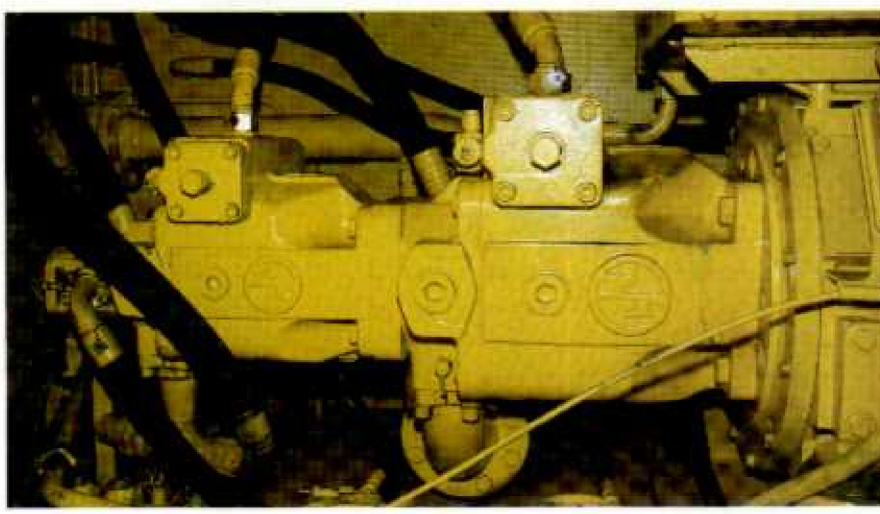


Fig. 112: Diesel motor with axial piston tandem pump

Fig. 114 shows the basic circuit of the mobile manipulator. The diesel engine drives a tandem pump A4VSO250DR + A4VSO125DR onto which the boost pump for the closed transmission circuit is mounted (**Fig. 112**). The slew drive and the cylinders are operated in an open circuit. All motors under secondary control are of the same type - A4VSO125DS. On the slew drive, an incremental encoder is installed in addition to the analogue tacho, for the purpose of positioning. Once more a motor drive was applied so that gear play could be eliminated in the subsidiary swivel angle position controlled loop. **Fig. 113** shows one of the motors for the slew drive.

Fig. 113: Axial piston motor for slew drive

The transmission operation can be described as follows:

During the previous hydrostatic deceleration operation, the accumulator system is 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 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 command 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 swivel over centre in the opposite direction and act as generators, thus recharging the accumulator system up to 260 bar. The pressure controlled pumps swivel back to zero stroke during the deceleration phase. 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 it to 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.

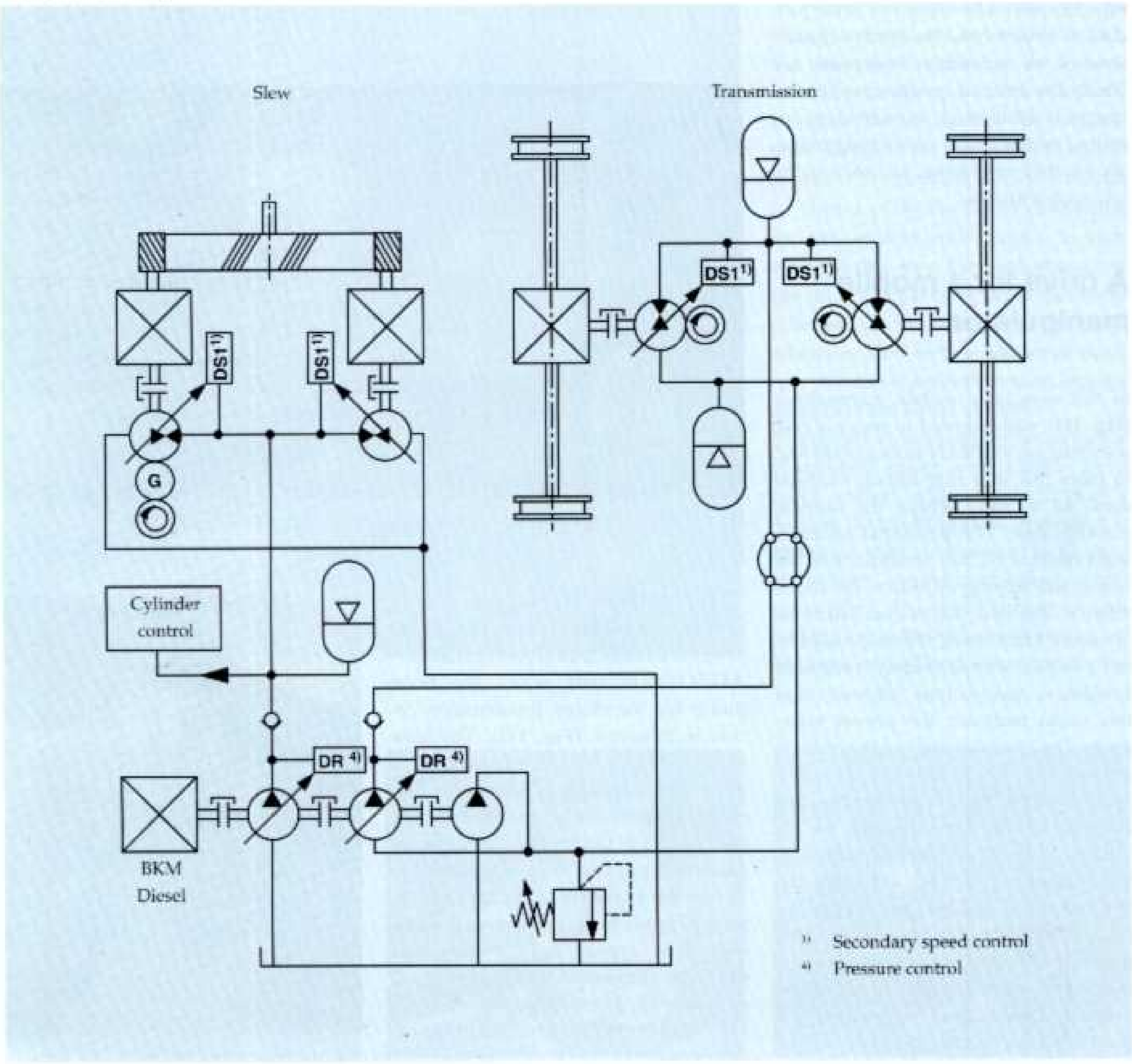


Fig. 114: Simplified hydraulic circuit diagram of the mobile manipulator

A drive for a bucket wheel excavator

One of the first applications in the development of secondary control was in mobile machinery, a system being installed in a bucket excavator, as shown in **Fig. 115**.

The device, with an operational weight of 4500 kN, has a capacity of 1390 m^3 / hr and a drive power of 2 x 300 kW.

Although conventional hydrostatic drives with control of the primary unit have been employed in bucket wheel excavators since 1972, D.C. motors have been preferred for high powered devices.

The disadvantage of this type of drive is the high weight of D.C. 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 over and above the high cost for the D.C. 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 varied of 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 achieved compared with an equivalent electrical drive system. The weight of the bucket wheel head was also reduced by 45 kN.

In total, the operational weight of the bucket wheel excavator could be reduced by approximately 350 kN by using secondary control. The reduced amount of piping due to the common oil supply was also a significant contributory factor.

Figs. 116 and **117** 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 two pumps. In a conventional drive system a total of seven pumps would be required. The auxiliary pumps for the boost and pilot systems are not shown.

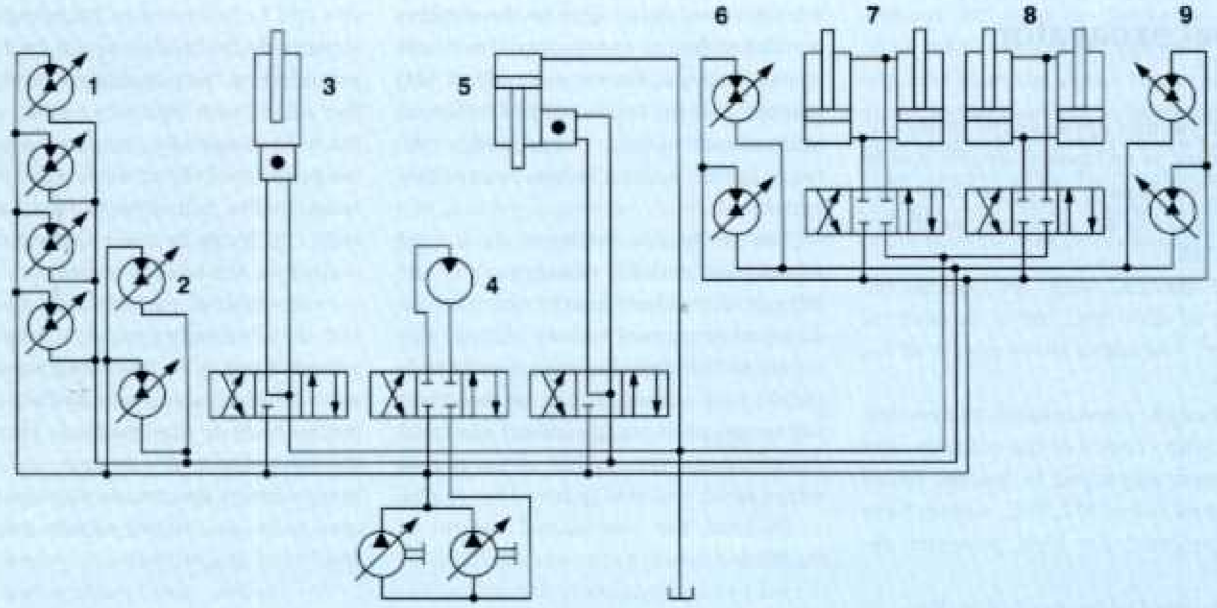
In the case of the system with secondary control, redundancy is higher as, should one pump fail, the machine can still be operated at half power. Alternatively, redundancy can be further extended by the use of more pumps. As the oil column operates under conditions of nominally constant pressure, the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

The material costs at the secondary end in a secondary control system are greater than in a conventional system, as axial piston units controllable over centre have to be installed. However, the advantages of 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.

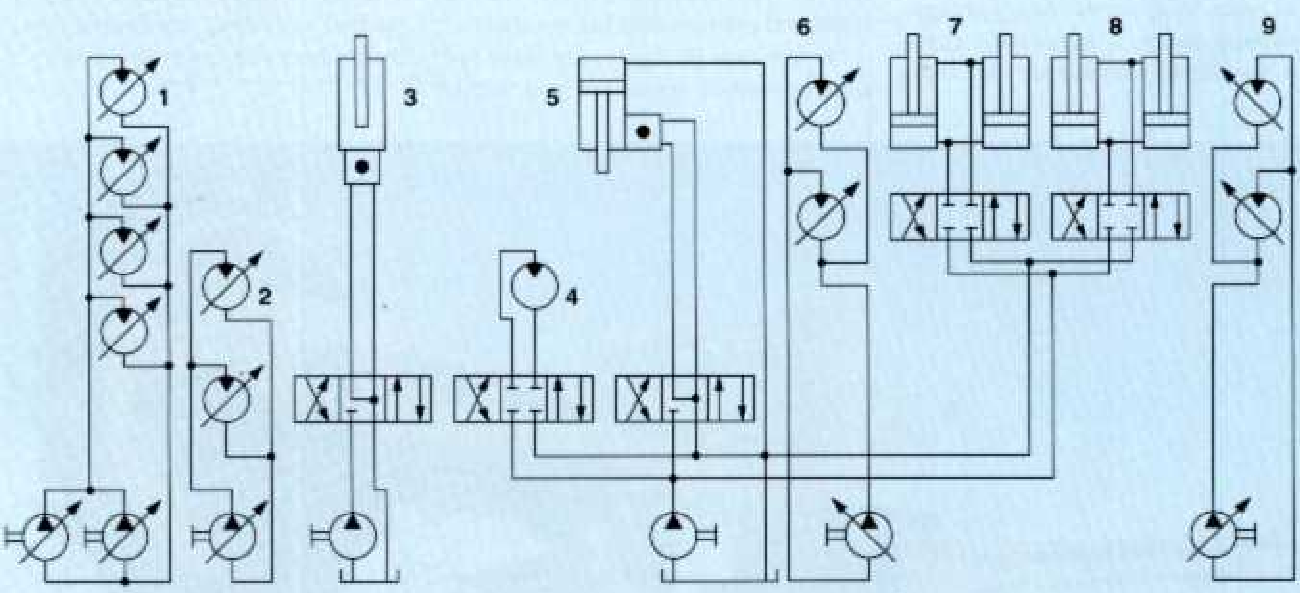
The advantages shown here can only be fully realised if the design requirements can be met and the device re-designed accordingly. This becomes more relevant when a greater number of actuators are attached to the hydraulic power lines. Also, depending on the duration of operation and the degree of parallel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

Fig. 115: 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. 116: Drive for a bucket wheel excavator, simplified illustration of a drive with secondary control



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. 117: Drive for a bucket wheel excavator, simplified illustration of a conventional drive

A drive for a mine locomotive

In spring 1990, after nearly four years of development, the first secondary controlled mine locomotive was produced and put into operation at a Ruhrkohle AG mine after tests above ground (**Fig. 118**).

The diesel motor with a continuous power of 74 kW drives an axial piston pump type A4VSO71DR in swashplate design, fitted with a mechanical pressure controller. Power is distributed over two secondary controlled axial piston units type A4VSO71DS, each driving a single axis. Maximum speed of the locomotive is 7.5 m/sec and a tractive force of 4.5 t can be achieved at the hook.

In accordance with underground mining regulations the open and closed loop controls for the drive are designed explosion proof.

When designing the electronic control circuit the aims were optimum acceleration and deceleration conditions irrespective of load ratio, and also to ensure good wheel grip and prevent wheel slip. For this reason a differential was calculated from the speed actual value of both drive units, beyond which would result in an error signal. This error signal not only reduces the torque of the faster motor, it also increases simultaneously the torque of the slower motor, thus sustaining the tractive force of the locomotive and resulting in optimum acceleration conditions. The speed controller, where the error signal for both drive units is generated, receives the speed command value also the actual value determined for both drives.

In order to prevent excessive acceleration leading to wheel slip, a ramp is required. This will limit the rate of increase depending on the speed differential of the two axes, regardless of the pre-set command value. The gradient of the acceleration or deceleration

Fig. 118: Mine locomotive in operation

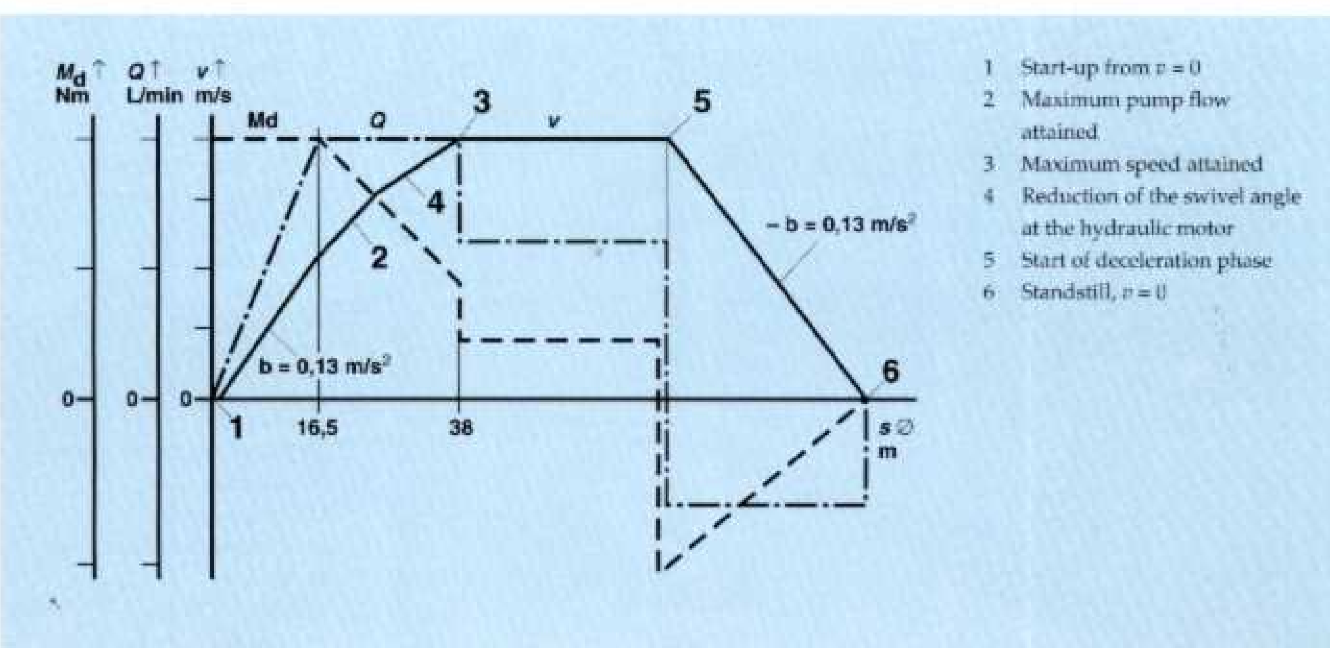


Fig. 119: Travel diagram of mine locomotive

signal is determined by means of a control voltage, which in turn may be altered depending on various operating parameters.

Finally the acceleration values can also be changed depending on the impressed pressure in the pressure circuit of the drive units. In the event of a pressure drop this enables an acceleration value which has just been set to be rejected until sufficient pressure has been restored in the circuit.

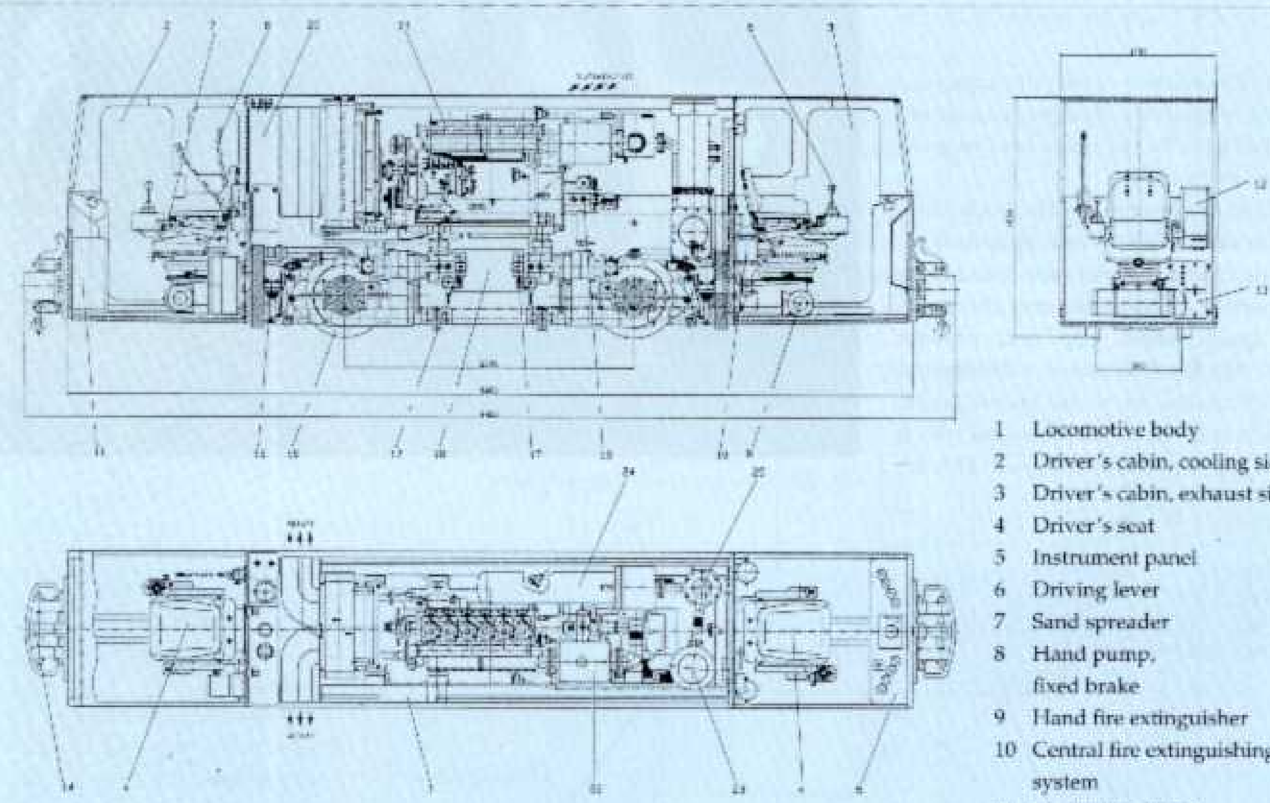
In underground operation "ASR" and "ABS" modes mean additional protection against sparking if the wheels go into a spin. Braking is in general purely hydraulic. In the deceleration phase the secondary units swivel over centre and start operating as pumps (**Fig. 199**). The resulting increase in system pressure causes the hydraulic pump connected to the diesel motor to swivel to a smaller swivel angle, then to cross back over centre and into motor operation.

Motor operation is limited by means of a mechanical fixed stop, as the diesel motor can absorb one third of its nominal power in shunting operation. The remaining braking power is transferred into heat by means of a pressure relief valve.

A mechanical safety brake is operated only in an emergency.

The electronic open and closed loop control system guarantees easy operating of the mine locomotive and permits easy monitoring of all functions. A project planned for the future is the incorporation of this transport system into an automated operation without driver, one which is perfectly feasible and should present no problems.

Fig. 120 shows the arrangement of the individual components in the locomotive.



1 Locomotive body
2 Driver's cabin, cooling side
3 Driver's cabin, exhaust side
4 Driver's seat
5 Instrument panel
6 Driving lever
7 Sand spreader
8 Hand pump, fixed brake
9 Hand fire extinguisher
10 Central fire extinguishing system
11 Electronic control
14 Buffer
15 Fixed brake
16 Wheels
17 Hydrailic motor
18 Fuel tank
19 Hydraulic pump
20 Hydraulic tank
21 Diesel motor
22 Catalytic converter
23 Ejector
24 Exhaust cooler
25 Anti-flame device

Fig. 120: Mine locomotive with ASR (anti-slip control)and ABS (anti-blocking system)

A transmission for an automated transport system

The first mass produced mobile machine with secondary control is the driverless container vehicle CT40, built for ECT (European Combined Terminals) in Rotterdam (**Fig. 121**).

This conveyor system transports containers weighing up to 40 t at the ECT Sealand Delta Terminal (**Fig. 122**).

The main task of the container vehicles is the transport of ISO containers within a terminal and between the loading stations and the maritime cranes to the warehouses for intermediate storage. They are designed so that they can dock at the loading stations and harbour cranes.

These vehicles are only one part of the comprehensive logistics system for controlling container movements within the port. A computer system controls and monitors the driving operations, transmitting the commands to the individual vehicles.

The vehicles themselves have an onboard navigation system that monitors the present position and accepts superordinate commands. They also have an electronic control mechanism. This mechanism controls all drive functions, the diesel motor parameters, the drive and steering systems, it monitors the warning and other signal lights and also the hydraulic and electronics system.

One of the main features of the vehicle is a system based on radar and ultrasonics for detecting any obstacles in the driving path, thus ensuring safe driverless operation.

The drive concept was dependent on a single drive unit for all rotatory and linear movements.

Energy transfer to all actuators is thus restricted to a single pressure line.

Due to the high dynamic response of secondary control the drive mechanism is able to position the vehicle with an accuracy of ± 20 mm irrespective of load. This is all the more remarkable as the vehicle is unsprung and the running radius of the tyres varies according to load.

After a six-month trial period using a prototype a further eight vehicles underwent endurance tests. This meant that at the end of the load tests 40 vehicles were able to receive the go ahead for production.

In 1993 the first automated container terminal in the world was put into operation in Rotterdam using these 48 transport systems.

Closed loop control of the tranmission was by means of system-optimised

Fig. 121: Container vehicle CT40

Fig. 122: ECT-Sealand Delta Terminal

control electronics, in Eurocard format with the VT12000 as the basic component, with control and monitoring electronics for all speed control functions such as ramp, speed control with subordinate swivel angle control and monitoring functions (**Fig. 123**).

The monitoring signals are sent to the PLC to ensure cut-off in the event of an emergency.

The drive electronics are connected to the super-ordinate computer systems via an interface for switch and analogue signals.

With secondary controlled hydrostatic drives the secondary side is generally designed for larger flow volumes than are available from the primary side.

Although this guarantees high torque availability when starting up and high speeds with low torques, power taken from the actuators must be limited in order to prevent pressure drops.

This means that if the speed command value cannot be attained due to an excessive output torque, the secondary unit will operate only below the power limit pre-set by the primary side.

The analogue calculation circuit on the power limiting card determines the maximum possible swivel angle from the adjustable power limit and the current speed of the secondary unit, and influences the swivel angle command value of the standard control electronics VT 12000. In the event of reduced power being available to the secondary unit at low diesel speed or due to power consumption of the hydraulic steering, pressure reduction will be initiated via a pressure switch and will also influence the power limitation by means of a switched input.

A speed command value will also be pre-set by means of the power limiting card to the diesel motor depending on power requirement, and the speed will be matched to the power required. This will keep the drive continuously within the optimum speed range for fuel consumption and service life.

The power used is a D.C. / D.C. converter which converts the 24 volt on-board supply into ± 24 volts for the electronics.

(See also page 59).

These positive results with the first series led to an extension of the terminal, orders being given for a further 60 vehicles.

The control and monitoring electronics of these vehicles carry out a conversion from analogue signal processing into digital, thus relieving the PLC of monitoring and control functions. This conversion to digital also greatly facilitates commissioning and maintenance.

The example demonstrates that secondary control can also be implemented into mobile machinery applications with increasing automation.

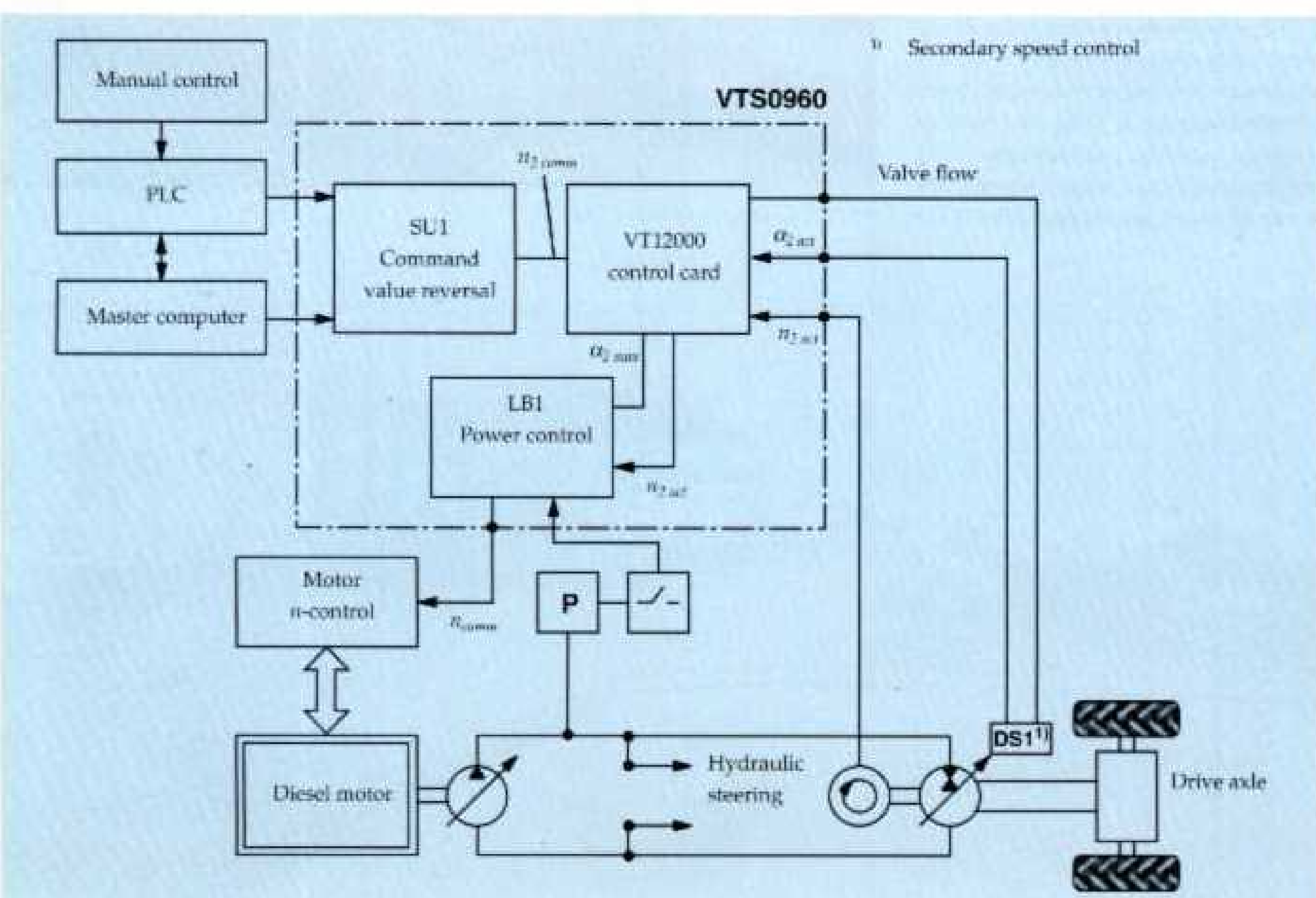


Fig. 123: Block diagram of drive

Driverless transport system

In 1994 around 30 million containers were handled in the ports of South East Asia, more than anywhere else in the world. Loading, unloading, transport and intermediate storage will continue to increase, meaning that many ports will have to increase their capacity by expanding their container terminals. To this purpose they are also continually looking into technological advances.

For one of these ports a remote control transport system was completed at the end of 1994 for handling 20, 40 and 45 foot ISO containers (**Fig. 125**). Secondary control was the choice here due to the high demands placed on the drive system.

Numerous tests regarding suitability and reliability of the complete system had preceded this. These tests involved subjecting three prototypes to a range of tests on speed, positioning accuracy, maintenance and repair, service life and influence on the handling cycle.

In order to extend the intervals between servicing as far as possible where the system is in operation 24 hours a day, there is an integral diagnostic system that automatically displays when the next inspection is due. By means of a laptop computer the vehicle parameters are called and listed, thus enabling any problem to be dealt with individually.

The diesel hydraulic drive system, comprising diesel motor, axial piston pump with built-on boost pump, cooler, fluid tank, filters and other hydraulic devices, is an independent module accomodated in the centre of the vehicle. All line connection points are fitted with rapid action couplings, enabling the drive system to be removed completely from the vehicle for servicing.

Diesel motor power is distributed between two driven axes. The block diagram shown in **Fig. 124** is of similar design to **Fig. 125** when considering the second drive motor. The speed command value is entered into a speed controller (master). The swivel angle of this secondary unit is assigned to the second hydrostatic motor (slave), so that equal torques are set at both motors. The tachometer of the second hydrostatic motor controls the speed of the second drive unit and compares it with that of the master.

If overspeeding occurs with the second hydrostatic motor due to slipping, the speed will be regulated automatically so that both motors run synchronously once more.

The second unit will, however, only run for a short time under speed control.

(cf. page 62).

Even with this remote control transport system the operator has already suggested that the control and monitoring electronics when running in series should be changed to digital signal processing, in spite of the fact that analogue technology has fulfilled the requirements up until now.

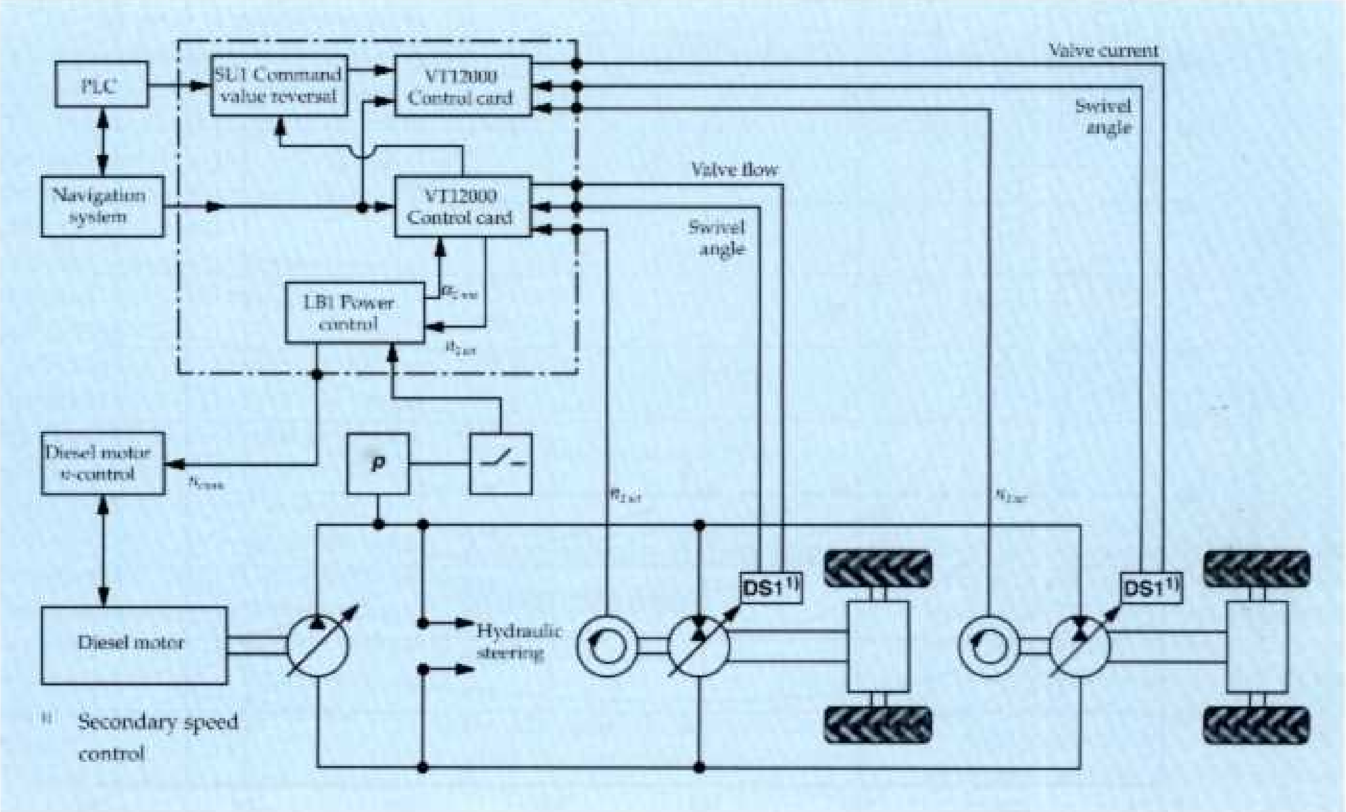


Fig. 124: Block diagram of drive

Technical data of transporter:
Empty weight:approx. 20,000 kg,
Load capacity:approx. 50,000 kg,
Total weight:approx. 70,000 kg,
Length of transporter: 15,500 mm,
Width of transporter: 3,000 mm,
Height of transporter: 1,700 mm,
No. of driven axles: 2,
No. of steerable axles: 2,
Distance between axles: 10,000 mm,
No. of driving wheels: 4,
No. of braking wheels: 4,
Engine:Mercedes Benz OM 442A,
Power at 2100 rpm:260 kW,
Performance data:
Max. speed:25 km/h,
Max. steering lock:± 40°,
Load capacity:
2 x 20 foot container,
1 x 40 foot container,
1 x 45 foot container.

With secondary controlled transmissions, as already mentioned, the diesel motor is given a speed command value which depends on the relevant power requirement. The primary side will thus be kept within a specific speed range that gives optimal fuel consumption.

As the drive speed is de-coupled from the diesel motor speed below maximum power, secondary control can go one step further towards reducing the primary energy requirement, something that other drive systems are unable to do.

Fig. 126 shows the lines of equal efficiency entered for a secondary unit, dependent on the relationship of displacement (swivel angle) to operating pressure. As with the diesel motor secondary control permits free movement in this characteristic field, whilst maintaining speed and torque.

If the vehicle is moved at 300 bar pressure partially loaded, this will occur for example at a swivel angle of 20 % with an efficiency of 70 % (point 1). If this operating state is sustained for a while the logics in the electronics will identify this operating point as one that needs improving, and will initiate a movement in the characteristic field by reducing the pressure.

At 150 bar for example (point 2) the swivel angle will double to 40 % at an efficiency of 86 %. This process has no effect on the drive speed.

Fig. 125: Driverless transport system for containers

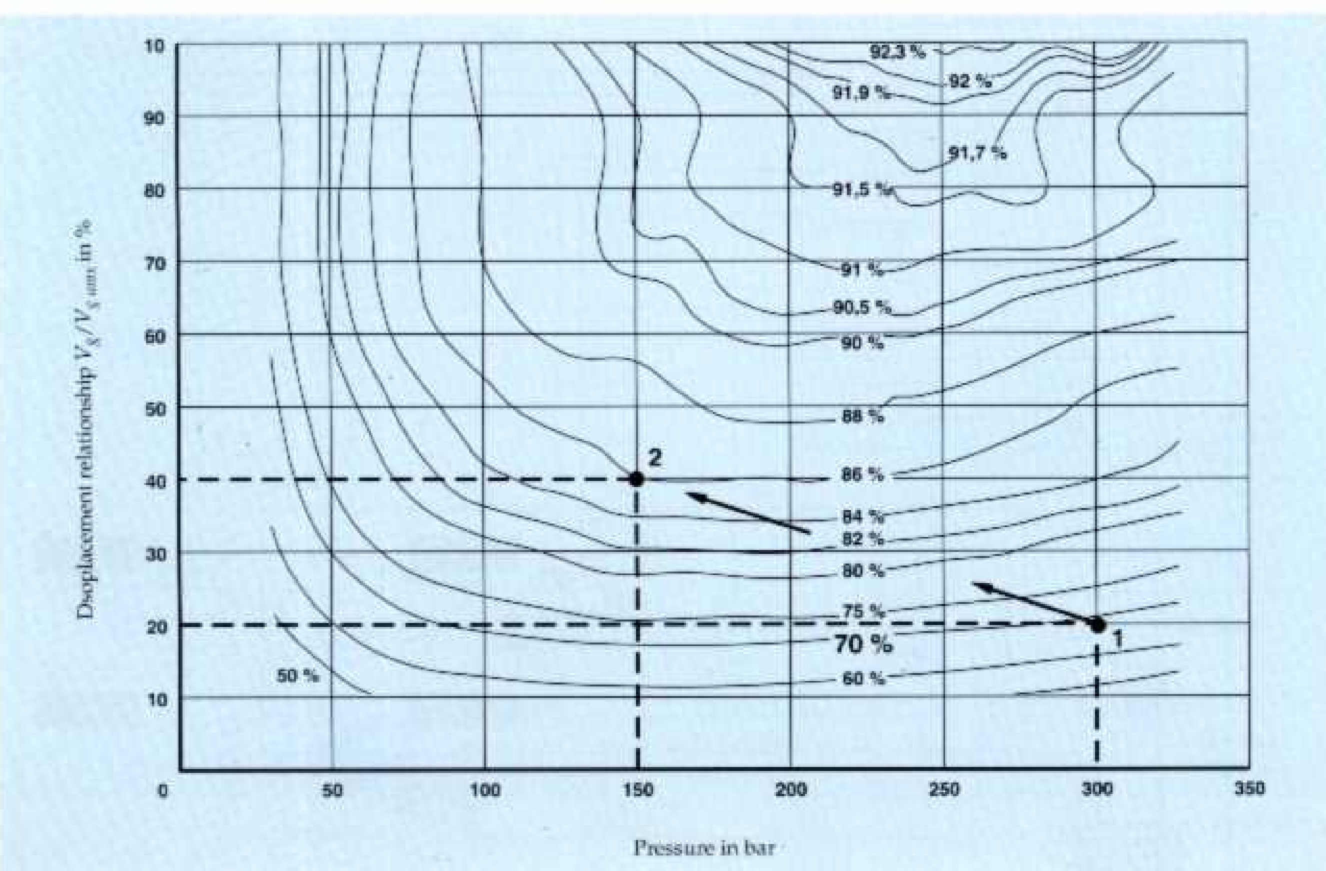


Fig. 126: Characteristic field of a secondary controlled axial piston unit

Drive for a turntable with energy recovery

A turntable with a moment of inertia of 13,500 kgm² is to be positioned in 2 seconds at 90° or 3 seconds at 180° in a foundry moulding machine.

The length of time for one operating cycle for the upper and lower casing must be 26 seconds, with 10 seconds given for filling and pressing.

As the positioning accuracy referred to a radius of 1400 mm may not exceed 0.5 mm, and the drive must be able to recover energy due to its large moment of inertia, a secondary controlled drive was selected.

Fig. 128 shows the basic circuit of the drive for the turntable. The acceleration power of approx. 80 kW is generated by the electric motor (7) with a power of 18.5 kW and a hydraulic accumulator (5). The kinetic energy produced during the deceleration phase is hydraulically stored for the subsequent acceleration.

The high level of positioning accuracy required can only be attained with the use of an incremental transducer (11). This device is coupled directly to the main load (1) so that the backlash of the required planetary gear (2) has no effect on the accuracy.

Speed is regulated by means of a tachometer built onto the hydraulic motor (3) (**Fig. 127**).

Positioning is carried out by the standard DAX 5 digital two-axis control (9). The DAX 5 cam control gives accurate position dependent outputs for further conditions, as well as the 90° and 180° positions to the freely-programmable controller (10). It is possible to change all machine parameters such as acceleration, speed, time delay and position via the DAX 5 display with the aid of a key-operated switch and password.

Once again the choice was the tried and tested VT12000 control and monitoring electronics (8) with PID speed control and subordinate PD swivel angel control. This device monitors not only speed, but also the swivel angle and angular acceleration.

This data is interlinked, prepared and transmitted instantaneously as switch signals. A voltage stabiliser and amplifier output stage for the servo valve are also integrated into the equipment.

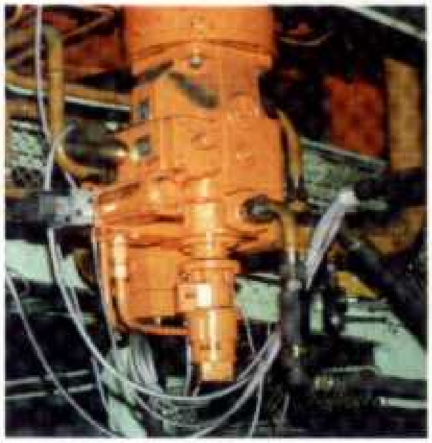


Fig. 127: Secondary unit with drive

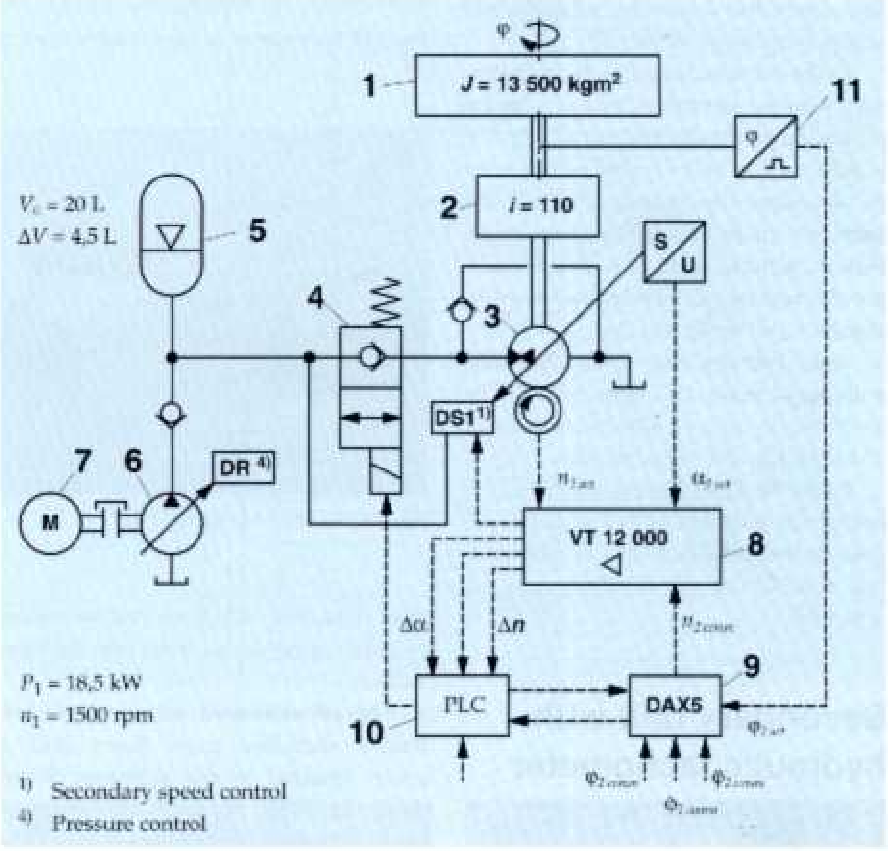


Fig. 128: Basic circuit of a turntable drive

A turntable drive for glass presses

Fig. 129 shows the basic circuit for an additional turntable drive. This turntable is used for moving and positioning glass presses for monitor tubes.

This drive also converts the kinetic energy of the turntable that is produced during the deceleration phase into hydraulic energy, storing it in a hydraulic accumulator for use during the acceleration phase. This way the primary unit need only replace any system losses and can thus be designed based on the nominal power of the secondary unit.

Acceleration power is generated by a pressure-controlled axial piston pump (1) with a flow of 40 L/min at 270 bar and two high pressure accumulators of 10 and 20 litres respectively (3).

Flows of up to 500 L/min can occur at the secondary unit during the acceleration phase.

A 10 litre hydraulic accumulator (4) is installed on the low pressure side. The turntable is fitted with a rim gear. The drive goes from the secondary unit (2) via a backlash-free transmission (5) with pinion onto the rim gear.

In order to maintain stability in the closed loop circuit, an incremental transducer (6) is installed without backlash at the rim gear of the turntable to determine the position.

In addition to the hardware and software safety functions of the incremental transducer the secondary unit is fitted with a centrifugal switch (7).

In the event of a malfunction, energy supply to the secondary unit will be cut off by the hydraulic isolator (8). At the same time the unit (2) will switch over to generator operation, causing the turntable to brake. If the positioning system control is lost, a deceleration process will be initiated by means of directional valves (9) and (10).

Axial piston pump (1) is equipped with a gear pump (11); operation will be in closed circuit with a boost pressure of 15 bar on the low pressure side.

With the DSR digital electronics a positional accuracy of 0.3 angular degrees at the secondary unit, corresponding to 0.004° at the turntable, can be achieved.

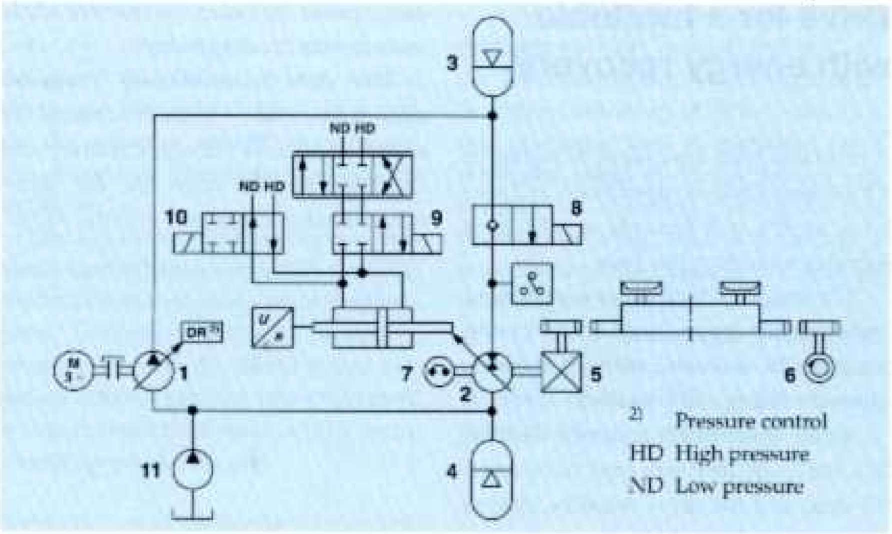


Fig. 129: Basic circuit of the turntable drive for glass blowing

Fig. 130: Chemical-treating ship

Secondary unit with hydraulic tachometer

The latest oil tanker disasters have demonstrated yet again the technical problems involved in skimming oil spills off the surface of the water.

Tankers carrying chemicals pose an increasing danger. The mud flats off the German coast, as one of the most precious and sensitive ecosystems, must be protected from oil and chemicals with all available technical resources.

One of the most efficient systems to combat this problem is by means of the chemical-treating ships developed in Bremen (**Fig. 130**).

The shallow design of these ships renders them suitable for use on the mud flats. The oil is removed by means of a skimming device built into the bow section.

The oil skimmed off via two controlled shell-like traps flows into a sump situated in the skimmer. From there the oil is pumped into the first loading tank by means of two hydraulically driven pumps. This tank contains an inlet section which leads the oil-water mixture without turbulance to the floor of the tank. The following loading tanks, which are situated upstream from this tank, are filled by overflow cascading. They are emptied by two hydraulically driven eccentric spiral pumps situated in the pump chamber. The chemical tank is filled and emptied by a hydraulic hose pump from on deck. A rake device is fitted before the skimmer for separating coarse or fibrous matter by transporting it by elevator from the grille onto a conveyor belt above the skimmer. The conveyor then disposes of it into a solids container.

It is seldom that complete oil slicks have to be dealt with, there usually being isolated drifting contaminated areas. Power consumption of the skimmer pump drives and thus of the speed is undermined by continual fluctuations which have to be controlled manually with conventional drives.

There is a similar problem with the design of the drives for the transfer pump unit with which the loading tanks are bilged. Here, however, the viscosity fluctuations are not so great, as the oil is separated in the tanks when static. In accordance with the specification for newly-built ships, a constant flow should be used when loading and unloading. In other words speed fluctuations of the pump drives brought about by fluctuations in viscosity of the medium used must be automatically compensated for. The viscosity of the medium can range from the low viscous oil-water mixture to 40,000 cSt for heavy oil.

On the basis of this specification a secondary controlled drive was designed for the pump drives, whereas the remaining hydraulic actuators such as winch, crane and hydraulic cylinders are connected to a conventional constant pressure circuit of 200 bar operating pressure. The secondary control system permits the skimmer and transfer pump units to be maintained at a constant pre-selected speed irrespective of any external torques being caused by viscosity fluctuations of the medium. The drives are equipped with a hydraulic tachometer (**Fig. 131** (1)) for setting the speed.

The pilot oil flow for the hydraulic tachometers is adjusted at an upstream flow regulator (3) in a control stand situated on deck. The self-adjusting pressure before the tachometer is used to control the hydraulic dependent adjusting device that regulates the swivel angle of the hydraulic motor. The pilot pressure for the tachometer will lie within the range of 10 - 45 bar for speed control. This pressure range corresponds proportionally to a swivel angle of 0° to maximum for the hydraulic motors. If the set pilot pressure increases upstream of the tachometer during operation, the tachometer and therefore the drive motor will rotate too slowly. The pilot valve of the drive motor is controlled by means of the control pressure removed, causing the hydraulic motor to swivel out to enable a higher torque to be output, thus increasing the speed. The increased oil consumption is compensated by the pressure circuit with impressed pressure. If the pilot pressure is too low the speed of the tachometer and hence that of the hydraulic motor will be too high. The swivel angle of the hydraulic motor will be reduced by actuating the pilot valve, thus reducing both output torque and speed.

The use of hydraulic tachometers as in **Fig. 132** is reserved for special cases. It is, however used when operating in toxic or explosive environments. The level of speed accuracy attainable is dependent solely on the change in volumetric efficiency of the tachometer.

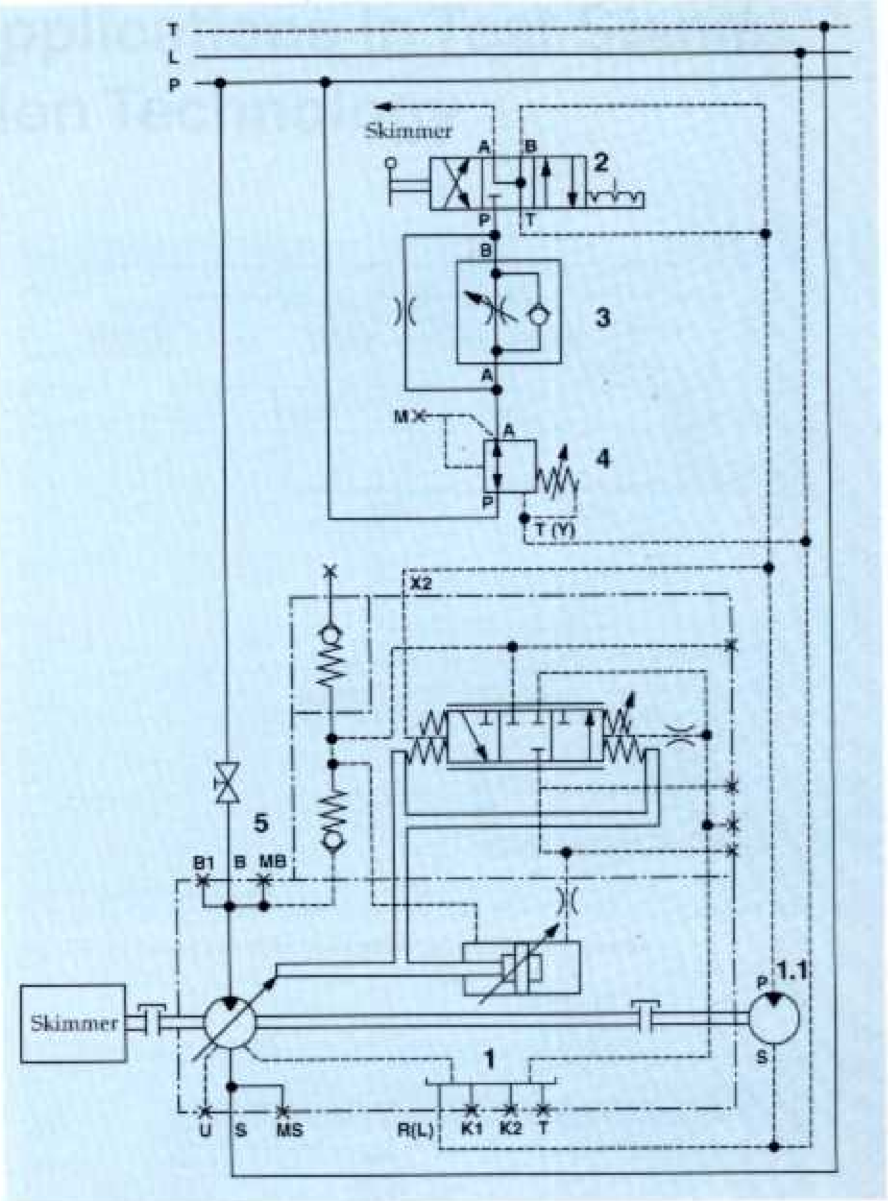


Fig. 131: Secondary unit with hydraulic tachometer

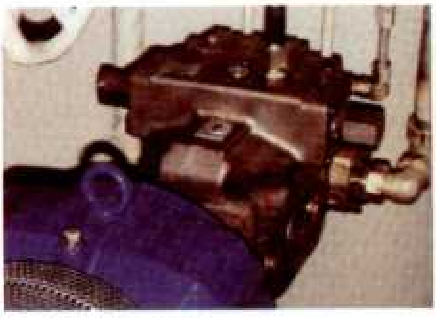


Fig. 132: Drive for skimmer

Fig. 133: Drive for supply pump

Examples of Applications in Test Stands and in Simulation Technology

Introduction

The development centres for the automotive industry and its associated branches have been applying secondary control increasingly since 1982. Up to that time, the limited dynamic response of individual components and lack of digital computers meant that predominantly only stationary tests were possible on electrical and conventional hydraulic test stands.

Today, in addition to secondary control, power electronics, measurement and monitoring electronics, fast and versatile computer systems are also available for high dynamic response tests using test stands. By combining these components we now have a range of possibilities for moving more tests off roads and test tracks and into the laboratory, with a virtually perfect simulation of actual driving operations and conditions.

This enables tests of components on the road to be limited to those which are ready for production.

This simulation of driving conditions on suitable test stands brings with it the advantage of reduced test times, yet at the same time increasing the number of tests.

The requirement for testing under real conditions in the laboratory, as they actually exist in the vehicle on the road, means testing components, assemblies and structures right through to the complete vehicle. Depending on the energy flow, we can differentiate between three types of test stand that are available. These are suitable for the following:

- Pure braking systems (**Fig. 134**).
- Systems with energy feedback in which only the energy losses need to be used as the "power input", as the energy output of the test piece is re-used as a second input (**Fig. 135**).
- Systems with energy recovery into the hydraulic or electrical power lines (**Fig. 136**).

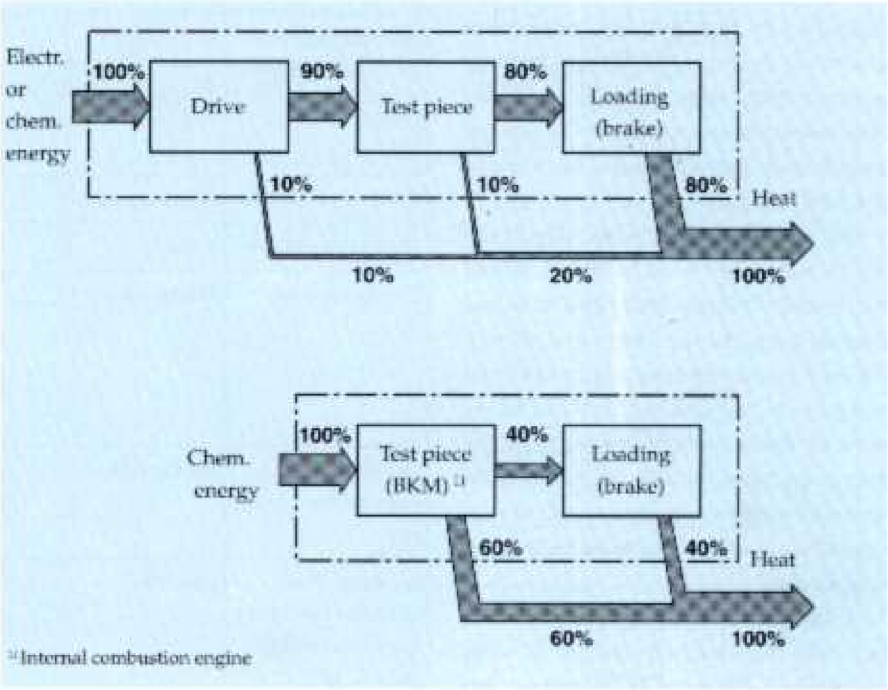


Fig. 134: Test stand principle: Energy flow in pure braking systems

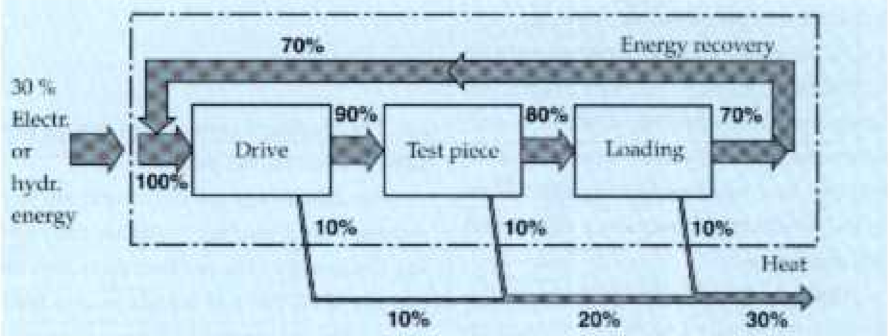


Fig. 135: Test stand principle: Energy flow with energy feedback

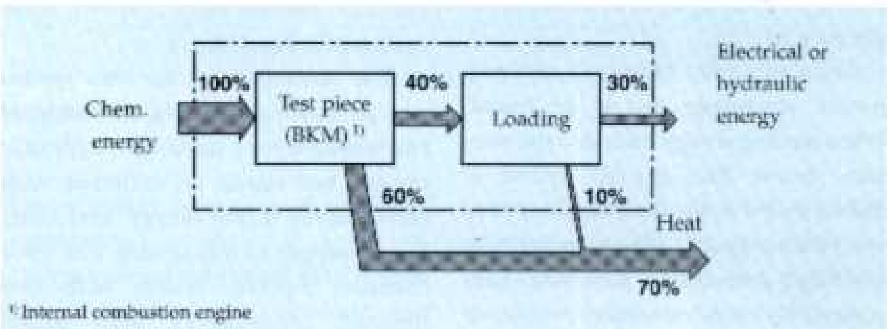


Fig. 136: Test stand principle: Energy flow with energy recovery

With pure braking systems such as water turbulence or eddy current brakes, the total primary energy will be converted into heat. As this available heat cannot be utilised for cost reasons, it is given off into the atmosphere, for example via a cooling tower. Although capital costs for the test equipment alone are relatively low, high energy consumption, together with considerable maintenance costs for the energy supply and heat transferral have a significantly negative effect.

Back tensioning systems (**Fig. 135**), which have been used in electrical and hydrostatic units for some time now, are considerably cheaper from the energy aspect, because the power output by the test piece to the loading unit can be re-used by the drive without the need for energy conversion. Only losses from the unit and test piece, together with the acceleration power, have to be externally supplied.

If the back tensioning system cannot be used, the loading energy is converted into electrical or hydraulic energy (**Fig. 136**), and is then supplied to other actuators such as a back tensioning system by means of the relevant distribution supply network.

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

Table 4 shows a comparison of the three drive systems such as are used on test stands. It is easy to see that, due to the dynamic response in four quadrant drive, the axial piston unit under secondary control is superior to all other systems.

From the point of energy recovery further 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 a pure sinusoidal current. Dynamic operating conditions are covered by the hydraulic accumulator. The electrical power line is thus not subject to overload peaks.

	Eddy current power brake	DC machine	Secondary controlled axial piston unit
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 hydraulic central 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 4: Comparison of drive systems in test stands

The advantage of hydraulic energy recovery and storage without converting the energy into another form can be utilised if all the test stands in one testing centre are connected to a common hydraulic system.

Such a system which has been undertaken for a well-known German automobile manufacturer is shown schematically in **Fig. 137**.

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 axial piston pumps designed for 280 bar. This piping system was extended and, in the first extension stage, test stands for engines, drive shafts and gearboxes were connected. All of these units operate under secondary control with impressed 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 be 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, together with the losses of the units under test and the hydrostatic units are therefore

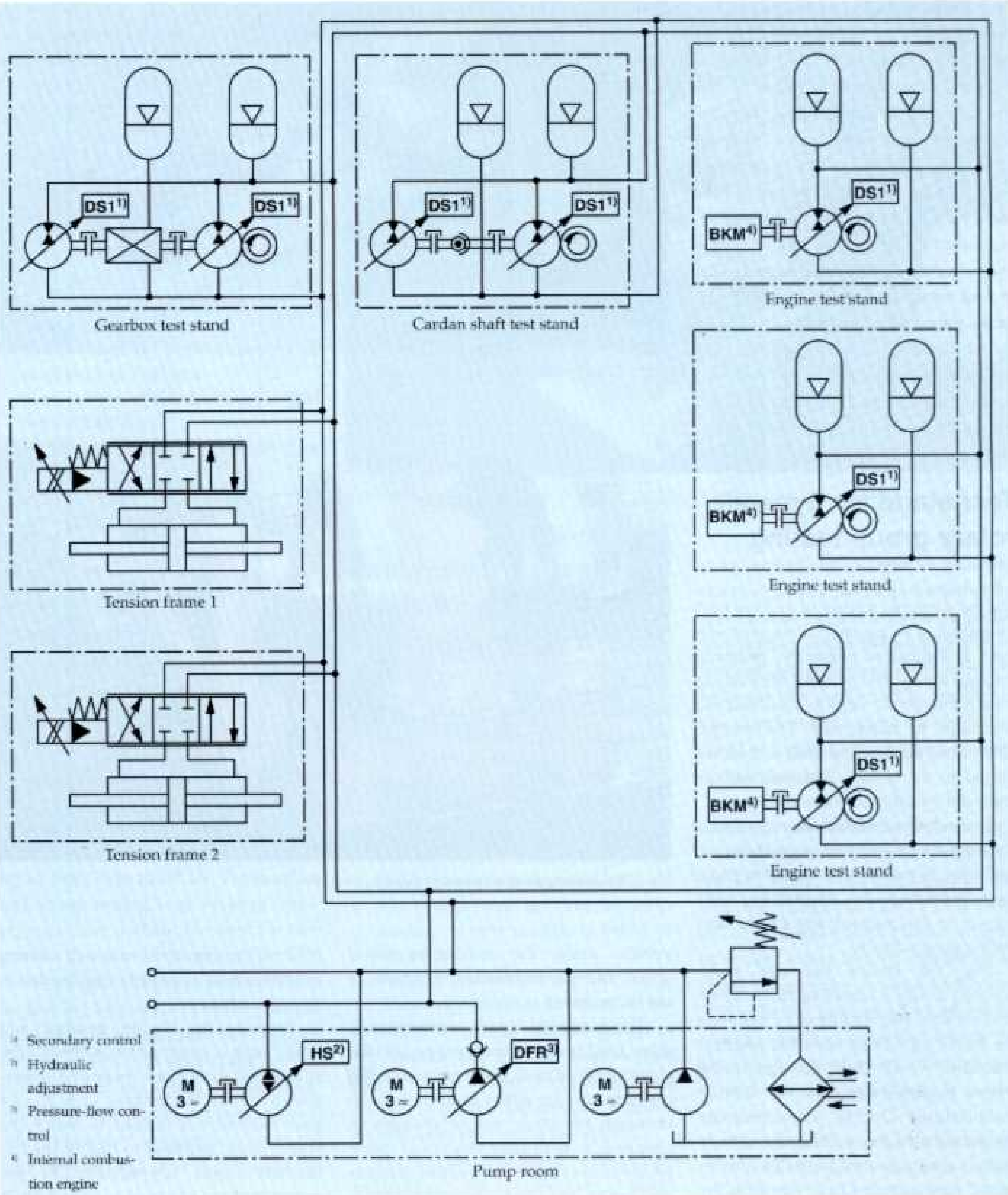


Fig. 137: Ring main system for a test bed with energy recovery

all that need to be fed into these systems.

As all test stands work completely independently of each other the question of energy balance is of particular importance.

Pressure compensated axial piston units are installed in a common pump room. These 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. Any power deficit is also 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 or 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.

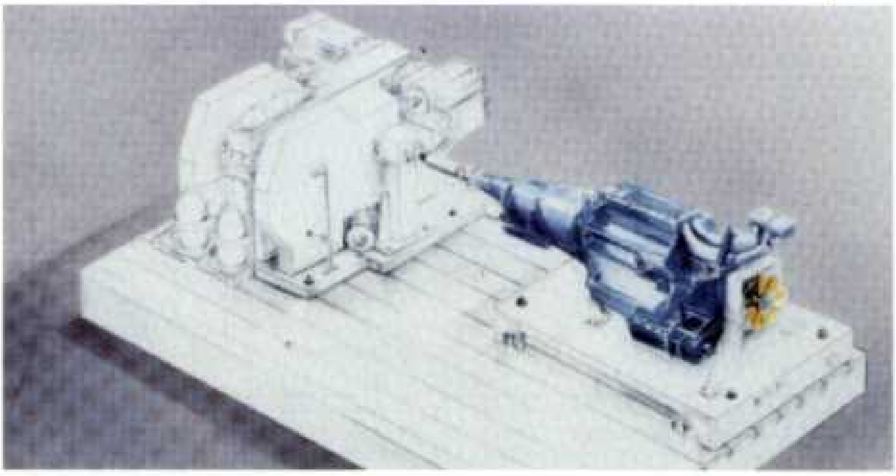


Fig. 138: Test stand for drive components

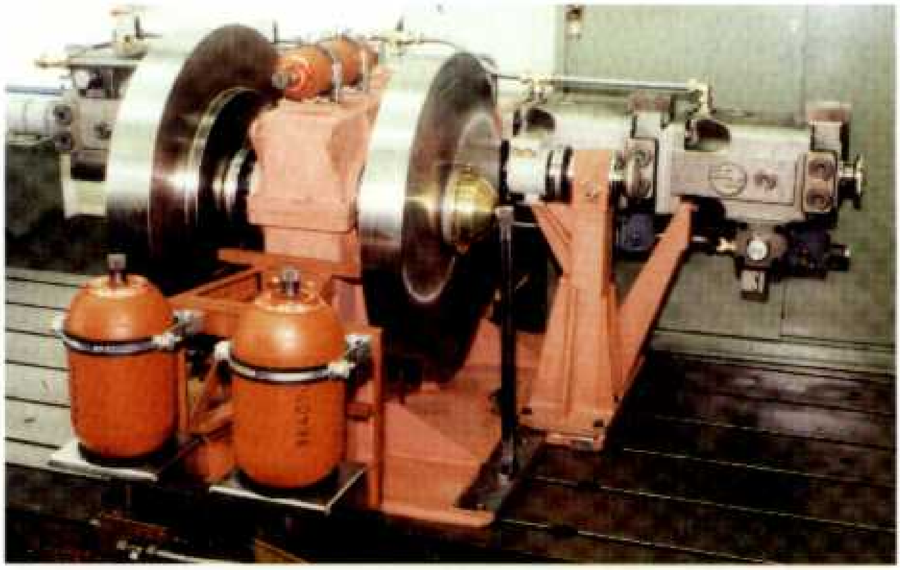


Fig. 139: Loading device for test stand for drive components

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, 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 regarding construction and adaptability with respect to the various test pieces and testing programmes.

Fig. 138 shows the test stand adapted to test a complete drive transmission line. The loading units simulate the travel conditions from the smallest unloaded vehicle right through to the largest loaded vehicle of the specific manufacturer. During acceleration or deceleration of the vehicle the vehicle mass is also simulated to enable any required measurement to be made on the stand without the need for re-building or re-setting.

Fig. 139 shows the arrangement of the rotating masses and the A4VSO...DS1 axial piston units under secondary control.

For the simulation of the mass the torques are inserted on the load side. These are calculated from the pre-set 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 either much too small or much too large, severe changes in torque (e.g. with 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 flywheel elements which can be selected 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 are kept within limits, as only a small part of max. 20-30% of the currently selected rotating mass needs to be added or subtracted in the simulation.

Between the engine, gearbox and brake unit a drive shaft is fitted. The speed and torque are measured immediately before the braking unit. A spur gear reduction is utilised to match the speed range required to that of the hydrostatic units. Depending on the power and speed range required either two or four hydrostatic units are installed. In order to simulate resistance of the vehicle travelling, the gradient, rolling and air resistance values, vehicle frontage area, axle ratio and the dynamic rolling ratio of the tyres are all pre-set. The air resistance is calculated depending on vehicle speed. The gradient value can be varied as required during the test 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 tranmission line with two or four driven wheels.

The design data of the test stand in **Fig. 139** is as follows:

- Speed range (in both directions of rotation) 0 to 7000 rpm,
- Load torque (referred to n_{max} = 7000 rpm) ± 2000 Nm,
- Vehicle weight (range) 700-4500 kg and
- Mass simulation at loading unit with a control constant in the range of ± 0.2 kgm^2.

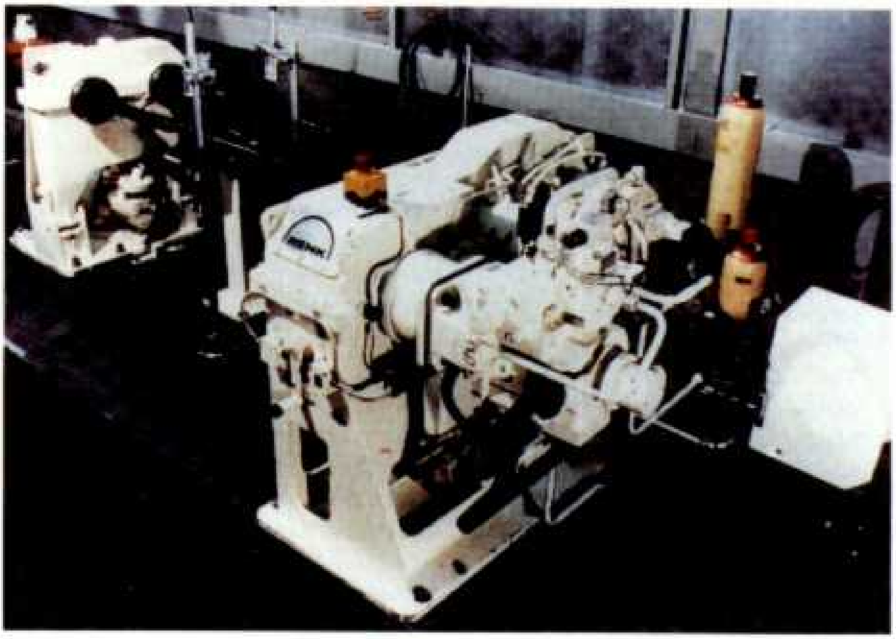


Fig. 140: Test stand for drive shafts and cardan shafts

A drive shaft test stand

The drive shaft test stand shown in **Fig. 140** 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 (see "Test stand for dynamic rotary group testing") can be most sensibly used here without the need for conversion into any other energy form. By means of gear coupling, four units can be coupled together and tested simultaneously. The intermediate bearings are 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.

Fast response test stand for internal combustion engines

The following tendencies are apparent from development in the automobile industry:

- 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 and
- Reliability is being improved.

These requirements have had an unavoidable influence on the design of test stands 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 that technical parameters such as pressure, flow, temperature, force, velocity, acceleration and gas relationships must be measured and evaluated, it also stipulated that the unit must be capable of dynamic endurance tests in which the loading and responses change automatically as would be found in a road test. The features of this stand are the dynamic control of speed and torque and the measurement of these values.

In the automobile industry it is common with test stands for internal combustion engines to use power converter fed DC shunt wound motors. In this way, the energy produced by the internal combustion engine can be accepted by the DC motor and fed into the three-phase power lines via suitable circuitry. This calls for extensive and expensive control circuitry. However, due to the large moments of inertia, the dynamic control of such a test device is insufficient.

The recently proposed introduction of field excited power converters 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 moments of inertia 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 for this decision were:

- that on acceleration and deceleration, an energy exchange with the hydraulic accumulator could take place without the need for dynamic feedback into the electrical power lines and
- the electrical supply could be arranged for average usage rather than for the peak powers required.

The following requirements were placed on the test stand:

- Max. power: $P_{max} = \pm 290$ kW,
- Max. torque: $M_{max} = \pm 550$ Nm,
- Speed range: $n = 600$ to 7000 rpm and
- High dynamic response of speed.

The loading machine must, when uncoupled from the test unit, be able to be driven in a triangular wave form from 1000 rpm to 7000 rpm and back again within 1 sec.

Fig. 142 shows the test cell with the loading unit consisting of a splitter box with two speed-controlled axial piston units in parallel. 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 **Fig. 141** 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. Depending on energy flow the loading machine reacts accordingly and the engine under test either drives or is driven.

The moment of inertia of the 290 kW loading machine is 0.126 kgm^2, referred to the high speed shaft. The average natural acceleration power required to run from 1000 to 7000 rpm in 1 sec will, at $P = 35$ kW be extremely small. The low moment of inertial results in only minimal torque peaks from the test piece. It is therefore possible to use high dynamic response measuring torque shafts in continual operation. The torque determined is very close to the test engine in dynamic response.

A switching safety circuit ensures that the power feed to the loading machine is broken immediately should a fault occur. Within 1.2 sec the speed will be reduced to zero. 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. 143 shows the basic hydraulic circuit. Due to the impressed 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 has no effect on the dynamic response of the system.

The energy required to accelerate the loading machine itself is taken from the hydraulic accumulator which restores the energy during the deceleration phase. This in turn means that the measures which would be required to stabilise 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 stipulated that a speed response curve had to be produced. This curve is shown in **Fig. 144**.

The diagram shows the actual and command values for speed against time, measured on the high speed shaft of the loading device. The engine under test, which could hardly have withstood this process over an extended period of time, was uncoupled during this test.

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

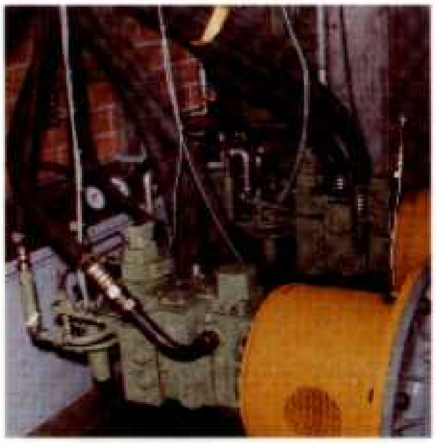


Fig. 141: Three phase motor with pressure controlled axial piston units

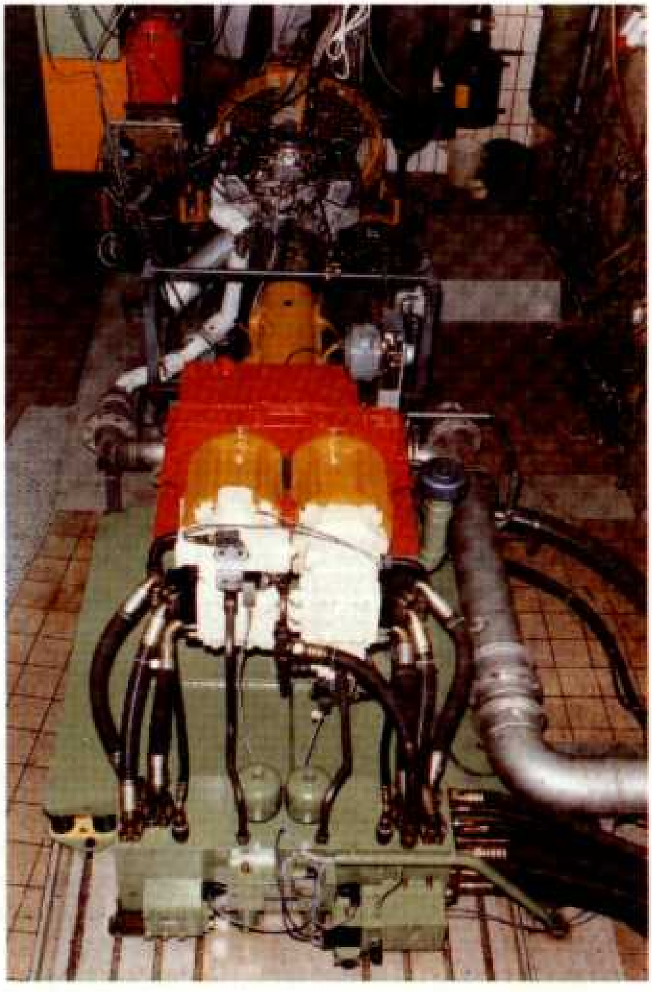
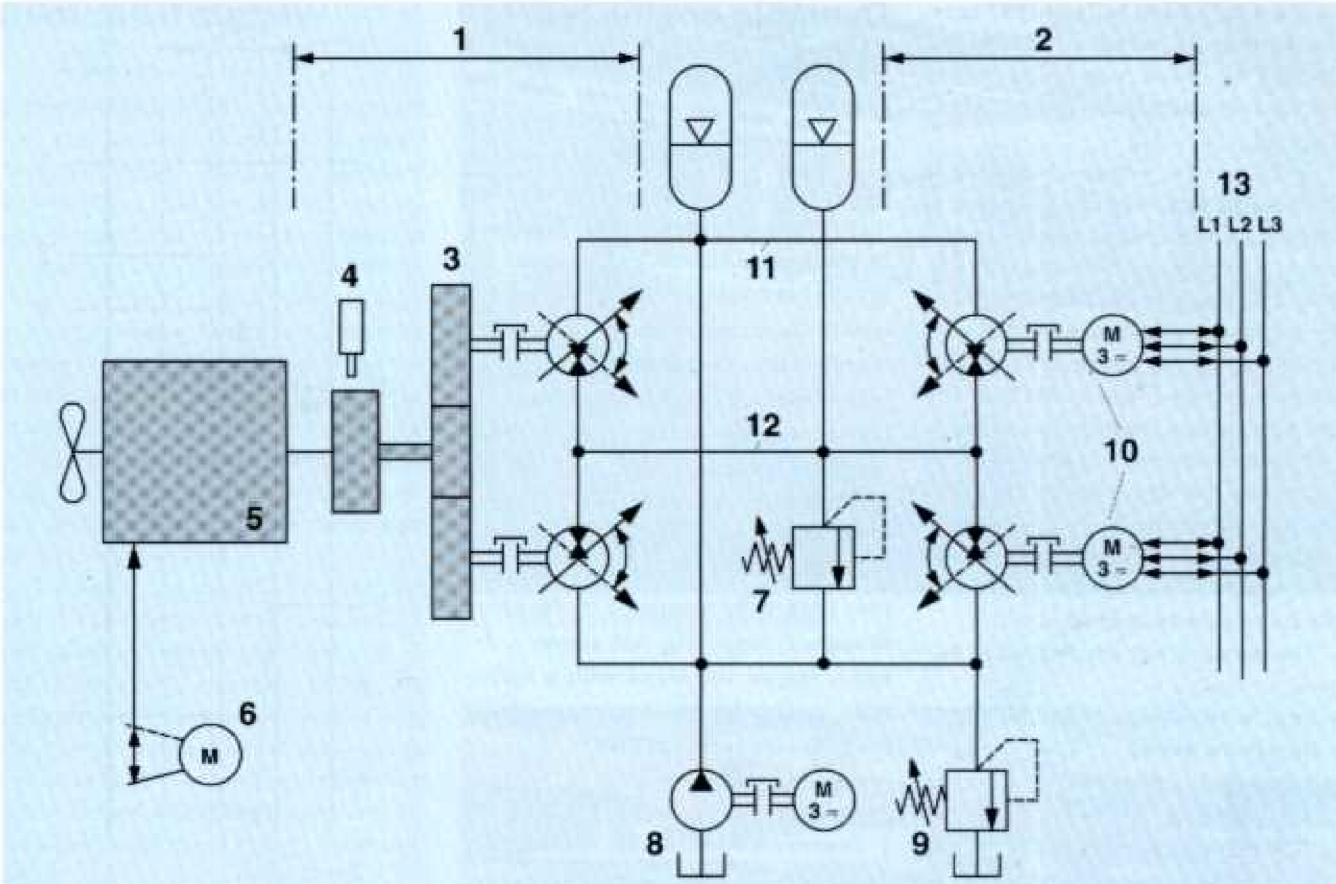
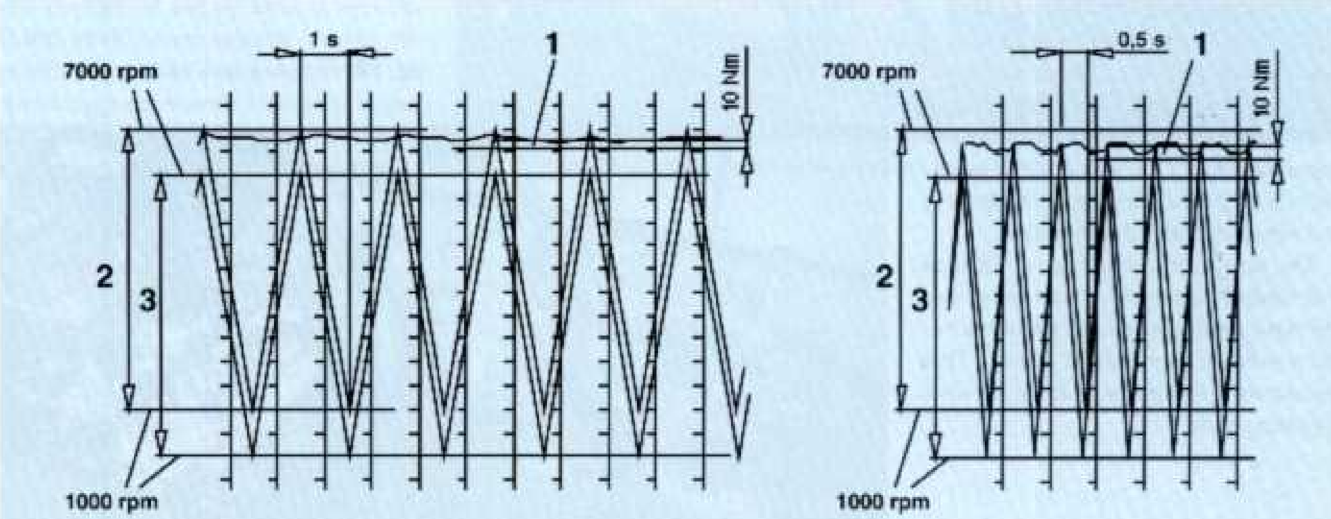


Fig. 142: Overall view of loading machine with test piece



1 Loading machine
2 Power supply unit
3 Intermediate drive
4 Speed sensor
5 Test piece (internal combustion engine)
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. 143: Basic circuit of test stand for internal combustion engine



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

Fig. 144: Measured speed curves

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

With the test engine connected a curve will be produced as shown in **Fig. 145**. The internal combustion engine rotates at 5,700 rpm with the throttle fully open. The torque at this time is 310 Nm. The engine is pulled down to 3,500 rpm by the loading machine in 0.9 sec. This causes the torque to rise to 400 Nm. The loading machine must therefore decelerate both the engine and itself in order to generate the higher torque. It fulfils this duty completely.

The pre-set values for speed control of the machine and torque for the engine control can be entered at will.

The signals given are derived as follows:

- from a simulation computer given the vehicle model,
- from actual values obtained during test drives or
- from a synthetic programme set out by the engineer.

  For these to be effective, the closed loop control must fulfil the following tasks:
- 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 and
- Electronic pressure control of the axial piston units of the primary units via pressure transducers.

The axial piston units which had been running since December 1983 were checked after a running time of 7,500 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 deterioration.

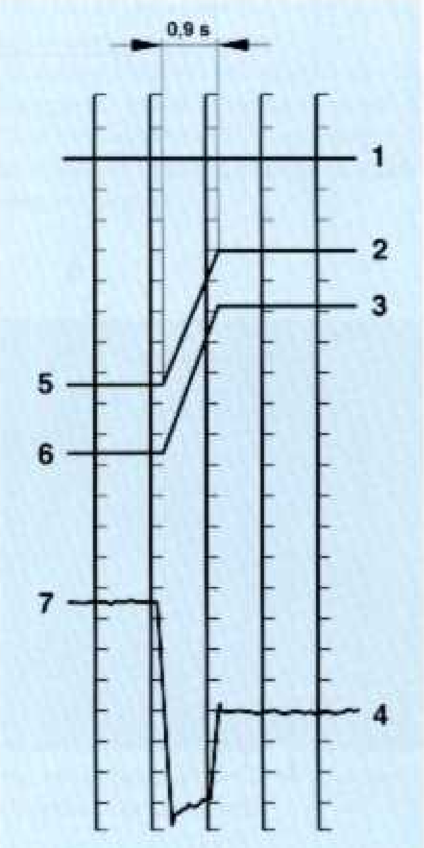
On first commissioning, a decision was made based on the assumption that the main components only require a basic overhaul after 10,000 hours. This corresponds to approximately 5 years' operating time.

Dynamic engine test stand

The demands of automobile manufacturers for a test stand are not limited merely to static characteristic diagrams. In particular internal processes in the vehicle have to be simulated, in order to enable the majority of the tests to be transferred to the laboratory. If the parameterisation values for start-up and gear change are known, the test equipment has to ensure that road driving conditions are simulated true to nature. Even vibrations in the drive transmission line as they occur after a change of gear have to be simulated in the lower frequency range. **Fig. 146** shows a dynamic engine test stand with a hydrostatic unit under secondary control and the following technical data:

- Braking power 180 kW,
- Positive drive power 125 kW,
- Speed 9,000 rpm and
- Torque 380 Nm.

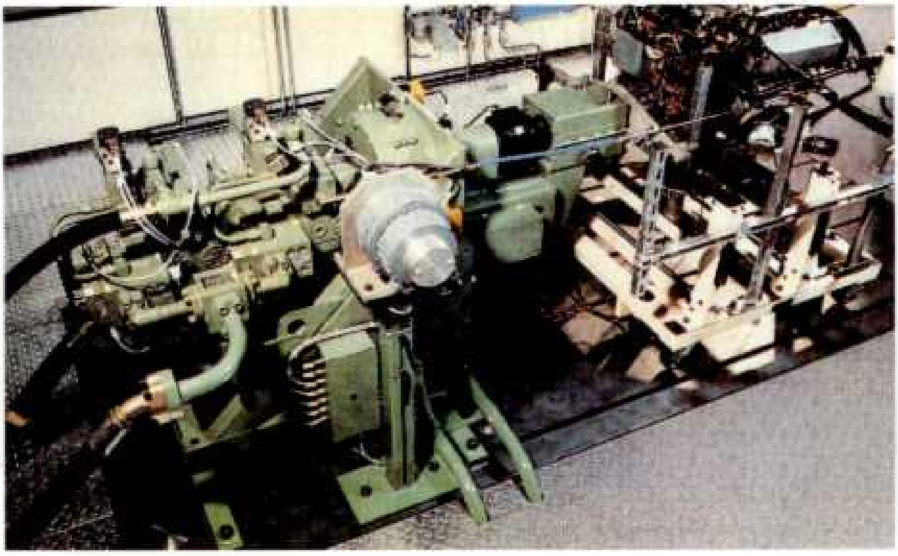
As this is a single test stand i.e. there are no other hydraulic actuators, the braking energy is converted into an AC current via the electric motor and fed back into the electrical power line.



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

Fig. 145: Speed and torque curves with engine connected

Fig. 146: Test piece with loading unit



Central oil supply

Before the days of secondary control it was not possible to connect a rotary drive to a central oil supply, unless a throttle valve was installed in the energy line and also if it was accepted that the energy balance would be unfavourable. With this drive technology a high dynamic response can be achieved with changes of speed and torque, as the hydraulic spring is pre-tensioned up to the throttle valve (servo or proportional valve). However, as the losses are quite high the heat exchangers here are usually designed to run on 80 % of the electrical power installed.

This cost can be reduced considerably if a sufficient number of secondary controlled test stands are connected on the actuator side, as shown in **Fig. 148** with the central oil supply. In this case the heat exchanger can be designed to run on only 30 % of the electrical power, which means that there is considerable reduction in primary energy consumption.

Twelve axial piston units with 250 cm^3 displacement are used for this purpose. Total power of all the electric motors will be 2,400 kW, the total flow amounting to 4,300 L/min at an operating pressure equal to a constant pressure of 280 bar. The axial piston pumps are pressure compensated.

Fig. 147 shows the tank station accommodated in the same room. The usable content of the oil tank is approx. 24,000 litres. Screw spindle pumps are installed vertically and these pre-fill the operating pumps with 10 bar via a fine filter.

These tests are carried out in open circuit operation.

Here are two examples of test stands connected to a central oil supply.

Fig. 147: Central oil supply, tank station

Fig. 148: Central oil supply, motor-pump stations

Test stand for axle drives

This test stand is used for testing the axle head component group consisting of hub, brake disc, wheel bearings and steering knuckle or wheel carriers on the laboratory test road.

Fig. 149 shows the external loadings acting on the axle head, using the example of a driven wheel. In addition to the two wheel forces applied to the contact surface of the tyre i.e. the wheel load and lateral forces, there are also the longitudinal forces applied to the centre of the wheel i.e. the braking and drive torques. With these environmental factors such as travel wind and weather conditions have to be considered i.e. a supply of cold air must be blown onto the individual components in order to simulate normal driving conditions.

The test stand is designed according to this loading and can be seen in **Fig. 150**.

The stand comprises the rear axle shaft, a drum which accommodates the wheel rim, a loading lever to initiate the three radial forces, an oscillating mass and a loading unit. Thus all loading of a non-driven wheel is simulated.

With a driven axis a drive torque is generated in addition to this via the lateral shaft by means of the Triplex drive components.

The design of the test stand is based on the measurements of the axle head requirements on test routes and in extreme manoeuvres.

All values can be simulated depending on the period of time measured in the vehicle and can be reproduced exactly.

The complete test stand has a total length of 6 metres (**Fig. 151**). The through drive transmission line which, for the loading, starts at the axial piston unit, consists of a sub-structure made up of several sections with oscillating bearings.

An emergency brake is situated between the axial piston unit and the flywheel. A speed / torque measuring component with speed direction transducer is arranged next to the flywheel on the right.

The loading lever with wheel force cylinders contains the bearings for the external drum which is driven from the left hand side via the cardan shaft.

On the right hand side of the wheel is the continuation of the transmission line via the lateral shaft (vehicle drive shaft) and also another speed / torque measuring component.

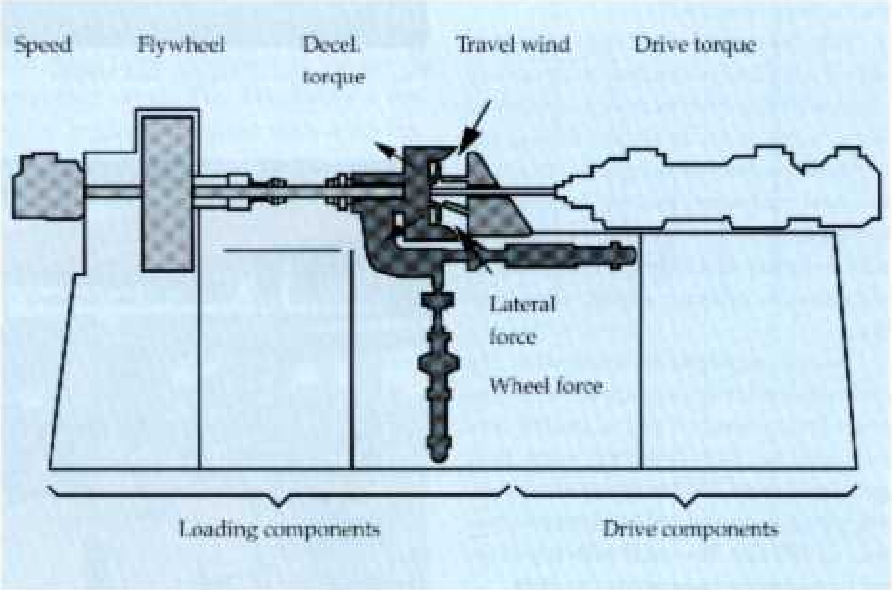


Fig. 150: ... and derived test stand concept

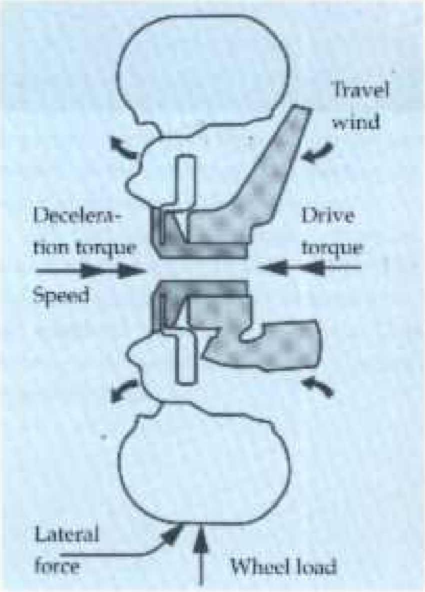


Fig. 149: Loading on axle head...

Fig. 151: Axle test stand

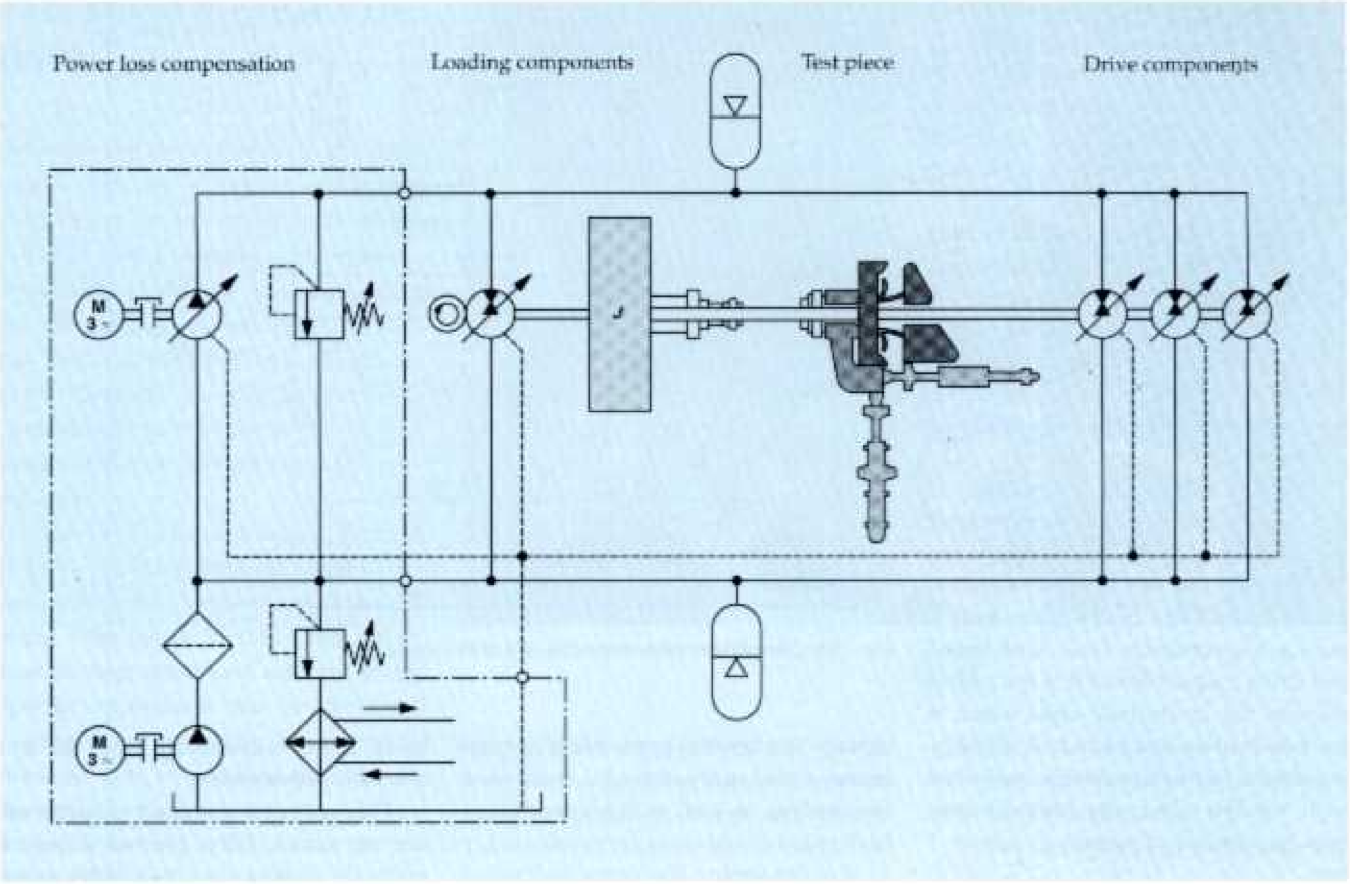


Fig. 152: Block diagram of axle test stand

Because of its rigidity the wheel disc was mounted with five clamping jaws on both sides in a two-part external drum for use as force input, corresponding to the vehicle tyres. The inner drum houses a clamping unit for the wheel disc. When operating the inner and outer drums are screwed together. For mounting purposes e.g. when changing the brake linings, the wheel complete with drive unit can be moved to the right with an integral sliding table. During this process the inner drum remains connected to the wheel disc. With wheels of varying sizes e.g. diameter, width, material, rim horn dimension, the drum and clamping unit must be matched. The external drum floats on bearings in the force input arm. This design principle was selected from several alternatives and has proved to be a good choice.

Finite element method calculations have shown that with the force system selected in the key area of the wheel disc the stresses on the test stand agree well with those in the actual vehicle. These calculations also demonstrated that, as a result of the radial forces introduced, any displacement of the drum axis or of the point of application of force does not increase to beyond the maximum permissible level.

Adaptors at the points of application of force of the load lever permit adjustment of different wheel radii and shifting of the application of force.

As can be seen from the overall design of the test stand, the concept selected is only feasible with hydrostatic drives.

Block diagram **Fig. 152** shows that the axle drive test stand is a back tensioning test stand with the possibility of re-utilising the braking energy.

The triple combination on the drive side was selected because of its dynamic characteristics, as three small units can produce a much higher dynamic response than a single one of equivalent size.

In normal operation with small drive torques the torque is generated by the first unit, the other two motors running without any torque.

With extreme manoeuvres e.g. rapid start-up operations, all three motors are simultaneously controlled and combine to generate the torque.

The four hydrostatic units in swashplate design are driven from the central oil supply for several test fields. If only the loaded side of the test stand is driven i.e. non-driven axes tested, under certain operating conditions the hydraulic motor will operate as a pump and feed back to the ring main. If a single driven axis is tested both sides are coupled together via an energy interlink block. In this case there is a mechanical back tensioning of the drive train. Only the power losses are drawn from the hydraulic ring main.

The flywheel in the mechanical section was designed for the manufacturer's smallest vehicle. When testing larger vehicles an increased inertia is simulated by means of the load unit using $M_I = J_g \cdot \omega$

The hydraulic accumulators situated on the high and low pressure sides in the hydraulic circuit increase system dynamic response and, with the help of a boost pump, prevent cavitation from occurring on the low pressure side.

Measurements are restricted to the determination of the radial forces at the load lever and are carried out by means of force measurement devices on each hydraulic cylinder.

Two specially developed measuring elements are installed for the determination of torque and speed. These are especially suitable due to their compact size and insensitivity to lateral forces.

The closed loop control is made up of components available from analogue computers.

All closed loop circuits operate on the cascade control principle, in which a directional control is subordinate to the force control circuits. With deceleration torque control the braking pressure is used as a subordinate value, with speed and drive torque control it is the swivel angle of the hydrostatic units which is used as a subordinate value. A specially developed frequency-voltage converter with variable filter characteristic permits the control of speeds as low as 1 rpm.

The control of deceleration torque and drive torque is of particular interest with the drive axle test stand.

Fig. 153 shows the comparison between command and actual values for these torques. With deceleration measurement (top curve) the clearance of the brake piston must first be overcome. This leads to unavoidable overshooting to a greater or lesser degree, depending on the control setting.

The simulation accuracy of the drive torque (bottom curve) is, on the other hand, considerably better. Great torque variations are not simulated to their full extent. The recognisable phase differences between command and actual values are caused by the relatively high flexibility of the drive train and the high dynamic response of the hydrostatic units.

The deviations observed here have in the meantime to a large extent been eliminated by installing ITFC software for iterative compensation of the transmission functions.

The axle drive test stand was designed for endurance testing. Testing of a brake disc, for example, requires several days solid testing. This meant that the test stand had to be fully automated. This applies to the areas of command value pre-setting, control and monitoring, as well as the interception of data to be measured and evaluated.

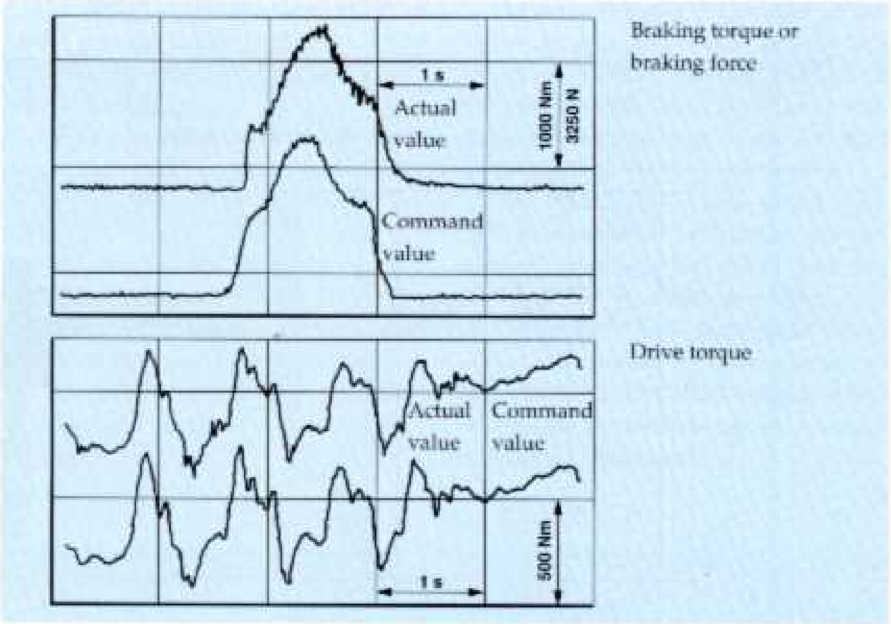


Fig. 153: Comparison of command and actual values

For pre-setting the command value the measurements taken in the vehicle are stored in the process computer memory. The computer outputs the command values to the analogue computer via an interface and receives the actual values back. The actual values are classified and monitored in the computer.

A PLC carries out the control and monitoring of the test stand. The stand is connected by means of a control panel and access is gained to the various functional operations (set-up, operation with load unit, operation with load and drive unit etc.). The control demands a specific flow from the central oil supply. After the relevant hydraulic pumps have been connected the demand is acknowledged and the test stand can be connected to the pressure circuit by means of the connection/cut-off block.

The second important function of this computer is to monitor the test stand whilst in operation. Values to be monitored include temperature of the bearings on the load lever and on the oscillating mass, the temperature in the tank of the brake fluid unit together with its fluid level, speed of the hydrostatic units, the wear sensor on the brake lining and leakage from the hydraulic components.

This control is also used to switch off the test stand. There are two different methods of achieving this. With rapid stop the hydraulic motors are used to bring the drive to a standstill. This operation is carried out when for example the wear sensor of the brake lining switches. In this case the test piece must not be damaged by this action. With an emergency stop the emergency brake will also be put into action and the load lever will be returned to the mid-position by means of the centering cylinders. After the drive has reached a standstill the test stand will be hydraulically separated from the ring main. The drive will be brought to a standstill in ≤ 4 sec at maximum speed (2200 rpm). The emergency stop is effected if for example there is a rupture in the drive shaft. In this case the test piece is essentially overloaded.

The axle drive test stand described combines for the first time the functions of a stand for testing deceleration of an oscillating mass with that of a multiple axis axle loading test stand. The universal design permits comprehensive testing of all braking operations for round journeys, short journeys, sequential stopping and high speed braking as well as operational stiffness tests on all parts of the axle head.

Test stand for rear axle drive

Research and development in the automotive industry is concerned not only with design and relevant calculations, but also with the method of computer simulation of driving and tests using the test stand. As each of these methods has both advantages and disadvantages, these can actually be utilised to complement each other in the various stages of vehicle development and construction.

There is a noticeable trend, however, towards reducing the time required for road testing by increased use of the test stand. Prerequisite for this is that the actual driving conditions relevant to the specific application can be simulated exactly. In other words the test stand must be able to produce the time sequences of velocity or motor speed and power or torque, as these occur during actual operation of the vehicle.

To achieve this goal a combination of processes are being increasingly used, thus bringing out the advantages of computer simulation with those of the test stand. A test stand is controlled by a computer on which the dynamic response of the complete vehicle system is simulated. Such a process is known as system simulation using test pieces. This method permits the dynamic response of a test piece to be tested in combination with the complete vehicle system. This way it is possible to optimise matching of the system at a point in the development at which the vehicle is not yet available with all its components.

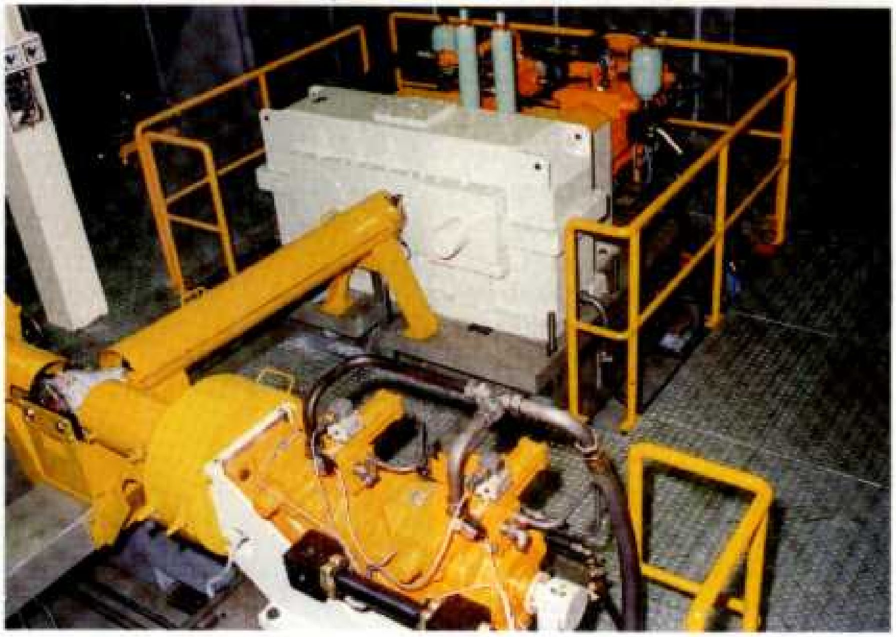


Fig. 154: Rear axle drive test stand

High demands are also placed on the dynamic response of the test equipment for solving this task.

These factors played a decisive role in the design of the rear axle drive test stand shown in **Fig. 154**, as the dynamic response of the secondary controlled unit had been proved to be sufficient in other applications.

This illustration shows the overall structure with two secondary controlled

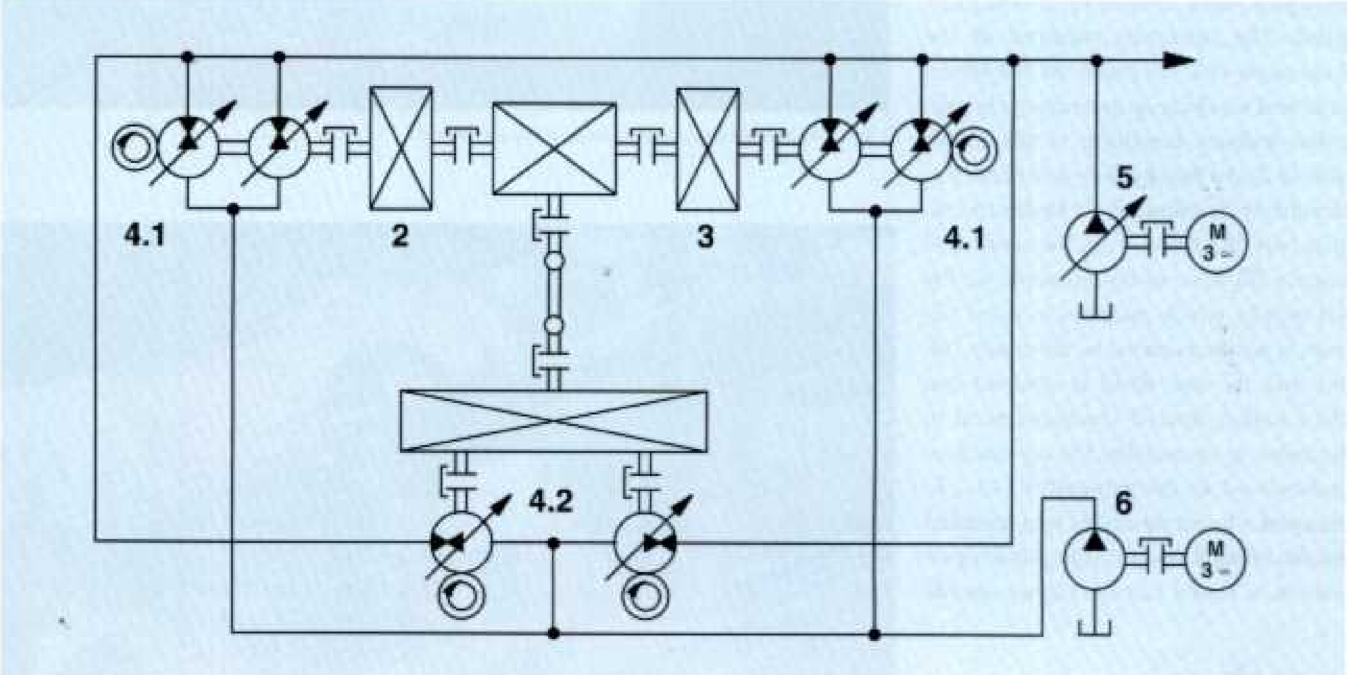


Fig. 155: Circuit diagram showing principle of hydraulic back tensioning

axial piston units working as motors and connected in parallel via the transmission. The internal combustion motor is simulated here.

The tandem version was chosen for the load units to enable the required drive power to be generated for the wide speed range.

All axial piston units on the drive and load sides have a displacement of 250 cm^3.

It would also have been feasible to use a single unit of 500 cm^3 displacement instead of two, although this would have limited the dynamic response.

The conventional differential gear to be tested has the task of distributing the torque generated by the internal combustion engine evenly over the drive shafts and hence over the wheels. It also has to compensate on corners for the speed difference between the wheel on the outer curve and that on the inner curve.

As far as the control concept of the load unit is concerned, this means that the control of both speed and torque has to be possible. This concept also permits testing of the limited slip differential gear and of the locking differential.

Fig. 155 shows that the principle of hydraulic back tensioning is being applied. The operating medium of the load units (4.1) arranged on the wheel side and working as generators, is supplied without throttling to the motor side (4.2). An energy flow will occur hydraulic → mechanical → hydraulic, so that only the losses from the central oil supply (5) have to be replaced. As the oil supply, which operates in open circuit, is accommodated in the pump cellar, and the test stand is situated one floor higher, special measures need to be taken to ensure that the oil can flow unhindered to the generators (4.1). To this end a boost pump (6) was installed on the oil tank. As this pump also operates in a closed circuit, higher speeds

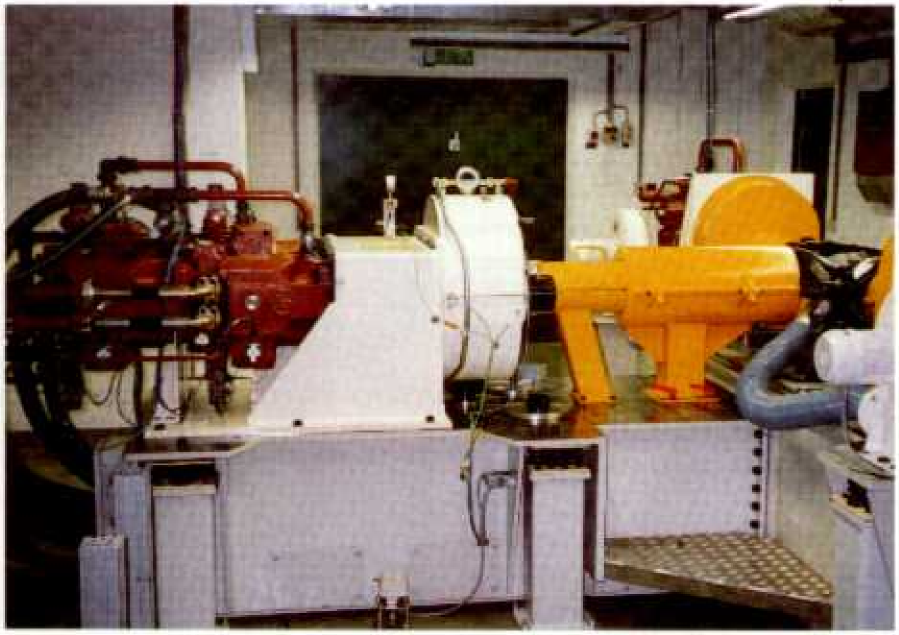


Fig. 156: Rear axle drive test stand

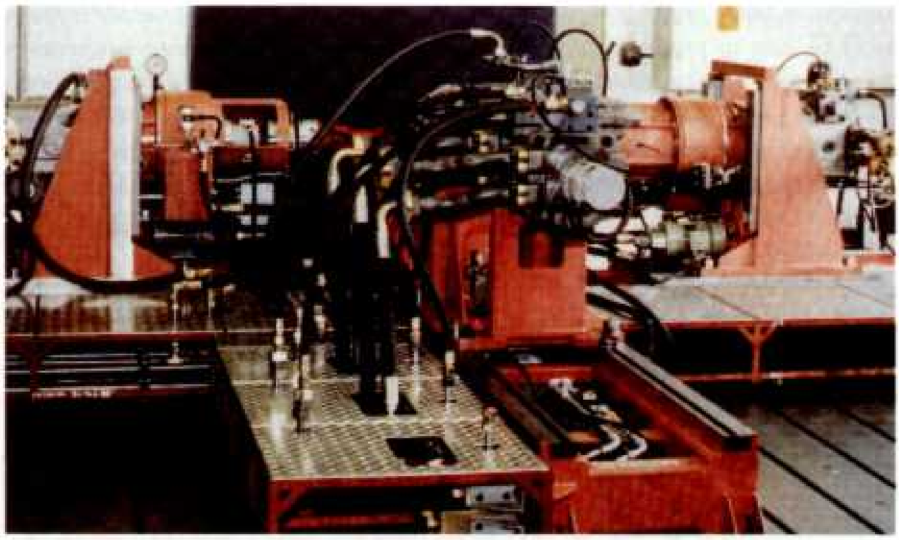


Fig. 157: Overall view of rear axle drive test stand

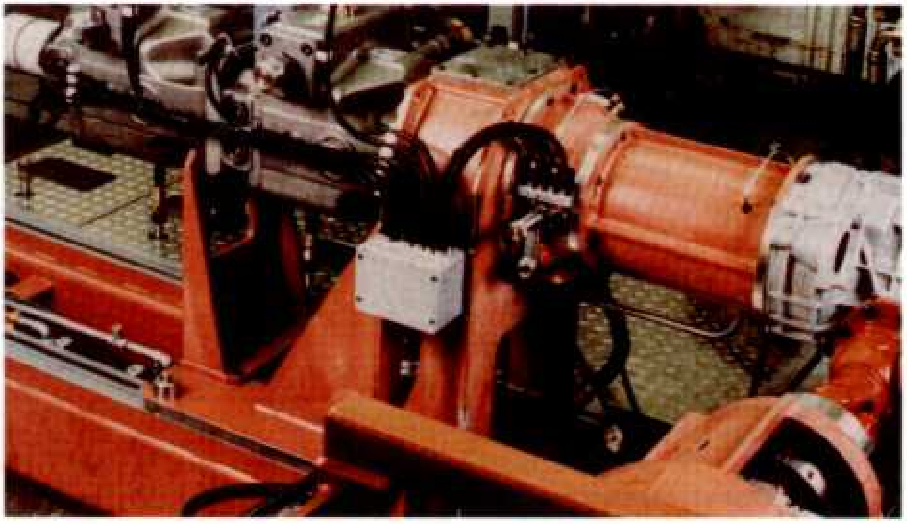


Fig. 158: Drive side of rear axle drive test stand to Fig. 157

are achievable within the permitted range of this circuit.

In this case, however, the speeds required by the test piece were considerably higher, and therefore additional matching to the secondary unit had to be carried out via transmissions (2) and (3), which were connected in series.

Fig. 156 shows a slightly different design of test stand for rear axle drive in the same development centre.

Additional transmissions connected in series are also necessary with this version.

In **Figs. 157** and **158** we can see the test stand from another manufacturer for a rear axle drive.

This also has planetary gears connected in series, which can actually be by-passed by means of a dog clutch when at a standstill. The two speed ranges produced cover a wide conversion range of vehicle transmissions economically.

The torque and speed measuring shafts are also built into the planetary gears, thus saving on space and additional moments of inertia. This also prevents any problems with balance, especially at high speeds.

As we can see by looking at the drive side, where the internal combustion engine is simulated, the design offers a high energy density with extremely low moments of inertia and associated high dynamic response. Two A4VSO250DS axial piston units are used, in tandem with a directly connected tachometer. The braking units on the wheel side from the same series feed the braking energy back to the drive units, so that only approximately 30% of the power flowing via the test piece has to be supplied externally.

This test stand has an oil supply of its own.

It is also possible with this variant for the three secondary controlled drive or braking systems to be either speed or torque controlled. The speed differential between both output drives on the wheel side are pre-set by means of an additional closed loop control process. In order to cover the total conversion and speed ranges when testing automatic and non-automatic transmissions, in addition to the two speed ranges of the switchable planetary gears, matching of system pressure is effected in the high pressure side of the hydrostatic unit.

The technical data required for the test stand are as follows:

- Four quadrant operation including zero speed at maximum torque,
- Maximum speed gradient of the hydrostatic unit 3000 rpm per sec referred to the test piece (the matching of the planetary gear by gearing must be taken into account),
- Differential speed between both output drives must be variable with approx. 2000 rpm per sec and
- Torque of both output drives must be variable with approx. 50000 Nm per sec.

In this way the system data will be kept within the limiting values.

Test stand for an all-wheel drive

Using the test stand for an all-wheel drive as shown in **Fig. 159** tests can be carried out considerably more cheaply than on the road or on the test ground. Some tests are only possible using the test stand:

- The test runs are independent of weather, traffic and driver and they demonstrate high reproducibility.
- The load and speed profiles are independent of distance profiles and traffic conditions.
- With service life tests considerable time can be saved, as the load collectives can be collated as required.
- The complete test procedure can be automated.
- Measurements are easier to carry out than on the road.
- A complete vehicle is not required - a considerable time-saver, especially during the development stage.
- Parameters can be entered to pre-set various loading conditions such as total weight, axle load distribution and trailer operation.

All these factors lead to considerable saving on development time and costs.

Fig. 160 shows the complete assembly of the test stand without primary station. The internal combustion engine, which could also have been replaced by a hydrostatic unit, was left here, to enable the test piece to be correctly loaded in terms of dynamics.

All four outputs of the all-wheel drive are either speed or torque controlled. The secondary units, which are of tandem design, include torque measurement shaft, tachometer and centrifugal switch fixed onto modular housing units standardised in the factory. They are connected to the test piece via torsionally stiff couplings.

They are connected on the primary side by means of the hydraulic isolator. This can be used to connect the secondary controlled axis so that it can drive or decelerate any other machine.

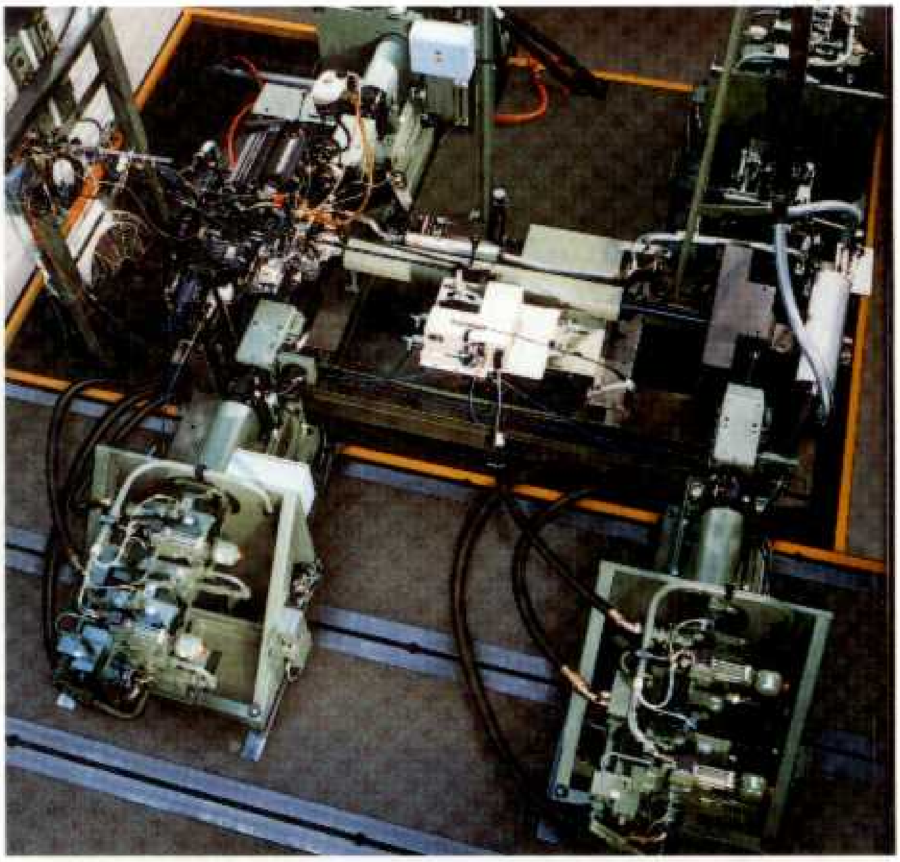


Fig. 159: Test stand for an all-wheel drive

Fig. 161 shows a tandem version of a hydrostatic load unit. The test stand manufacturer standardised the complete A4VS series internally from 40 to 1000 cm^3, and can therefore cover a power range of up to 800 kW or alternatively 1600 kW if using tandem units.

Extended time tests should preferably be carried out on the test stand in simulator operation. Pitch, rolling and air resistance tests must be simulated, together with accelerating and decelerating power of the translatory vehicle movements. Special emphasis should be placed on the exact distribution over both drive axes. A typical load collective for an all-wheel drive vehicle includes testing of the axle compensation transmission and the clutch or differential controlled torque distribution over the front and rear axles as well as the associated locking mechanisms.

With all tests the simulation of the slipping wheel plays a significant part, whereby slip conditions can be set via parameters on the simulator. The simulation must guarantee correct torque distribution, even when the locking mechanisms are in operation.

The load units are freely positionable on a subframe, and the dimensions of different types of vehicle with respect to wheel base and track width can be covered. The test stand is designed in such a way that vehicles with 2-wheel or 4-wheel drive can be tested. There is no limit with respect to torque distribution between the driving axles.

The primary station shown in **Fig. 162** is also of modular design and built to internal standards as stipulated by the test stand manufacturer. The same applies to the hydraulic accumulator blocks, the control electronics, test stand control and computer control, so that every application can be covered for dynamic rotary drives of modular design.

This standardisation and its possibilities with regard to standardising components and component groups is an important step in system technology. Even with respect to price it permits substitution of controlled electrical drives.

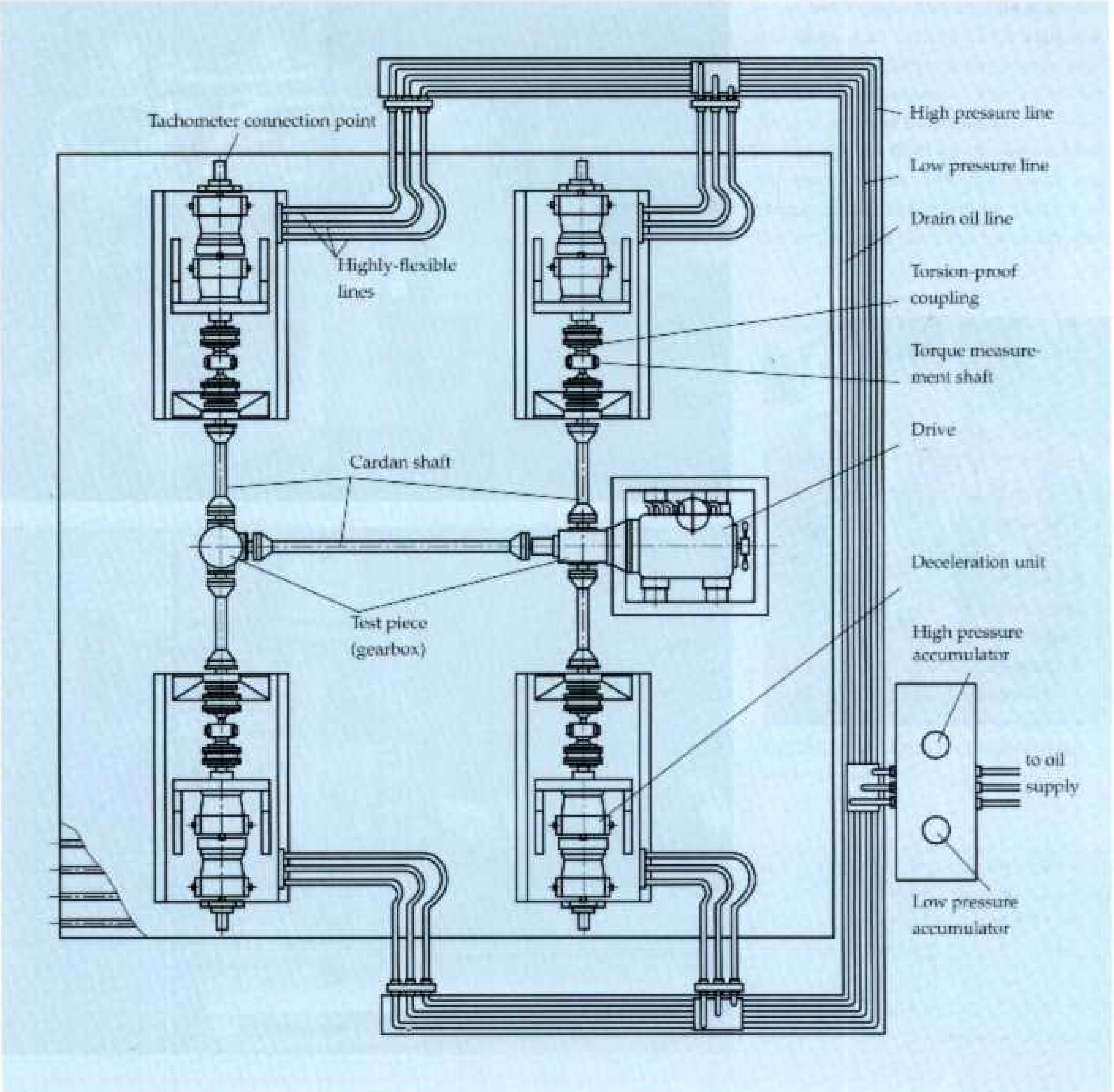


Fig. 160: Complete assembly of the all-wheel drive test stand

The technical advantages are plain to see.

Conventional simulation methods using D.C. or A.C. power converter fed motors are either very complex and expensive or else they are not capable of simulating slipping wheels. Especially with the 4-wheel drive test stand the moments of inertia of the electrical machines disturb the slipping simulation. The moments of inertia would have to be completely eliminated in the simulation. This is hardly feasible on the grounds of power and stability and the associated loading of the electrical power supply.

If this is carried out by hydraulic means it presents no problem.

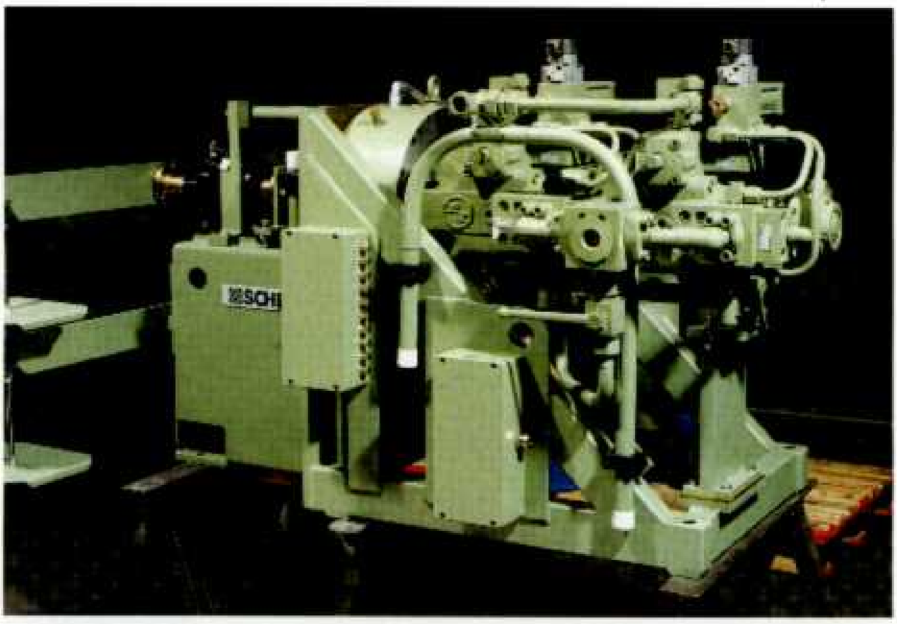


Fig. 161: Loading machine in tandem design

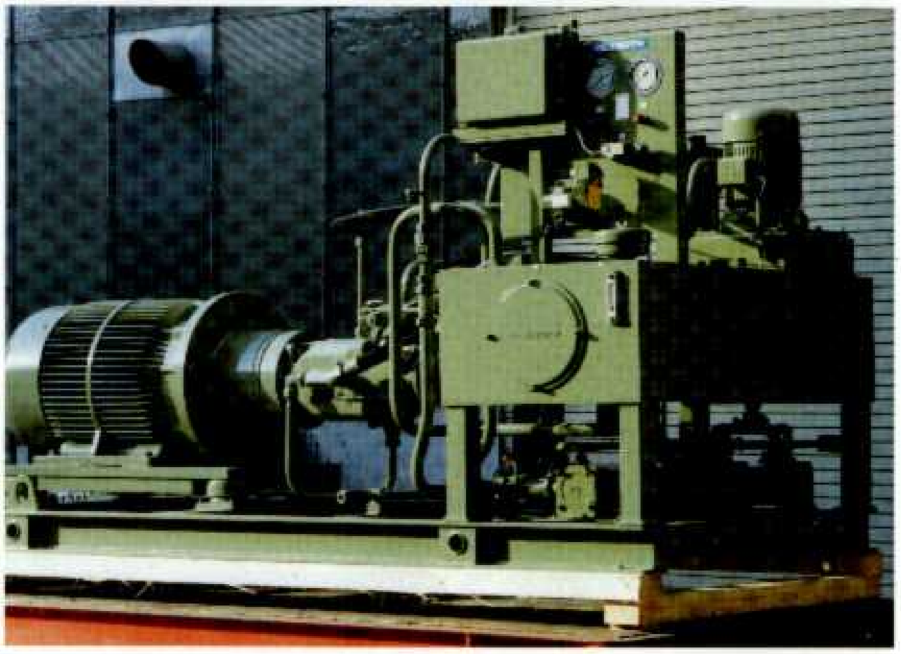


Fig. 162: Drive station

Test stand for automatic transmissions

From the functional viewpoint automatic transmissions can be divided into a transmission section and a control section. The important control range increasingly contains an electronics section which determines when and how the switching operation is to take place.

The actual transmission section consists of the converter, possibly a converter by-pass, the planetary sets, the

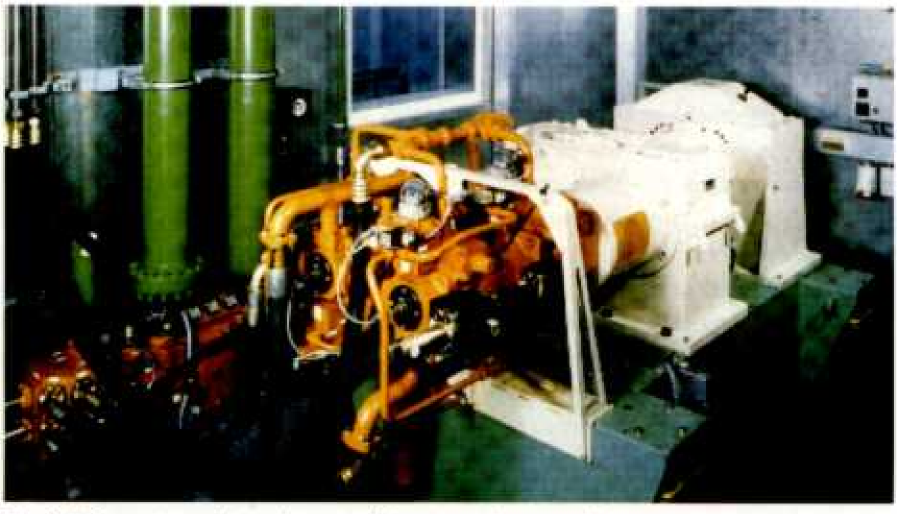


Fig. 163: Generator section of test stand for automatic transmission

multi disc clutches or band brakes and the free wheeling elements.

This design permits gear changing without disturbance of the tractive effort under load.

Fig. 165 shows the schematic diagram of the functional test stand.

In range A the internal combustion engine is simulated by two axial piston units (1). Input power is a maximum 350 kW. An axial piston unit (2) is coupled to this to simulate the moments of inertia. By means of the torque $M_T = J_g \cdot \omega$ the moment of inertia of the internal combustion engine deviating from the design in question is simulated under consideration of acceleration.

The speed of the axial piston units can be infinitely varied between 170 and 2000 rpm, enabling a speed range of between 600 and 7000 rpm to be set on the test piece by means of the gear ratio.

The static input torque at the test piece is 500 Nm. This can be superimposed by a dynamic torque of

$M_T = \pm 100$ Nm.

The dynamic response determines the moment of inertia of the drive train from the axial piston units via the transmission to the test piece.

This amounts to $J_{total} = 0.094$ kgm² referred to the high speed shaft, the greatest part, $J = 0.0774$ kgm², being used up by the transmission and the cardan shaft.

Situated behind the test piece in area C is an oscillating mass of $J_1 = 70$ kgm², which represents the vehicle mass of the smallest car produced by the particular manufacturer. Two axial piston units (3) simulate larger vehicle masses and drive resistance. These units are designed for left or right hand drive by means of the cardan shaft and gearbox.

These act as generators and direct the energy back to drive train A. Only the relevant acceleration power required and the losses from the test piece and the hydrostatic units are replaced by primary units (4) and (5).

The impressed pressure in this case is 280 bar.

Fig. 164 shows the drive side of the test stand.

In the foreground are the axial piston units with control block substructure, which will cut off the energy supply in an emergency. The units are flanged onto the transmission.

The cardan shaft can be seen behind the transmission, with the speed and torque measurement shafts in front of the entry point into the test piece, be-

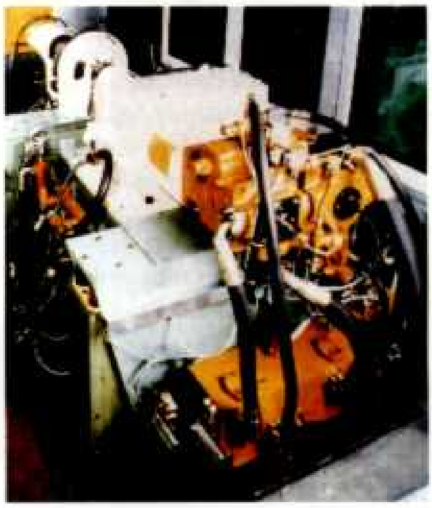


Fig. 164: Drive side of test stand for automatic transmission

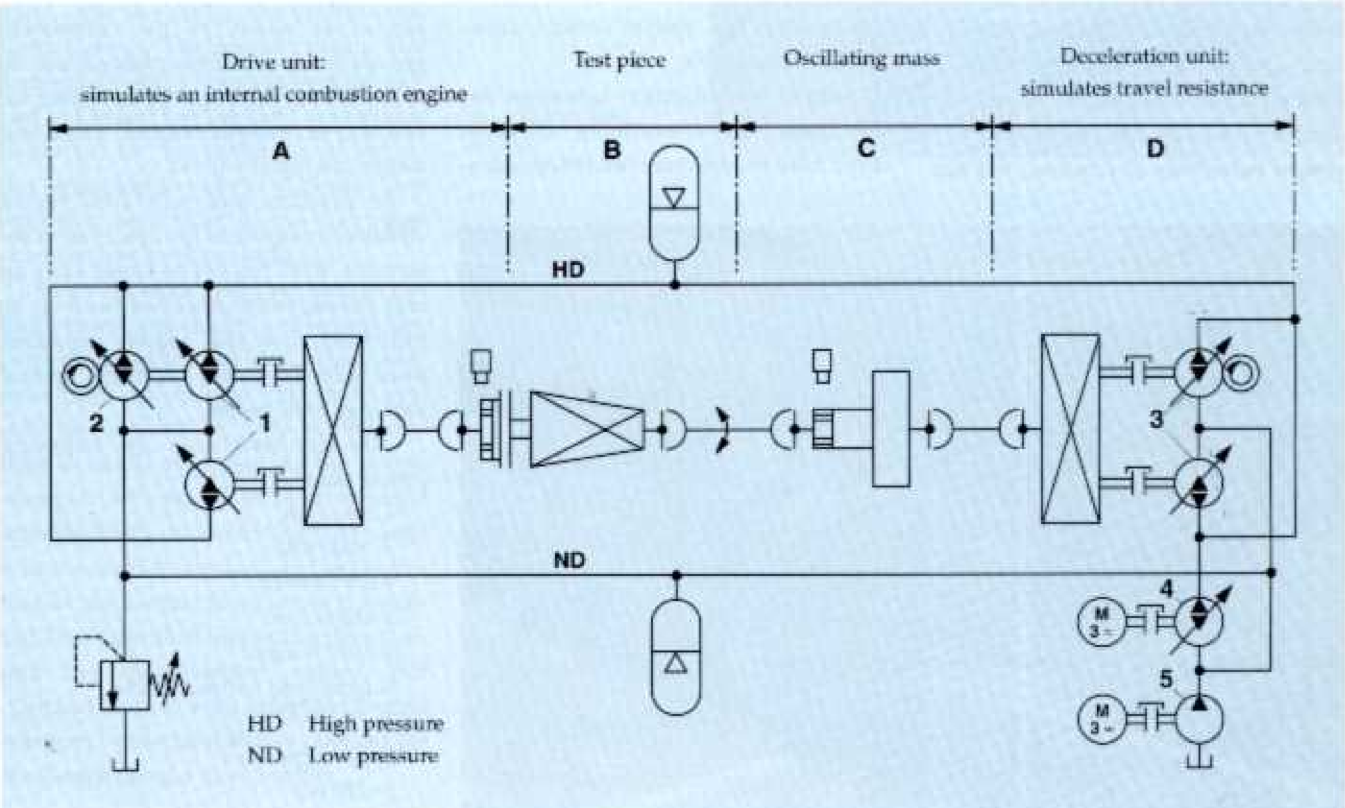


Fig. 165: Circuit diagram of test stand for automatic transmission

hind which both values can be measured again.

The generator section (**Fig. 163**) is separated from the drive train by a partition wall.

Both hydraulic accumulators for the high and low pressure areas are fixed to the wall in the background.

This test stand has proved that simulation of mass and drive resistance is realistic and reproducible.

The fact that hydrostatic drives are in the foreground of the highly interesting field of test stands for the automobile industry is not only due to the high dynamic response. Significant developments in the last decade have led to considerable improvements in the reliability and the price / power ratio of the vehicles put into operation.

This trend has been helped greatly by the corresponding developments in the electronics field.

Experimental dynamic response test stand for identification of drive trains

When designing drive trains it is important to know the dynamic behaviour of the individual components, as the dynamics have a considerable effect on the system behaviour of powered machinery, drive trains and mobile machines. Digital computers can be used to simulate system behaviour, if exact mathematical models of the individual components such as shafts, flywheel elements, flexible couplings or frictional couplings, transmissions etc. are available.

Two drive train components, about which little is known with respect to dynamic behaviour, are the hydrodynamic torque converter and the hydrodynamic coupling.

In order to produce mathematical models of these two components a high dynamic response test stand (**Fig. 166**) was built at the Ruhr University of Bochum within the framework of a special research project. This test stand was equipped with secondary controlled hydrostatic units, due to the great demands to be placed on it.

Once again the hydraulic back tensioning principle was applied using drive and load units.

The drive unit is speed controlled, the load unit being torque controlled. The hydrodynamic test piece also permits simultaneous speed control at both hydrostatic units.

Simultaneous torque control, however, is not possible.

System identification processes are the basis for determining dynamic drive train models for simulation calculations. The first step is to stipulate a model application that is then, in the second step, matched and optimised to real systems by taking measurements on the test stand.

The identification of systems, that can be described by means of linear models, is carried out mainly over a set frequency range. The real system is excited at a fixed frequency and the response amplitude given a relationship to the exciter amplitude. This process is repeated for different frequencies, so that after taking a set of readings the frequency response is available in nonparametric form to be matched to the model.

Conditions for this type of identification are rapid adjustment of drive and deceleration units, dynamic response control and sufficiently large acceleration and deceleration torques, in order to generate harmonic excitation with exactly defined amplitude and frequency.

Non-linear component groups, such as the hydrodynamic couplings and converters already mentioned, are formed by linearisation around the operating point. By special excitation of torque or speed of the component groups even non-linear models can be achieved, so the requirements here for fast and accurate control of speed and torque are equally great.

In practice, especially with highly dynamic component groups, system behaviour with respect to rapid start-up and deceleration processes need to be examined. For this reason provision must also be made for these special loadings to be tested.

The test stand has the following power data:

Drive unit
P =200 kW
M_T=1060Nm
n=1860 rpm
J=0.0959 kgm^2
Supply unit (primary side)
P=85 kW
System pressure
p=280 bar

Fig. 166: Experimental test stand

Dynamic response example:

For an inertial mass of the drive train of J = 1.56 kgm^2 the following loadings can be driven:

- sinusoidal speed variations at 15 Hz n = 1000 rpm ± 50 rpm
- sinusoidal speed variations at 2 Hz n = 500 rpm ± 500 rpm and
- speed ramp from 0 rpm to 1000 rpm in 0.2 sec.

Analogue closed loop control of both equal-sized axial piston units is carried out in each case with a PD swivel angle control, which can have alternately an analogue speed or torque control connected in series.

The command values for the speed control, torque control or directly for the swivel angle control are pre-set by the digital computer.

System monitoring is by means of a PLC, which both controls the system and monitors speed, control variations, oil temperature, filter status as well as the combination of controls. Critical system conditions such as emergency operation will cause the system to shut down.

The actual command value settings, the measurement of data and their evaluation are all carried out by the computer system. The measurements and command value settings have to be carried out in real time. A fixed time grid is important for the evaluation of the input and output values, a closed loop operation permitting digital control or characteristic programming.

So far dynamic response tests of hydrodynamic converters and couplings, carried out on the test stand described, have been satisfactory. The drive system has proved to have sufficiently high dynamic response. As the drives always have to be driven right up to the stability limits due to the high dynamic response that is required, optimum setting of the control parameters is essential. Oscillations of the control circuit affect the measurement values, but digitised control with adaptive control concepts, something that is planned for the future, should improve this.

Rear axle acoustic test stand

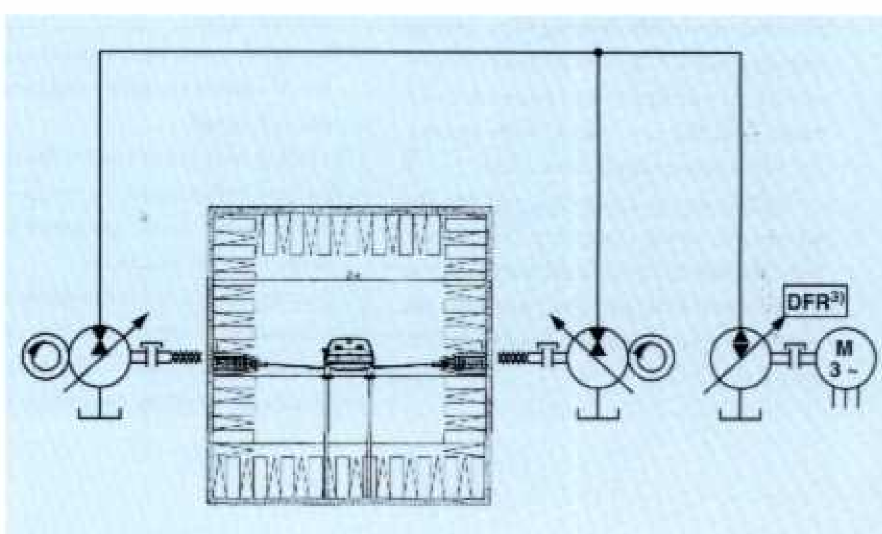
The ever increasing demands for protection of the environment and at the same time for comfort mean that automotive manufacturers are having to reduce noise pollution. Continuous development of this is, however, only possible on the test stand, which can carry out acoustic effects on the vehicle free from disturbances and with good reproducibility.

Such test stands must be capable of operating to the physical limits of sound propagation. They must also guarantee location of individual sound sources, the measurement of sound and sound direction characteristic, sound absorption and body insulation for the test vehicle to be driven under load.

Fig. 167 shows one of several sound test stands of one car manufacturer, the so-called free field chamber, which is equipped with secondary controlled hydrostatic units.

The free field chamber is fitted with a glass fibre absorption lining. The low-frequency tuned wedge-shaped glass fibre units are covered with a glass silk coating to prevent the transmission of air and impact sound from the environment. This test stand guarantees the "free" i.e. undisturbed sound radiation from the test piece in all directions, especially downwards. This is possible as all reflecting areas essential for operation of the test stand, are restricted to a minimum in number and area.

Fig. 167: Free field chamber



3) Pressure-flow control

Fig. 168: Hydraulic circuit, simplified version

In order to fulfil all requirements the vehicle must be "suspended" in the chamber. This is achieved with the help of a special steel structure with two driving tracks of welded round steel rods. A net made of steel rope was suspended as a walking area and for the area around the vehicle. Steel girders were used as supports with oscillation-damped bearings on the bare floor.

Suppression of secondary noise, which distort the measurement results, was the main task of the test stand designer. Thus a secondary controlled hydrostatic drive with four-quadrant operation was chosen, and not only because of the dynamic response and the fact that the requirements were relatively easy to fulfil. The loading unit could be integrated and sound-encapsulated into the absorption lining of the walls (**Fig. 168**). Heat accumulation does not occur when the unit is completely enclosed, as the heat is removed by the fluid.

Both hydrostatic units are connected to the drive axle of the vehicle via an axially moveable intermediate shaft and a final cardan shaft drive to match the various track widths. Special rims with inside bearings ensure through drive when the vehicle and wheels are stationary.

The driving simulation of traction and overrun permits simulation of uphill and downhill driving. The energy requirement for downhill travel is generated by a primary unit situated externally from the test stand, both secondary units being speed controlled.

Uphill travel is carried out by the secondary units operating as generators. The energy produced is directed to the primary side and fed back into the electrical power circuit. Recovery efficiency is ≤75 %.

Test stand for a tractor

As with passenger and utility vehicles, tractors have also attained a high quality standard and advanced development stage. For this reason a well-known manufacturer of farming tractors ordered a test stand, illustrated in **Fig. 169**. The test stand permits exact measurement of sound emissions on the tractor as a whole and fulfils all requirements with respect to dynamic response, setting accuracy and repetitive accuracy.

It is designed for a diesel motor power of 250 kW, that can be decelerated either on its own or in combination. It offers the following possibilities:

- all-wheel drive,
- rear axle drive and
- power take-off.

All hydrostatic loading units can be driven in four-quadrant operation. The following conditions must also be fulfilled:

- It must be operable up to a drive speed of 60 km/h, a rear wheel loading of up to 25 kNm and a power take-off loading of up to 2.5 kNm.
- The front axle unit must be usable on the cardan shaft without needing to extend the front axle.
- The front axle unit must be capable of carrying out the function of power take-off unit.
- The units must be accommodated in the available chamber and be sound-encapsulated.
- Tractor power must not be "heated".
- The test stand must be open and be suitable for other possible loading units in other positions.

For reasons of space the primary station must be accommodated in a separate room in the cellar. The secondary units can be used as hydraulic motors i.e. as drive units. When decelerating they work as generators, the braking energy of the diesel motor being fed into the electrical power circuit.

Fig. 170 shows the secondary unit for a geared rear axle, an additional flywheel, a highly flexible coupling and an intermediate cardan shaft. The flywheel is situated in the coupling flanges from the axial piston unit to the gear.

Torque transfer to the tractor is by means of a 25 kNm coupling and extension shaft to the support bearings.

The moveable third unit the " front axle and power take-off loading unit" can be installed with the integral rolling-lifting device at various positions.

Axle height can be infinitely set between 250 and 700 mm above the ground, and without a lifting mechanism, by means of the lifting cylinder.

All secondary operations can be carried out

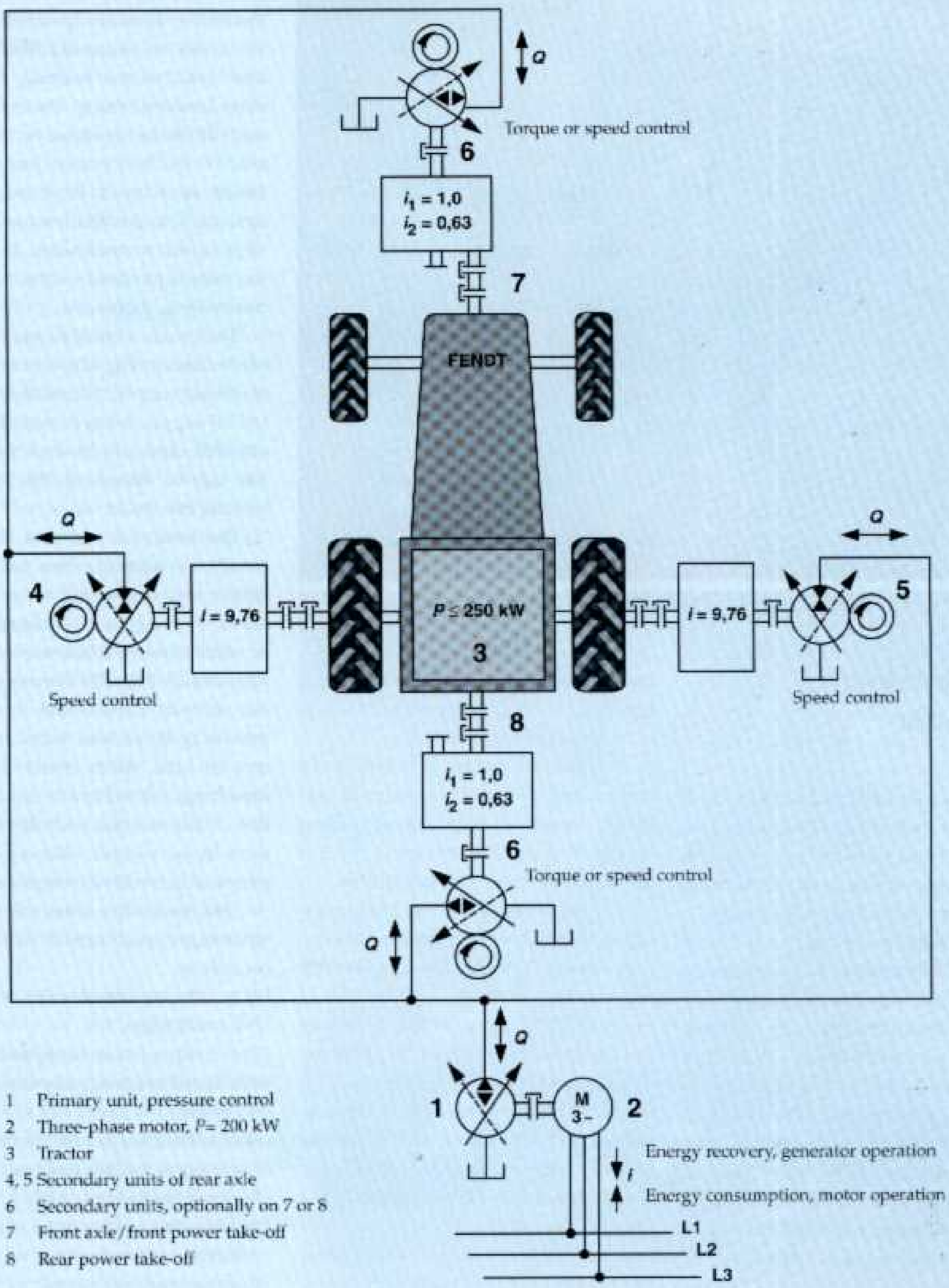
- individually and independently,
- in combined manual operation and inter-dependent or
- by computer.

Operation and monitoring takes place at the control box of the test stand measurement position. For on-site operation i.e. from the tractor itself, a portable control case offering limited individual operations and display is installed in the test chamber.

Detailed monitoring for the primary station is carried out directly at the station control box.

For test purposes the rear axle units can be driven in speed control, the mobile unit can be either in speed or torque control.

To ensure safety of the test stand all command values and all reaction actual values, limited in both directions, can be monitored.



1 Primary unit, pressure control
2 Three-phase motor, P = 200 kW
3 Tractor
4, 5 Secondary units of rear axle
6 Secondary units, optionally on 7 or 8
7 Front axle/front power take-off
8 Rear power take-off

Fig. 169: Hydraulic circuit, simplified version


Fig. 170: Secondary unit on the rear axle

Flat bed driving simulator

Flat bed test units have been well known for some time in the automobile industry. They are used as test stands for tyres and wheel rims, many tests having been carried out in this context.

Fig. 171 shows a flat bed unit driven by a secondary controlled hydrostatic unit and with an additional servo cylinder, with which disturbances from the road are transmitted to the wheel. The installation shown in **Fig. 172** can be used as a tyre test rig. The wheel is driven from the main drive belt by frictional contact with the tyre. The tyre held by the loading unit can swivel in two dimensions in relation to the direction of travel, meaning various angles of attack and different king pin inclinations can be simulated when the wheel is rolling. It is also possible to apply different forces to the flat bed in order to simulate any wheel loading required.

The disadvantage of this type of tyre test stand is that no traction or braking forces can be simulated when the wheels are running. A factor of considerable significance in tyre testing is thus ignored, meaning that realistic loading conditions cannot be tested.

In order for realistic tyre tests to be carried out, by generating traction and thrust forces at high wheel rolling speeds, it is possible to extend the test stand in which the vehicle is driven.

The circuit shown in **Fig. 173** was selected to minimise the primary energy requirement. Considerable importance was attached to energy consumption in this case. When the vehicle is travelling at maximum speed practically half the motor power is transmitted by means of the tyres onto the road, and the tyre tests extend over a long period of time. Both these factors mean that the energy consumption is quite considerable. For these reasons energy recovery and feedback into the hydraulic ring main, as already described with the back tensioning test stands is, therefore, beneficial. Both secondary units on the tyre or, rather, on the flat bed simulator, can be either speed or torque controlled. The tandem version of hydrostatic units on the flat bed simulator was selected because this version has a particularly low inertia. This means that rapid change of load, e.g. full braking or changing from traction to overrun operation and vice versa can be simulated without problem, without the rotating test stand mass seeming wrong. The rotating moment of inertia necessary on the braking side of the test stands can be maintained at a lower level than a mass equivalent to the smallest passenger vehicle of the manufacturer. In this way the mass equivalent of the real vehicle can easily be simulated.

The primary energy requirement, as can be seen in **Fig. 173**, is provided by a single oil supply. Connection to a central oil supply, if one is available, is also possible, especially as such a system today can be driven at 280 bar almost without exception.

The basic flat bed unit can be extended by coupling two flat beds together and connecting to an axle test stand. By using four flat beds a road can be simulated in the laboratory. This is illustrated in **Fig. 174**. A complete vehicle, securely fixed about its centre of gravity or by another suitable method, 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.

This simulation offers the following advantages over actual road driving conditions:

- It is independent of weather and traffic conditions,
- Test conditions can be reproduced,
- No instruments need be built into the vehicle,
- Test results are not influenced by the weight of the measurement devices,
- Direct evaluation of data,
- Vehicle and units can be observed at maximum speed and
- Constant test conditions.

The flat bed test stand of another automobile manufacturer, as shown in **Fig. 176**, is used primarily to test and improve on vehicle comfort. The dynamics of vehicle movement are also tested. Stability tests, especially endurance fatigue tests, do not constitute part of these tests.

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. Particular attention must, however, be paid to the drive and braking systems, as the speed of all four conveyors must be exactly the same. For this reason once again secondary control was used which could be connected to an available central oil supply. Highest priority was given to the test stand functions of vertical, longitudinal and sideways dynamics. This enabled the vertical movement operation to be transferred from the servo cylinder to the moving conveyor belt.

In order to fulfil this task, the conveyors are equipped with contact-free bearings. 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 to reproduce driving round curves, the two front units can 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 (**Fig. 175**).

One main argument for the use of modern test stands is the economy aspect, as both time and money savings are made.

Further advantages are the good comparability of test procedures and ease of parameterisation.

If we were to compare the test results using a test stand with those carried out under real driving conditions, we would see that the test stand results would be more accurate than those carried out on a manually controlled test vehicle with its much poorer reproducibility. Only the use of "driving robots" would improve accuracy.

One final thought emphasises the superiority of modern test stands. If optimisation in the development of particular vehicle components needs to be changed, such as transmission conditions, efficiency levels or flexibility, then this type of test stand can be adapted as

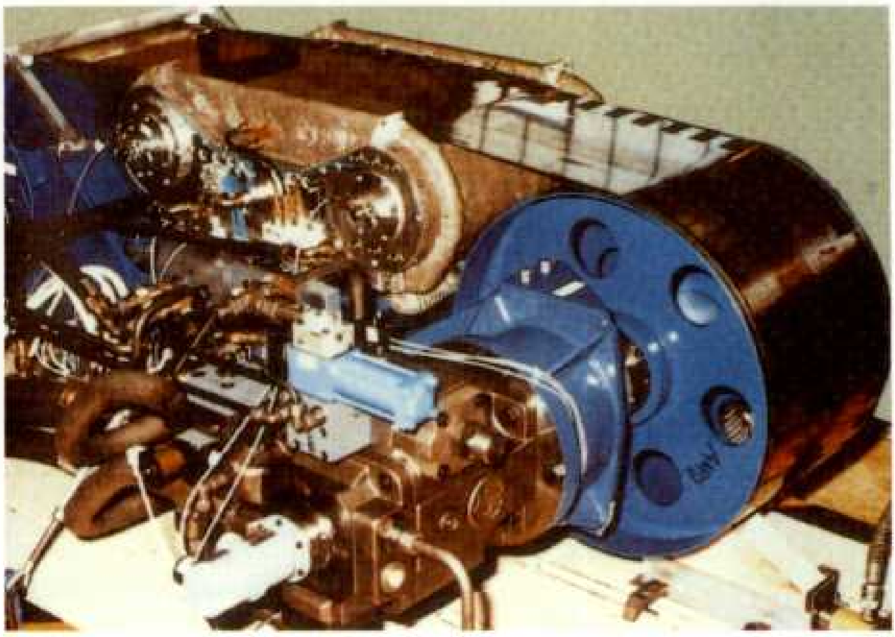


Fig. 171: Flat bed test stand

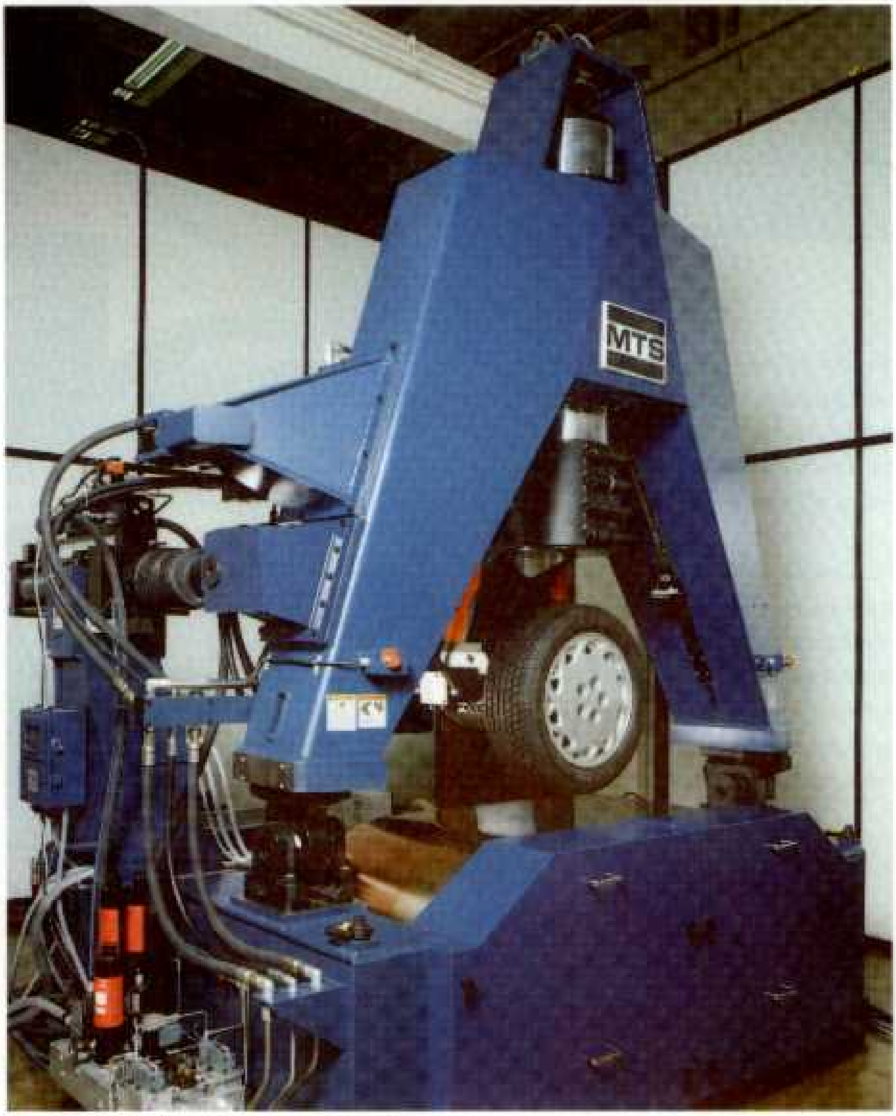


Fig. 172: Flat bed unit in a tyre test stand

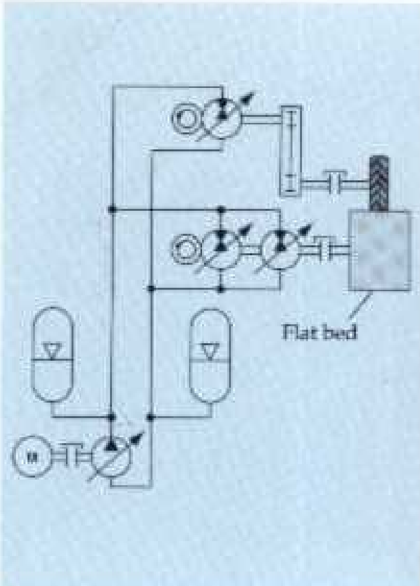


Fig. 173: Hydraulic circuit, simplified

Fig. 174: Flat bed road

required and in the shortest possible time, simply by changing the parameters either on the operating console or in the computer. In order to achieve this in road tests components would have to be changed, costing considerable time and money.

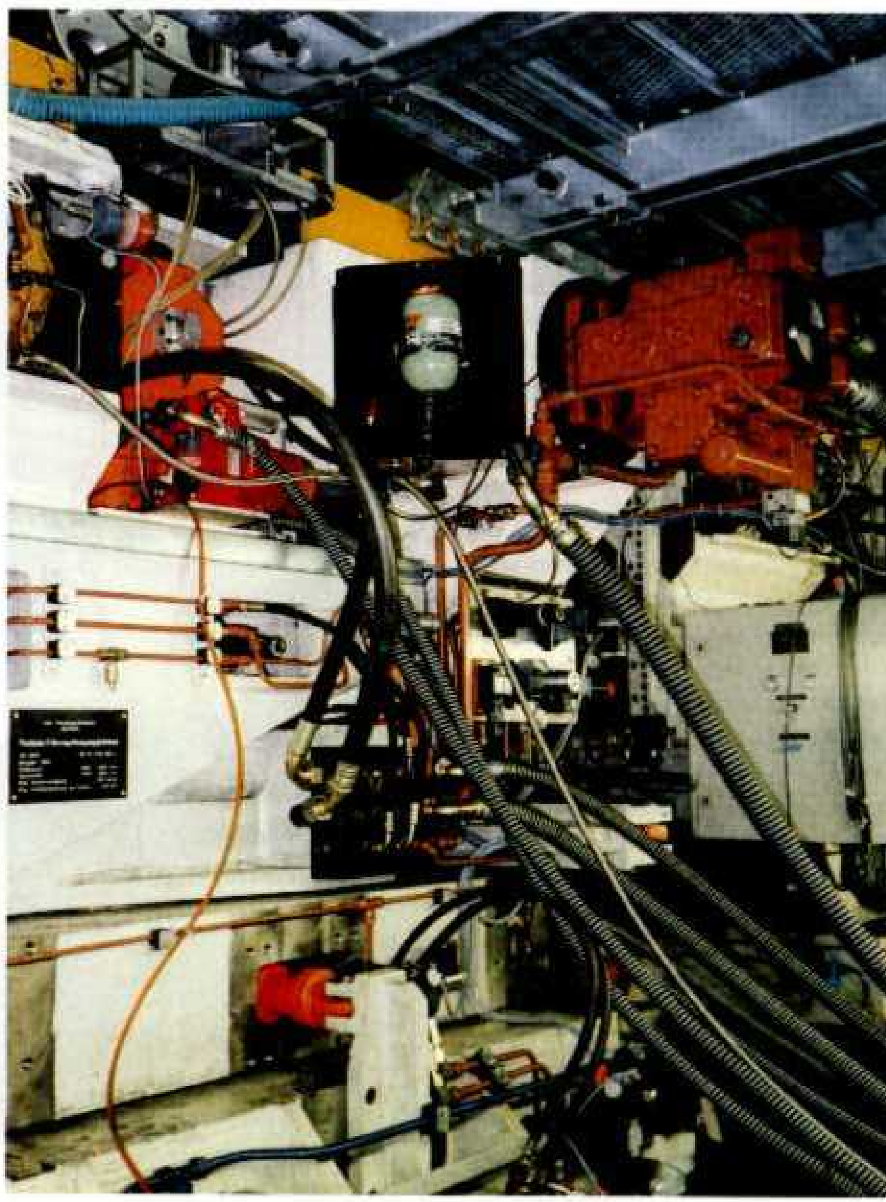


Fig. 175: Part view of flat bed road to Fig. 176

Fig. 176: Flat bed road

Summary

On the basis of the application examples selected using test and simulation technology we have demonstrated that secondary controlled hydrostatic units can be used together with computers in test procedures to simulate driving conditions. These test procedures permit the integration of vehicle components on the test stand into simulation systems, whereby the components not available as real parts can be simulated in their dynamic characteristics on the test piece components. This permits the effects of change between test piece and the complete system to be considered, even if the test piece components are the only ones available as real parts.

Over 100 test stands are now in operation, and so far no problems with respect to dynamics have arisen, and all requirements have been fulfilled.

This has resulted in secondary control achieving recognition in the last few years in the field of hydraulic control systems, where specific demands need to be met. System costs have thus been reduced around the primary station already available, which there again means continued application of this technology in the future, in automation for example.

Other features and advantages arising from this new control technology which must not be forgotten, include system monitoring and diagnostics, emergency control procedures, priority control for when required, monitoring of components and load collective measurements.

Low Loss Controls in a Hydraulic Ring Main System with Impressed Pressure

Low loss power take-off from a hydraulic ring main system with impressed operating pressure is best achieved with a rotary drive with variable hydrostatic units which can be swivelled "over centre". Utilising such devices, four quadrant operation 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 **Fig. 177**. 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 main actuator, a 4-way proportional valve (5), piloted via the tacho-unit circuit, is installed. The proportional valve now controls the pressure difference at the secondary unit dependent on 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 reduced at the proportional valve. Regenerative braking operation is also possible with this circuit. The losses are, however, higher than with a variable unit having over-centre control, due to the existence of the throttling valve in the energy flow line.

The same problems occur in a cylinder control in the same type of system, as the most effective solution involving steplessly changing the piston area is technically not possible. Three possible cylinder controls are shown in **Fig. 178**, **Fig. 179** and **Fig. 180**. **Fig. 178** shows a flow coupled system with control being achieved via the swivel angle of the pumps. Energy recovery is possible when the cylinders are being retracted. This energy is returned to the electrical power lines. **Fig. 179** shows a system with flow coupling by means of proportional valves. These take their power from pressure compensated pumps. The power loss here is dependent upon the pressure drop across the proportional valves. In the partial load condition, with high performance devices, this can be extremely high.

The most expensive, but at the same time the system with the lowest losses, is the system with pressure coupling

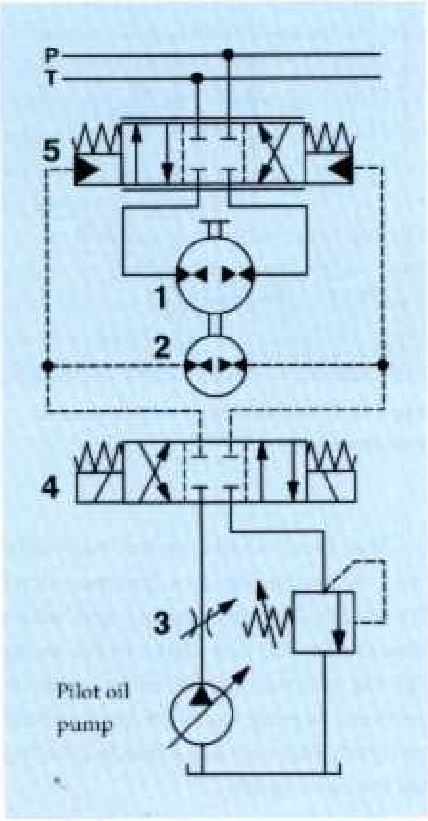


Fig. 177: Fixed displacement unit in a system with impressed pressure

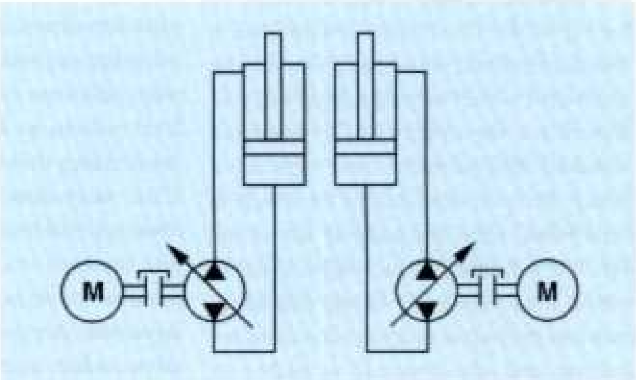


Fig. 178: Translatory hydrostatic drive; pump control (flow coupling)

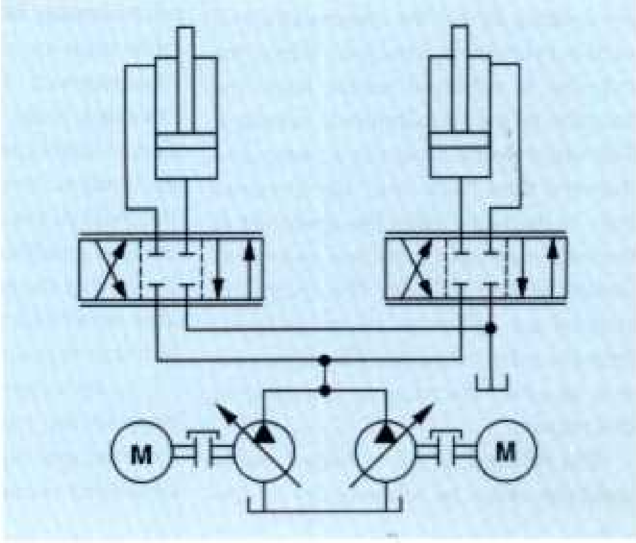


Fig. 179: Translatory hydrostatic drive; cylinder control (constant pressure system)

and hydraulic transformers shown in **Fig. 180**. In this circuit the losses are dependent solely on the efficiency of the transformers. This solution is, however, only economic when high-power and large units are to be employed e.g. on very large excavators or presses. As the cylinders are retracted, the potential and kinetic energies are stored in the accumulators in the common pressure line. The cooling power which must be installed is very much less than with throttling control.

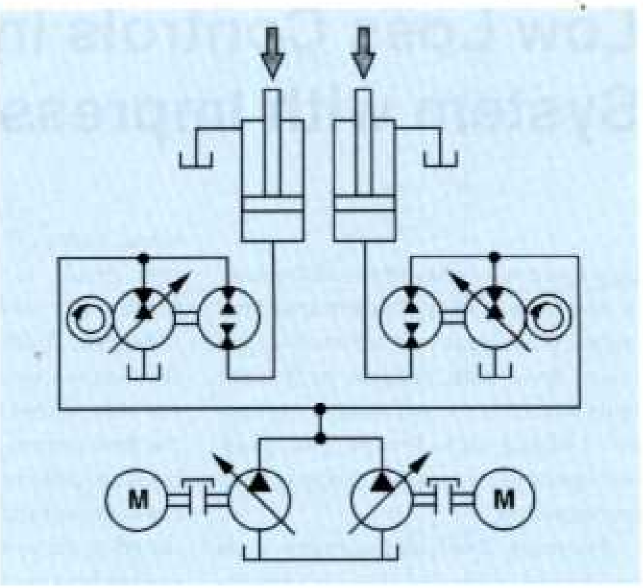


Fig. 180: Translatory hydrostatic drive; secondary control (pressure coupling)

The hydraulic transformer consists of a speed controlled variable displacement motor with one or more hydrostatic units of fixed or variable displacement. The units are coupled to each other and there is no mechanical input or output drive. The speed is pre-set by means of the secondary controlled unit.

Fig. 181 shows two possible cylinder circuits for connecting cylinders to a system with impressed pressure. These show a single acting cylinder with load in one direction and a double rod cylinder with alternating loads.

When the single acting cylinder at the top of the illustration is extended, a speed command value is given to the tacho-unit which in turn acts as a speed control for the cylinder. The fixed (or variable) displacement unit acts as a pump, the pressure output of which is dependent upon the load of the cylinder. The variable unit, acting as a motor whilst the cylinder is being extended, matches its displacement to the cylinder loading and the pressure available in the power line working under secondary control, so that the command speed of the cylinder is attained. When the cylinder is retracted under load, the function of the transformer is reversed. The variable unit works as a pump and delivers fluid back into the pressure line. At the same time, the speed set at the tacho-unit once more acts as a speed control for the cylinder. The energy recovered can either be stored or transferred to other actuators. The tacho-unit may be either electrical or hydraulic in this circuit.

The reversal of the cylinder movement can either be achieved by reversing the direction of rotation of the transformer or by reversing the swivel angle of the output device of the transformer, if this is a variable displacement unit. As reversing the swivel angle is much faster than reversing the complete transformer, this is the preferred method.

Fig. 182 shows the design of a hydraulic transformer using standard swashplate axial piston units. The design requires neither input nor output drive shafts as the drive shafts are connected together by means of a coupling. The tacho-unit is connected to the through shaft of the unit. If required, the through shaft of the fixed displacement unit can be used to drive a second actuator. An extra load can thus be powered in a simple manner from this point.

Normally, both axial piston swashplate units are selected to have the same displacement. However, as these are standard units, any required combination of units can be selected to suit the application, the operating pressure of the fixed displacement unit and the operating conditions. This means effectively, that the transformer can be operated either as a pressure transformer or as a flow transformer.

As no external mechanical connections to the transformer are required, the bent axis design of the unit can be simplified as may be seen in **Fig. 183**.

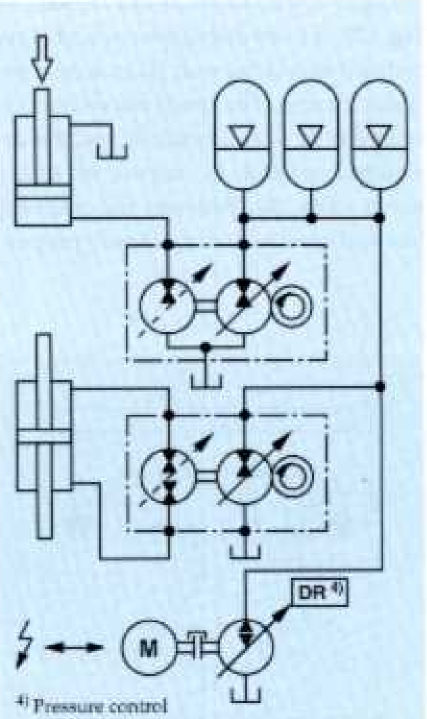


Fig. 181: Cylinder control with hydraulic transformer

The drive shafts of the two units may be connected to a common drive flange fixed to the housing and which also carries the gear drive to the tacho. As the pistons operate in opposite directions, bearing loads are substantially reduced. This also has a beneficial effect on the bearing life.

In addition, the axial dimensions and also the manufacturing costs are

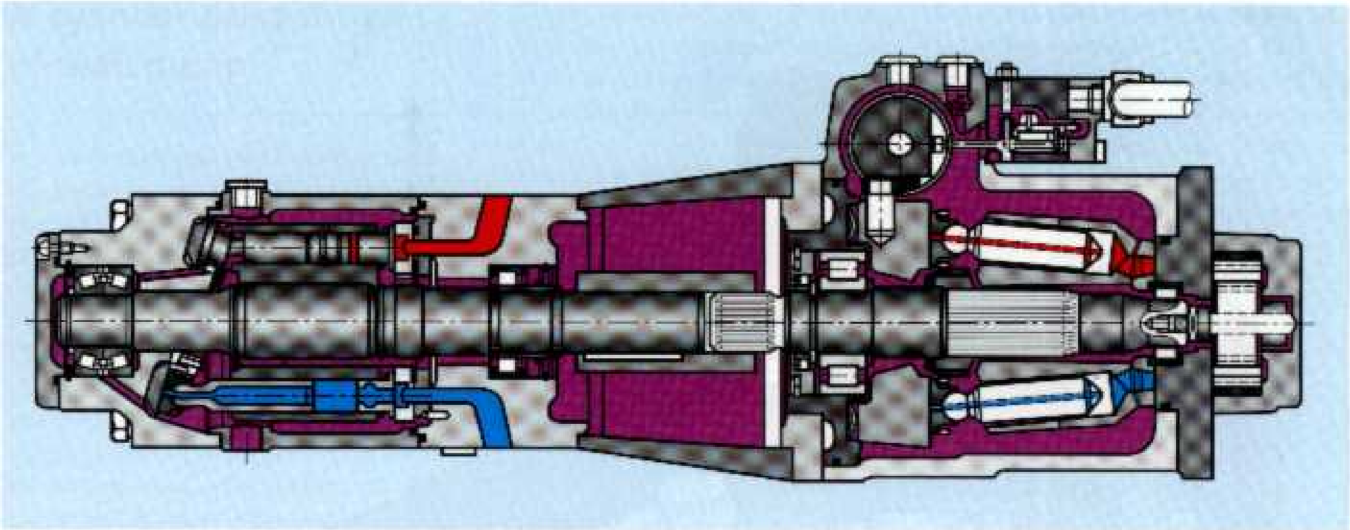


Fig. 182: Hydraulic transformer with swashplate units

considerably reduced. Any reduction in manufacturing costs must also have a considerable influence on the range of applications of the unit, making it more economic for lower powered applications.

As yet this transformer has not been constructed.

If, however, the application of a hydraulic transformer is not possible on economic grounds, the cheaper circuit shown in **Fig. 184** can be employed. Once more, the cylinders are operated from a system with impressed pressure onto which a number of units are connected. These units can operate either as generators or motors.

An internal combustion unit drives a pressure controlled axial piston unit with a mooring type control (mooring control means a two-quadrant drive system in one direction). A metering pump (D) is also driven by the engine to control the speed of the cylinders. The cylinders are single acting, the piston rod side being pressurised.

Only one cylinder at a time can be controlled by the metering pump, the piston speed being determined by the motor speed and the swivel angle of the metering unit.

If the pressure at the actuator side (V) is lower than the system pressure (H) when the cylinder is extending, the metering unit acts as a motor and drives the engine. If the pressure is higher, the metering unit acts as a pump, the extra power being taken from the engine.

When the cylinder is retracted, the process is reversed. Energy is recovered as the engine only needs to make up the difference in pressure between (V) and (H) on the diagram. The engine is also driven when the pressure (V) on the actuator side is higher than that on the system side with impressed pressure (H). If pressure on the actuator side (V) rises to the level of the maximum permissible value, a pressure switch operates the directional valves to unload the annulus side of the cylinders to tank. This has the effect of reducing the pressure by the ratio of cylinder areas. If more energy is returned to the system than the hydraulic accumulators can accommodate, the axial piston unit with pressure control and the mooring control feature then drives the engine with up to 30% of its power rating. Only after this time is it necessary to convert the extra energy into heat.

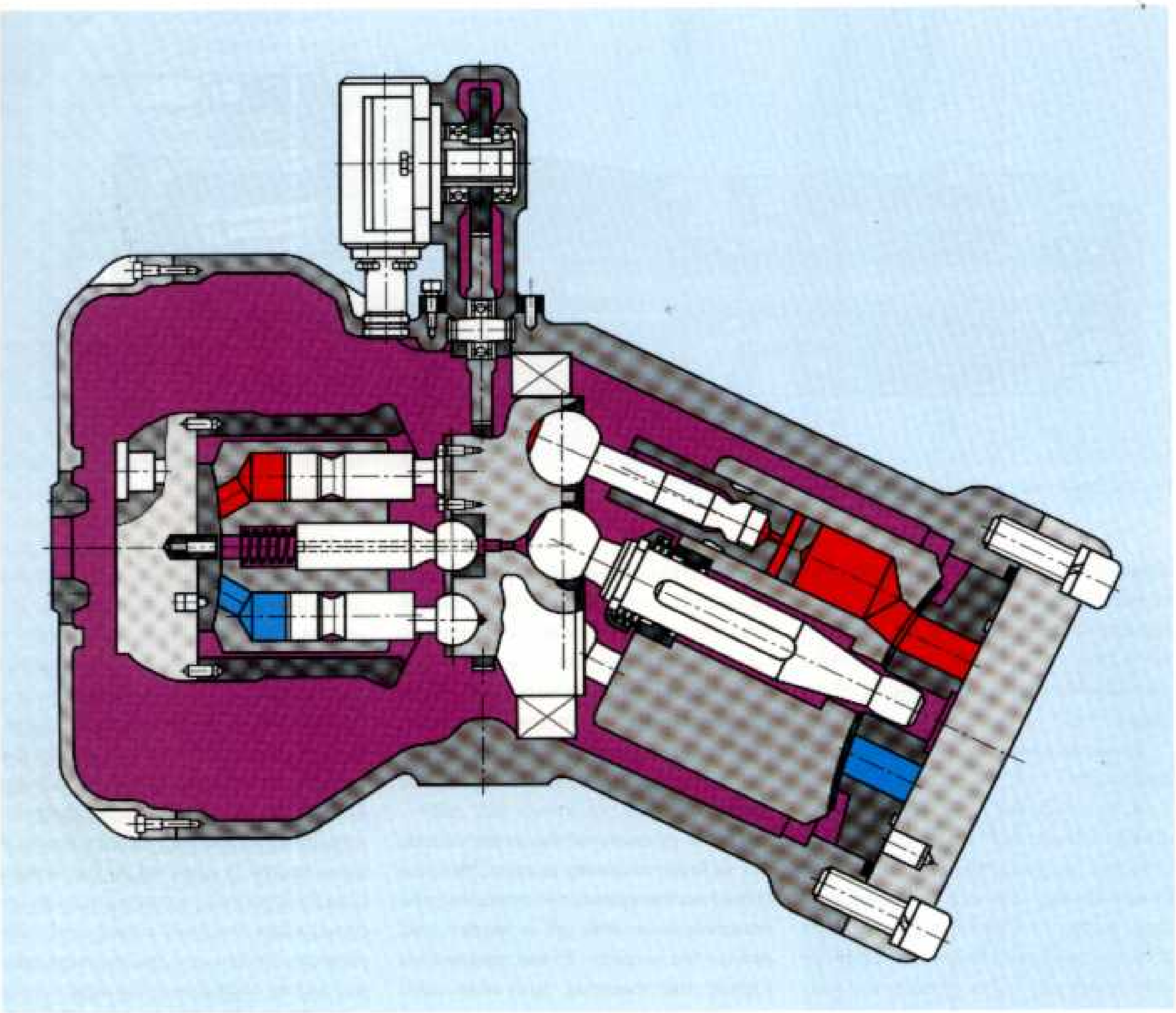
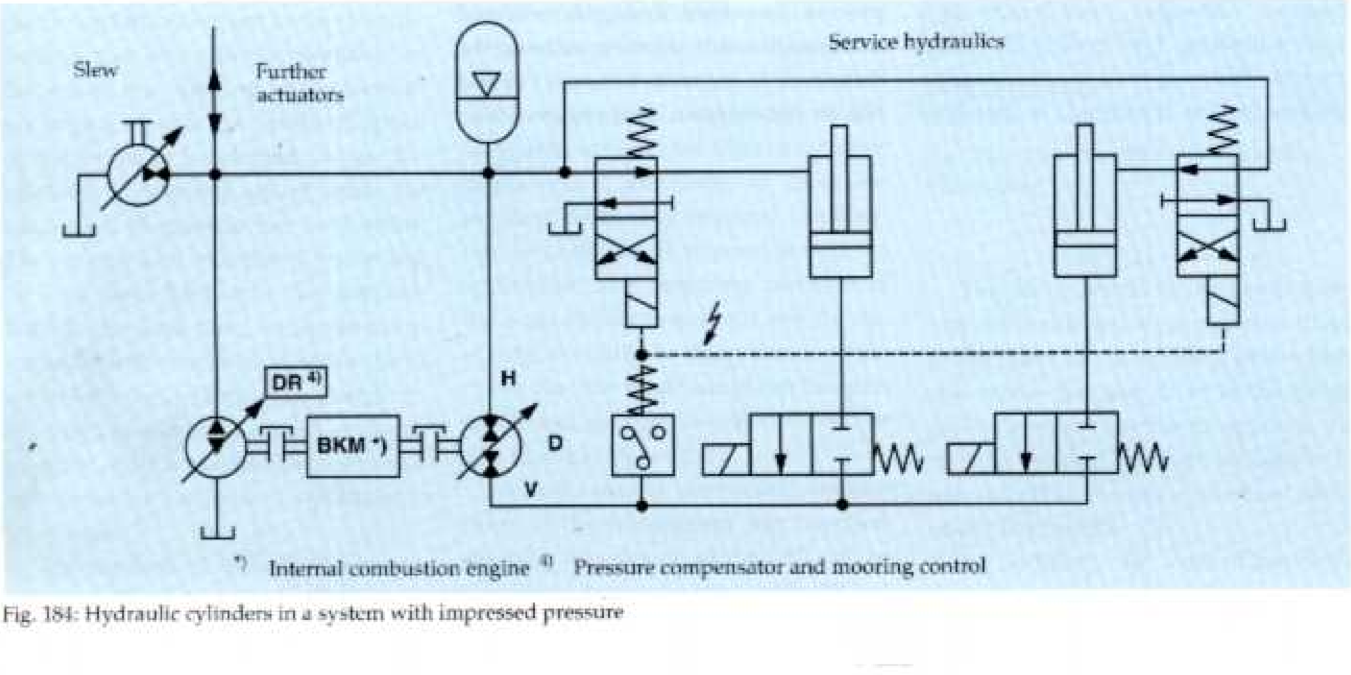


Fig. 183: Hydraulic transformer with bent axis units



*) Internal combustion engine 4) Pressure compensator and mooring control

Fig. 184: Hydraulic cylinders in a system with impressed pressure

A cylinder drive for an oil well pump

A well known feature of underground oil wells are the "nodding donkeys" (**Fig. 185**), so-called due to their regular nodding movements that are reminiscent of a donkey. These work hard pumping the oil from deep in the earth. The "nodding beams" of these machines, which are connected to the pump rods, are electrically driven via a pulley, a belt drive, a gearbox and a connecting rod. The weight of the pump rod is partially counterbalanced by a counterbalance weight. The velocity of the pump rod changes with the angle of the connecting rod in a sinusoidal manner throughout the cycle.

The need for greater flexibility in meeting the varying operating conditions cannot easily be achieved with this type of drive. In view of this, a direct hydraulic drive utilising a hydraulic cylinder was offered (**Fig. 186**). The hydraulic cylinder drives directly onto the pump rod and, with the exception of the short reversal time in the upper and bottom dead centres, remains constant throughout the stroke.

The conditions under which a crude oil pump must work are dependent upon many factors including the viscosity of the fluid to be pumped, the quantity of water and sand present, the condition of the bore hole and the gas pressure.

To this are added the problems of pumping very heavy oils. The pumping operation occurs when the pump rod is rising. On the down stroke, the pump chamber is filled.

If the down stroke occurs faster than the pump can fill properly, poor filling will occur. These problems are particularly acute in new boreholes where conditions are completely unknown. Flexibility is therefore paramount. Mechanical drives operate at the same speed in each direction - they have no choice. The maximum cycle rate therefore depends on the maximum downward speed. Under certain conditions this can be very slow. The hydraulic drive permits independent selection of very slow downward speed and the correct pumping speeds. This means that the lower cycle times can be achieved under otherwise identical conditions.

A further advantage is that the stroke of the hydraulic unit can be varied by simply resetting limit switches. This can be done to reduce production from the field without needing to stop the pump which would otherwise lead to the bore silting up.

When the drive was designed, the possibility of recovering the potential energy of the pump rod during the downward motion could not be overlooked. If all these requirements are

Fig. 186: Oil well pump with hydraulic cylinder drive

Fig. 185: Nodding donkey pumps

gathered together, it becomes apparent that a system with secondary control with energy recovery from a hydraulic transformer would provide the optimum solution (**Fig. 187**).

The single acting cylinder (1) is driven upwards by the hydraulic transformer (2) which also returns the potential energy to the accumulator (3) when the rod is travelling downwards. On the next upwards stroke, this energy is reused and any losses made up by the electrically driven, pressure compensated pump (4). The speed of the transformer and thus the speed of the cylinder is determined by the hydraulic or electrical tacho-unit (2.1). The cylinder is reversed by changing the direction of the hydraulic transformer or by moving the unit on the cylinder side over centre. The installed electrical power was 55 kW. This only covered the power required to pump the oil or approximately 50% of the power required during the pumping stroke.

The complete installation is shown in **Fig. 188**. The hydraulic transformer can be seen in the foreground. It consists of 2 tandem units - a total of four axial piston units type A4VSO125. To the right, on the through shaft of one of the units, the tacho-unit can be seen - a 6.5 cm^3 gear pump. The manifold with the controls for direction and speed are mounted on the bellhousing.

All components are easily accessible and understood and have fulfilled the requirements of :

- long service life working on a 24-hour basis,
- extreme reliability under adverse conditions and widely varying temperatures and
- easy operation and maintenance.

Another, somewhat modified solution for an oil well pump drive is shown in **Fig. 189**. Here the transformer principle is converted down to a certain extent, although recovery of the potential energy is completely retained. In this way the primary energy consumption is still low as with the previous example and remains virtually constant during a complete cycle.

The speed of the shaft, which consists of an electric motor (1) and the two

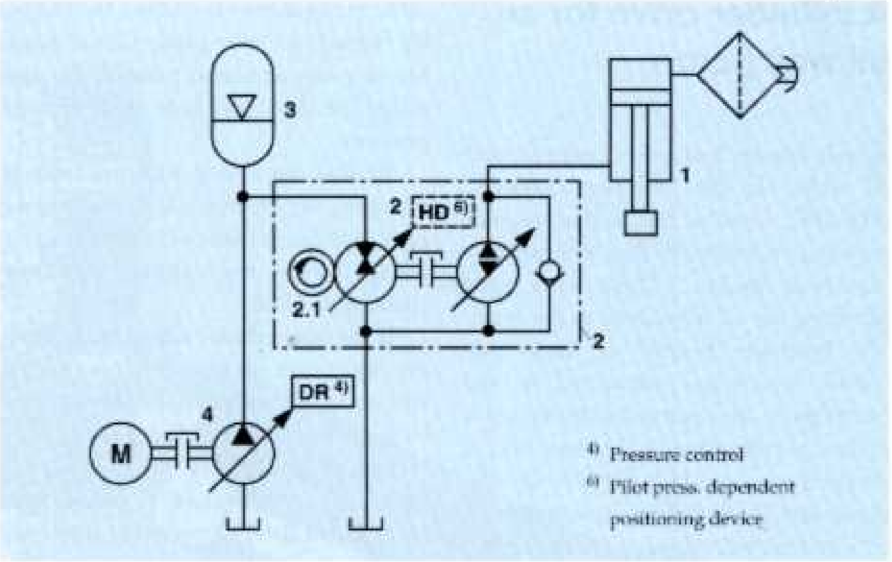


Fig. 187: Drive schema with hydraulic transformer

Fig. 188: Power unit for an oil well pump

tandem axial piston units (2) and (3), remains constant. Both units swivel over centre and can therefore operate alternately as motors or generators.

On the down stroke unit (2) operates as a hydraulic motor, drives pressure compensated axial piston unit (3), which operates as a pump. The potential energy from the rod is stored in accumulator (8) for reuse during the next pump stroke. With this stroke the tandem units reverse their operation. The upwards and downwards speed of the cylinder can be selected for optimum pump operation. For example, to achieve good filling a slow downwards stroke must be aimed at, whereas pump movement must take place quickly.

Changing direction is effected by means of a ramp and is therefore correspondingly gentle. No undesired peak loads will occur, nor will there be any mechanical overloading of the piston rod. This is a major advantage of the hydraulic drive over the mechanical drive. With the latter jamming may occur briefly during the downward stroke, caused by the penetration of sand for example. The drive does not recognise this and the result will be buckling of the cable with sudden tightening at the beginning of the stroke. This in turn can lead to damage of the piston rod and fatigue fractures.

This problem does not occur with a hydraulic drive, as any jamming interrupts cylinder movement. However, as axial piston unit (2) is still set to the down stroke, the flow that it requires will be sucked via valve (7). If the down stroke is not completed within a given time, indicated by limit switch (10b), a message will be sent to the PLC and pump operation will be stopped.

The advantages, referred to the service life of the piston rod, have to be considerable in the eyes of the manufacturer, as any exchange of faulty piston rods not only hinders production, but also means high repair costs.

All drive components as shown in **Fig. 189**, with the exception of the lifting cylinder, are accommodated in the drive unit (**Fig. 190**). The operating parameters such as the pre-setting of lifting and lowering speeds can be modified by means of a built-on console or from a central monitoring station. The electrical signals introduced into the system influence the swivel angle position and thus the displacement of the electrohydraulic positioning device of axial piston unit (2). Remote control permits optimum operating conditions to apply for the pumping operation, these depending on the degree of viscosity of the oil.

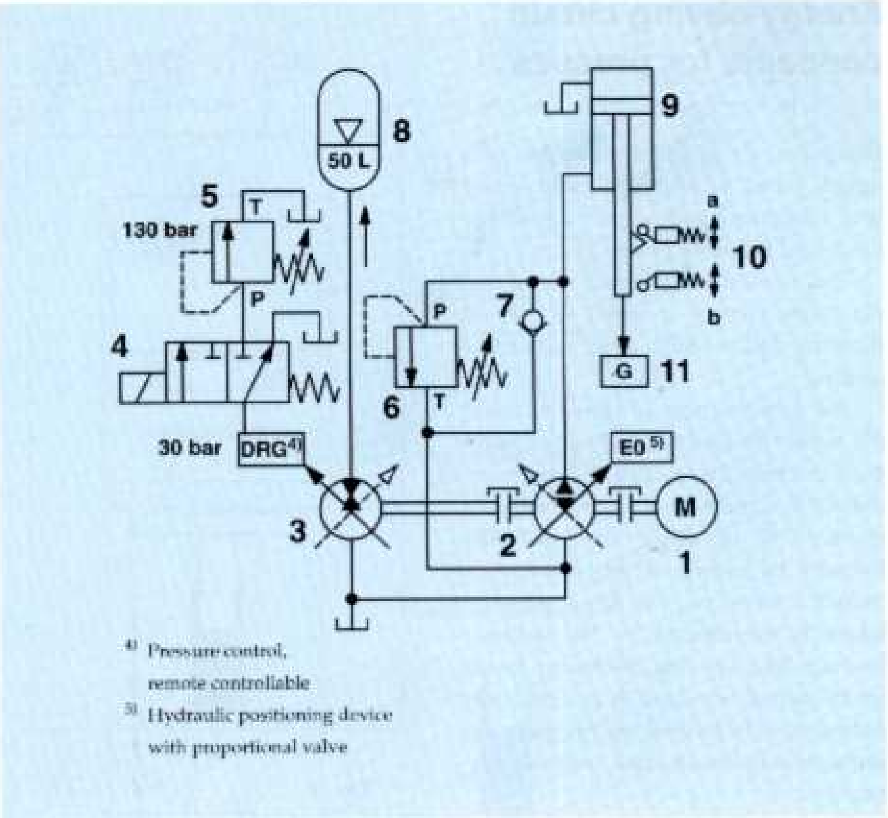


Fig. 189: Switch circuit of drive for an oil well pump

Fig. 190: Power unit with hydraulic accumulator for oil well pump

Energy-saving circuit concepts for presses

Developments in hydrostatics have, in recent years, to a considerable extent been concerned with improving energy usage and reducing power losses.

This has led to increased use of displacement control, in order to prevent throttling losses which occur with valve control.

The construction of presses could not apply this concept. Cylinder control used in conjunction with displacement control in a constant pressure system is effected by throttling the operating pressure by means of proportional or control valves on the load pressure which is determined by the cylinder loading. The ensuing throttling losses are converted into heat. It is technically not possible to recover energy from the actuator and feed it back into the system.

Throttling losses can be reduced by implementing displacement control and this will bring about not only a reduction in primary energy requirement and cooling capacity, but in certain cases a considerably lower pump capacity may be selected without its affecting the cycle time of the pump.

Energy recovery is of great importance as seen in the example of a deep drawing press.

In the usual circuit (**Fig. 191**) the hydraulically driven die cushion cylinder is brought into the starting position by means of a hydraulic variable displacement pump. The holding pressure during the working stroke can be pre-selected at the pressure relief valve. When the die cushion cylinder has attained the end position the hydraulic pump output will reduce to zero. After the press slide has been placed in position the down stroke will commence by displacement of the oil, and the pressure value set at the pressure relief valve will be requested. The energy produced will be converted into heat, the pump operating in idle running for this process.

Fig. 192 demonstrates how the energy balance can be improved.

The upwards movement of the pressure pad progresses in the way described via the pump flow at a pre-determined speed. On reaching the end position the positioning device of the hydraulic pump is switched over from flow control to the superordinate pressure control in two-quadrants (mooring operation). When over centre movement occurs in the same direction, the direction of flow and torque will change. After the press slide has been placed in position the hydraulic pump goes over to motor operation on starting the return stroke, and on maintaining the pre-set operating pressure forces the electric motor to act as a generator.

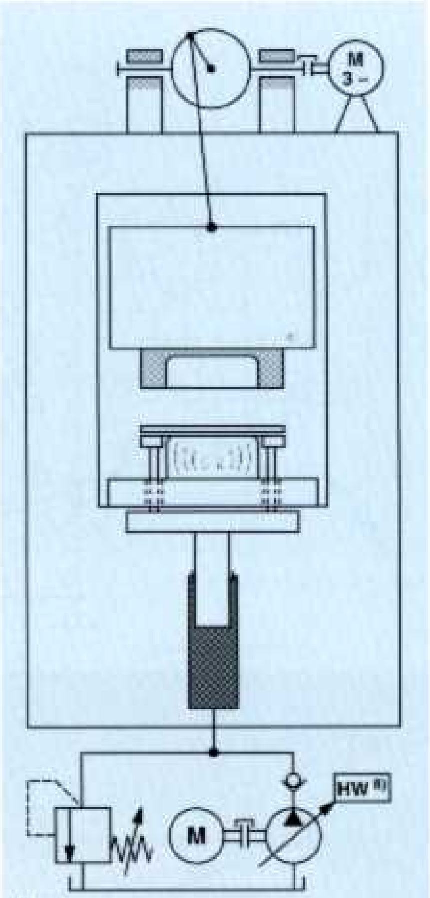
The energy produced is stored and fed back into the electrical power lines.

Another example of energy recovery with the same type of deep drawing press is shown in **Fig. 193**, in which the electric motor has been replaced by a secondary controlled hydrostatic unit.

Both slide and die cushion cylinder can be moved independently of each other. The two hydraulic pumps (2) and (3) swivel out in both directions and are therefore suitable for energy recovery.

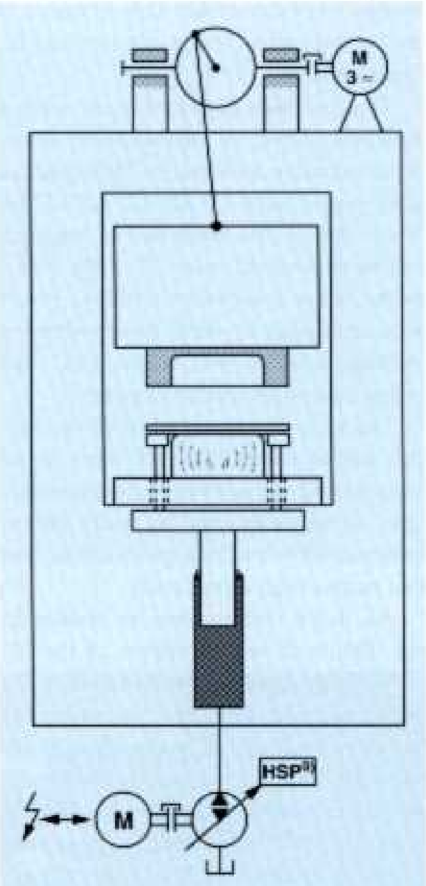
If the press slide on its downwards stroke does not come into contact with any solid material, the potential energy of the slide will be converted into electrical energy by the secondary unit (1) which is operating as a generator. When the press operation commences the secondary unit (1) acts a motor. The electric motor does not need to produce the total power, as the hydraulic energy of the die cushion cylinder is also fed mechanically into the secondary controlled circuit.

If the possibilities offered by secondary control are fully exploited a further simplification can be achieved on the primary side (**Fig. 194**). As the circuit for both press slide and die cushion cylinder is pressure controlled, both circuits can be directly joined, saving on one hydraulic pump. This does, however, restrict the press sequence. All



8) Hydraulic position dependent positioning device

Fig. 191: Conventional deep drawing press without energy recovery



9) Hydraulic servo adjustment with pressure control

Fig. 192: Conventional deep drawing press with energy recovery

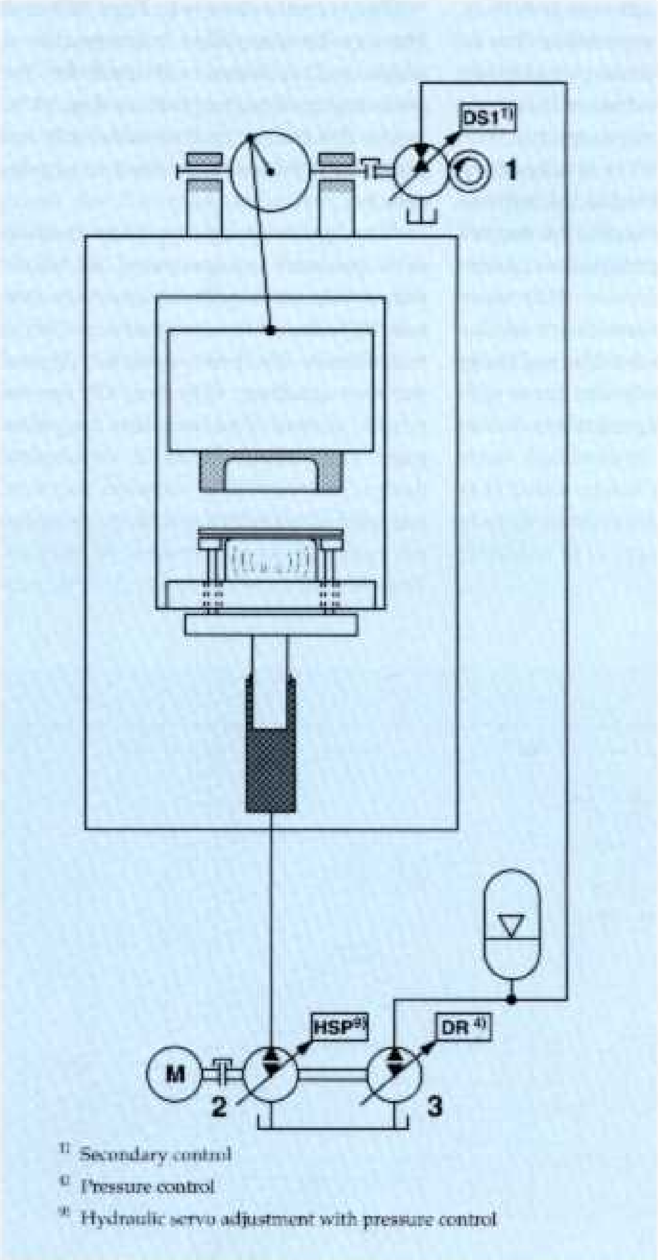


Fig. 193: Deep drawing press with secondary controlled slide

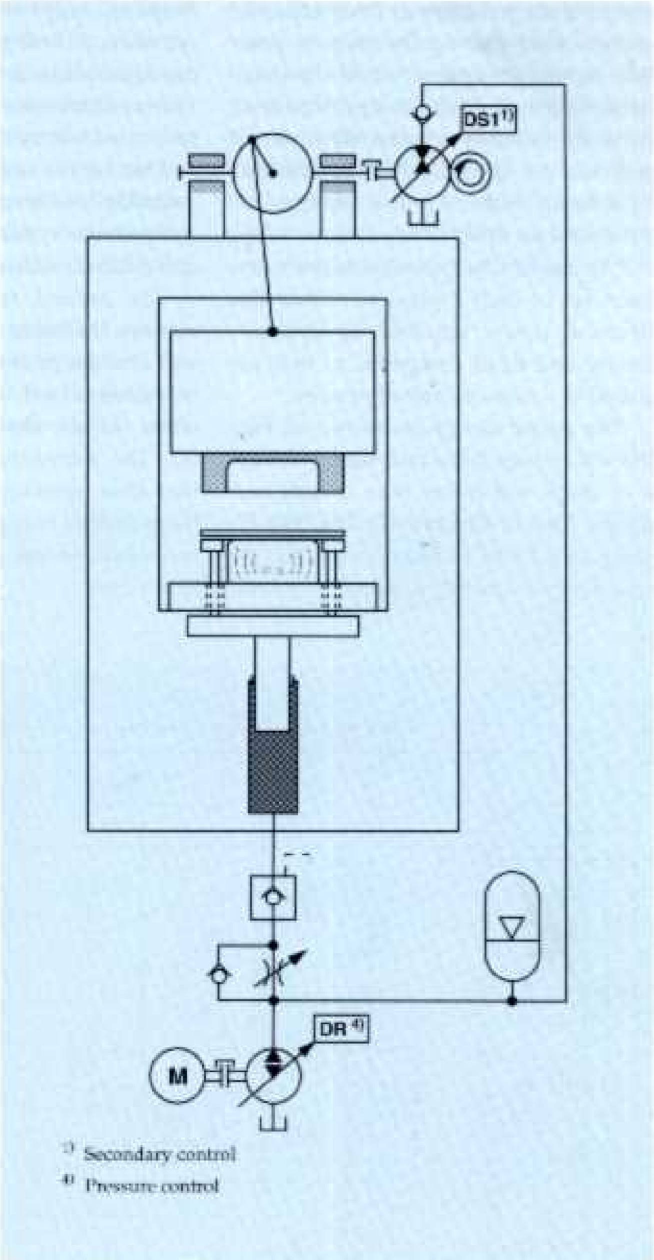


Fig. 194: Deep drawing press, secondary controlled

movements can still be carried out completely independently of each other, without any interaction.

The energy recovered from the die cushion cylinder is fed to the secondary unit without conversion. By eliminating the conversion the energy recovery level is improved still further. This circuit design leads to considerable savings on primary side capital costs. It does, however, make sense if the counter holding pressure does not need to fall below a specific value, as is the case in certain applications.

If the counter holding pressure is too low, the secondary unit must be of a larger design, the advantage on the primary side being compensated under certain conditions.

Several energy-saving circuit designs are described here, using hydraulic transformers in the manufacture of presses. **Fig. 195** shows the application of hydraulic transformers in a deep drawing press, whose slide (1) is moved by two or more double rod cylinders (2). Double rod cylinder (4) works as a rapid traverse cylinder.

The transformers can carry out an additional task. In addition to position or pressure control they can assume the synchronisation operation with multiple axis presses.

With this circuit variant the press cylinders (2) are pre-tensioned at both ends to a high pressure level. The press slide attains its force as a defined flow is drawn via the transformers. Due to the virtual constant pressure level displacement losses are to a large extent prevented. Another advantage is that the prevailing direction with rapid traverse takes place with short-circuited chambers. Pressure peaks such as with valve control do not occur.

Energy recovery is here once again of prime importance. During one section of the down stroke the potential energy of the press slide is converted by means of the transformer into hydraulic energy. After placing the slide in position on the workpiece, when the press stroke commences the energy produced at the die cushion cylinder (3) is recovered via the primary side mechanical equipment, reducing the energy requirement on the primary side.

Circuits of this type enable the manufacturer of such a press to reduce the electrical power required by approximately one third compared to that required by valve controlled presses.

The circuit design as shown in **Fig. 195** will change if the pull-back cylinder is of single rod rather than double rod design. This is illustrated in **Fig. 196**. In order to achieve a closed circuit for all cylinder movements, a direct hydraulic coupling with two single rod pull-back cylinders (5) will not be possible, due to the equal areas of the press cylinder (4). The rapid traverse speed must therefore be carried out by the transformer (13).

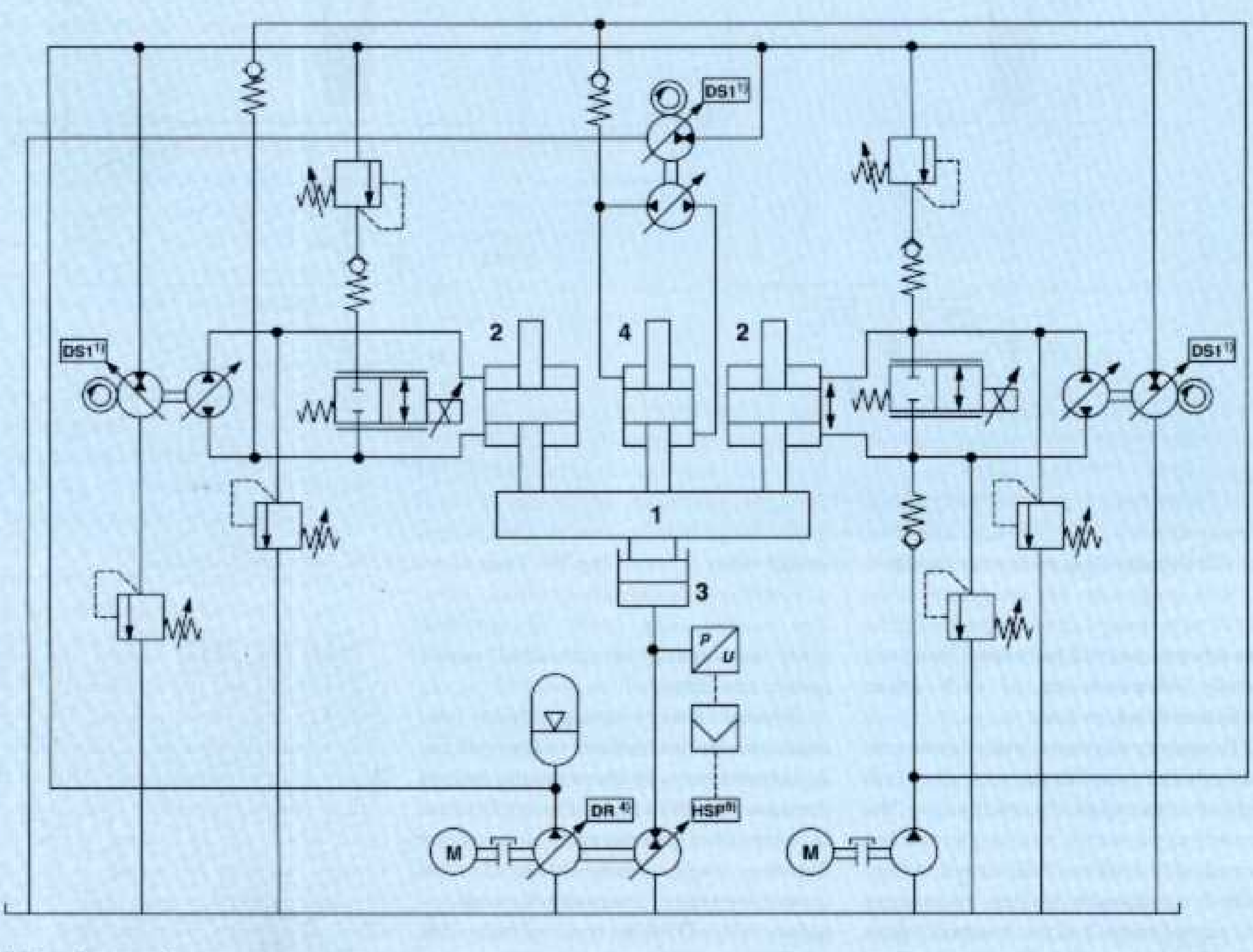
One of the units (13.1) is secondary controlled and is connected to the impressed pressure system, whereas the second unit (13.2) controls the pull-back cylinder.

The second transformer (13) now controls the rapid traverse down stroke and also the return stroke. During these movements both areas of the press cylinder (4) are short-circuited via valve (6). The transformer is installed such that after opening the safety valve (14) the potential energy of the press slide is recovered, as the unit (13.2) is working as a motor.

The circuits shown in **Figs. 195** and **196** can be simplified considerably if single rod cylinders are used for the press and pull-back cylinders (**Fig. 197**).

As this circuit is of considerable importance it is now described in greater detail.

The hydraulic drive system consists of a pressure compensated hydraulic pump (1), an impressed pressure system because of the accumulator (10), a transformer (3), press cylinder (4) and the two auxiliary cylinders (5) for the return stroke. Two variable displacement axial piston units in swashplate design, mechanically coupled together, are used as transformers. Both units can be operated as generators or motors. The first hydrostatic unit (3.1) is con-



1) Secondary control
4) Pressure control
6) Hydraulic servo adjustment with pressure control

Fig. 195: Hydraulic press with transformer and double rod cylinder

nected to the impressed operating pressure. It operates under secondary control, whereas the second (3.2) controls the cylinders. This circuit is designed so that the transformer only has to draw energy from the main supply that is required for the press cylinders i.e. the hydraulic and potential energy produced by the press and fed back into the pressure system. For this reason no throttling elements are installed in the energy transmission lines, all control valves being 2-way cartridge valves with open and close functions.

The complete press cycle of rapid traverse, the press operation, decompression, rapid return travel and flushing of the operating medium, is carried out by a single transformer. The speed of the press cylinder is determined by the variable speed of the transformer and the variable displacement which also determines the direction of movement.

The two auxiliary cylinders are required in closed circuit operation during rapid traverse and the press operation to effect movement of the press ram. These are designed so that the annulus area of these auxiliary cylinders, together with the annulus area of the press cylinder amounts to approximately the same value as the piston area of the press cylinder. A slight area overlap of the annulus side is actually desirable under certain conditions, in order to compensate for external leakage of the unit (3.2).

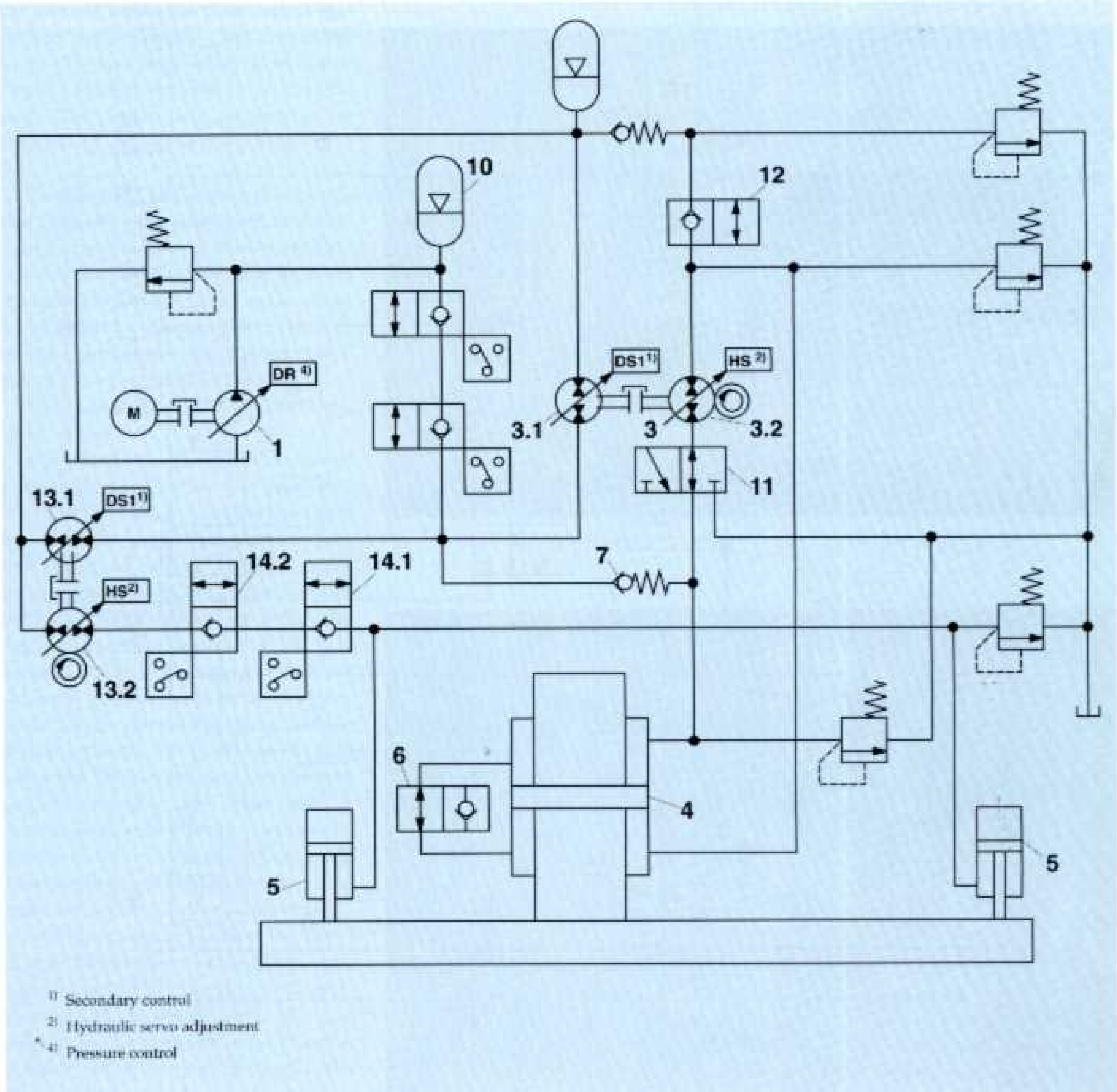


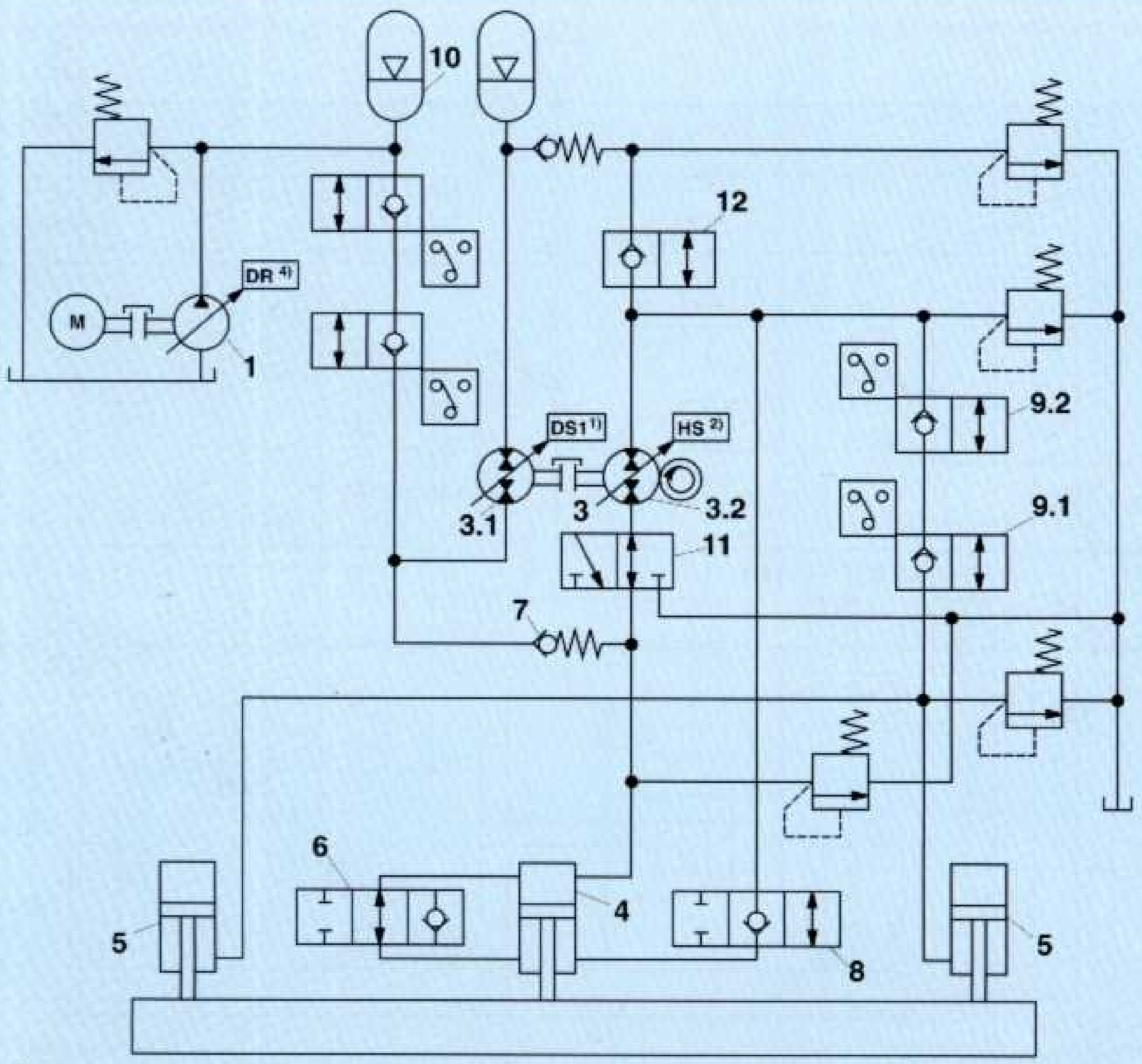
Fig. 196: Hydraulic press with transformer and single rod cylinder

Function of the press cycle

When setting up the press for rapid traverse the short-circuit valve (6) is switched to through travel and the annulus area and piston area of the press cylinder are connected to each other. The operating pressure that is generated by the pressure compensated hydraulic pump is fed via the check valve (7) as pre-tensioning pressure to both cylinder areas. This pressure is then directed to the annulus areas of the auxiliary cylinder via 2-way cartridge valve (8) and the pilot operated safety valves (9.1) and (9.2).

With the equal areas of the cylinder group and stationary transformer or with the unit (3.2) rotating at zero stroke the press ram will be in the rest position.

If the transformer is now activated and the hydrostatic unit (3.2), which is operating as a pump, is swivelled out, flow will be directed from the auxiliary cylinders via the hydraulic pump to the press cylinder, which will be moving rapidly downwards. The fluid will flow via short-circuit valve (6) from the annulus side to the piston side of the press cylinder. On commencement of the press operation the required speed will be set. The external loading will effect a pressure difference on the cylinder areas, and the short-circuit valve (6) will automatically close. The short-circuit valve has here the function of a check valve. The transformer unit (3.2) now controls the flow of all annulus areas to the piston area of the press cylinder and the operating pressure in the cylinder chamber will rise.



1) Secondary control
4) Pressure control
2) Hydraulic servo adjustment with pressure control

Fig. 197: Deep drawing press with transformer and retaining cylinder

After the pressing operation is completed the fluid will be decompressed by means of the transformer unit (3.2) which now drives the generator unit (3.1) as a motor.

The energy recovered is stored in the hydraulic accumulator (10) for later use. The same occurs if the potential energy from the press ram during the down stroke permits the unit (3.2) to operate as a motor.

Apart from flow losses, as only a small quantity of fluid will flow from the impressed pressure to the cylinder circuit, the temperature of the operating medium must be controlled by a flushing process, when the press ram is stationary for example. The reversing valve (11) will thus effect a temperature-dependent metered flow to the tank. This volume will be replaced by fresh oil by means of the check valve (7).

This flushing process can also be carried out during the press operation if required, for example if during decompression the medium flowing back via the unit (3.2) is directed not to the annulus area, but by means of valve (12) into the low pressure circuit of maximum 5 bar, and from there to the tank.

Here again backfeed is automatic via check valve (7).

Fig. 198 shows the operating side of a deep drawing press that has been modified by installing hydraulic transformers. The press ram is moved by four single rod cylinders situated in the corner areas. As the ram has to carry out large vertical strokes, the rapid closing movement is carried out by dead weight. The excess energy produced is fed back into the pressure system. Another feature is that the press table is driven by three die cushion cylinders. All seven cylinders are controlled by transformers (**Fig. 199**) and are connected to a common pressure system by means of secondary controlled units. Two pressure compensated axial piston units are installed on the primary side.

During stroke movement the speed of the transformers remains constant and movement is controlled exclusively by means of the swivel angle. The fluid taken from the lower cylinder chamber is fed to the upper end of the cylinder and complemented by a corresponding flow from the common pressure system. Downwards movement, which will no longer be parallel due to unequal forces placed on the ram during the press operation, is recognised at the cylinders and compensated for via the measurement systems.

The pressure decrease in the lower cylinder chambers permits an operating pressure system with a high pressure level, offering economic energy recovery. In this way power peaks are easily catered for. As the energy in the lower cylinder chambers can be recovered, the installed primary power can be considerably reduced from that required in conventional valve technology. The ram is maintained in a parallel position without the need for separate counterbalance holding cylinders or any other mechanical components, and there is no ram force lost.

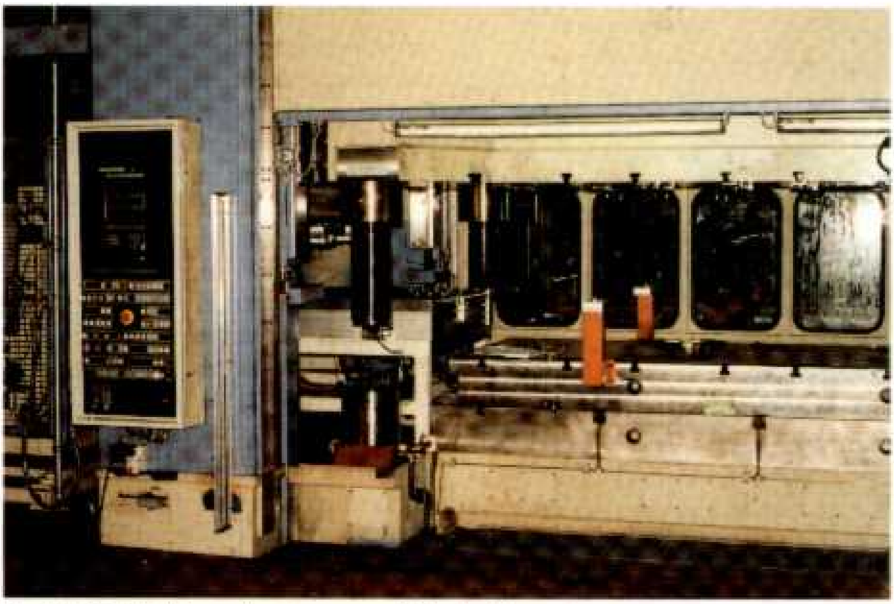


Fig. 198: Modified deep drawing press with hydraulic transformers

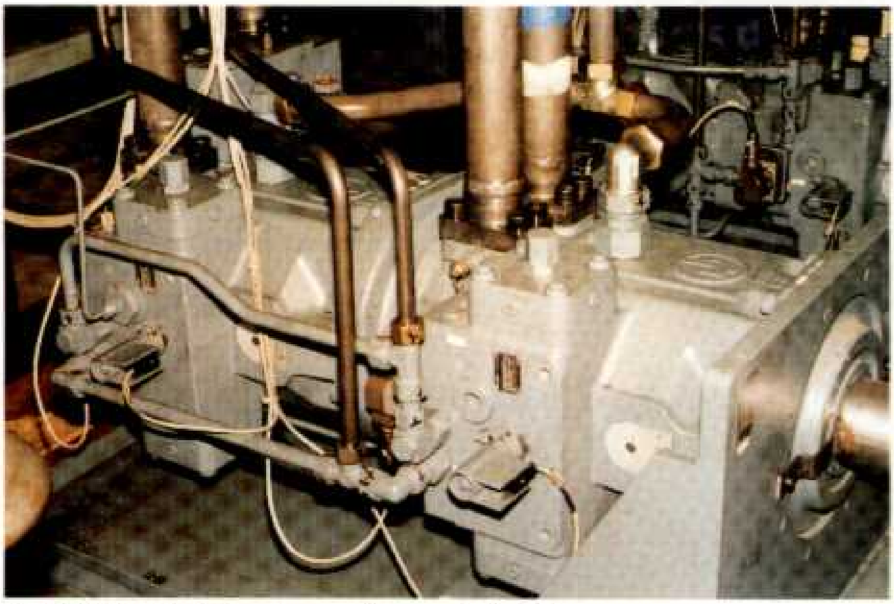


Fig. 199: The hydraulic transformer of the press to Fig. 198

Summary

In this chapter various possibilities for energy-saving control of hydraulic cylinders in a hydrostatic system with impressed pressure have been introduced. The hydraulic transformer plays a significant part, as it permits both cylinder control without the usual throttling losses, and also enables energy to be recovered from the hydraulic cylinder and fed back into the pressure system. The various diagrams show transformer circuits for different cylinder types such as those with a single rod, plunger or double rod.

The Speed-Variable Electrical Drive

Introduction

In electrotechnology D.C. and three-phase A.C. motors are used for speed-variable drives.

Electrical energy is normally generated in the form of a three-phase A.C. current. The direct generation of a D.C. current is limited to special cases. In Germany, however, around 25% of energy generated is required for D.C. actuators, therefore provision must be made in the supply network to convert the available A.C. current into a D.C. current.

Even a speed-variable A.C. motor cannot manage in the intermediate circuit without a D.C. current.

Considerable technological advances have been made in recent years with respect to the conversion of D.C. current into A.C. current. Development has progressed from the rotary transformer via the static mercury vapour converter with electronic control through to the semiconductor rectifier. The latter is superior to all other types of transformer. The technical possibilities of its development have as yet been nowhere near exhausted.

Although the static-converter fed three-phase A.C. motor has become much more common, the D.C. motor has still retained its importance due to its good characteristics. It can be:

- dynamically loaded up to three times the nominal torque,
- reversed for short periods,
- speed controlled in the ratio of 1 : 1000 and it also has
- torque uniformity with a 2% irregularity at approx. 10 rpm.

However, because of the increasing importance of speed-variable three-phase motors, those versions which may be used in all forms will be described here:

- three-phase asynchronous motor with cage armature,
- three-phase asynchronous motor with slip ring,
- continuously excited synchronous motor,
- rotating rectifier-excited synchronous motor or
- reluctance motor.

The speed characteristics of an asynchronous motor can be described as follows:

$$n = \frac{f_1}{p} \bullet (1 - s)$$

The speed can be varied by means of the following values:

- rotating speed n of the rotary field with the output value stator frequency f_1 and
- slip s between rotary field and rotor with the output values stator flux ϕ, rotor voltage U_2 and torque M with asynchronous motors.

At constant stator frequency f_1 and constant number of pairs of poles p the asynchronous motor can only be controlled via slip s. Special measures are necessary for this in the rotor circuit such as the switching in of resistors or the generation of a counter voltage. Leakage-free control is, however, only possible via a change in frequency of the stator voltage.

In a similar way the power of the three-phase motor is detrimentally influenced by the slip and the rotor resistance R_2. Here too the stator frequency and the rotor standstill voltage U_2 must be variable:

$$M - \frac{U_2}{R_2} \bullet s - I_A \Phi.$$

where:

n = speed in rpm
f_1 = frequency in 1/sec
p = number of pairs of poles
M = torque in Nm
s = slip
I_A = rotor current in A
ϕ = stator flux $\Phi - \frac{U_1}{f_1}$ in Vs
Index 1 = stator size
Index 2 = rotor size

The characteristics achievable with the asynchronous motor are comparable with those of the D.C. motor.

As can be seen from the torque equation, the torque progression can only be influenced by voltage.

The torque is formed from the product of stator flux ϕ and rotor current I_A. The stator flux is proportional to the voltage and period, and the rotor current is also influenced by the slip. In order that the torque be sustained via the frequency during speed control, the maximum value of the magnetic flux must also be sustained, in other words the quotient U_1/f_1 must be kept constant. The static frequency converter is used for this. This is connected to a three-phase mains supply and the variable values are fed to the rotary field motor via a D.C. intermediate circuit (**Fig. 200**).

Basically the function of a frequency converter is to convert a fixed three-phase mains supply into a variable D.C. voltage intermediate circuit, which is

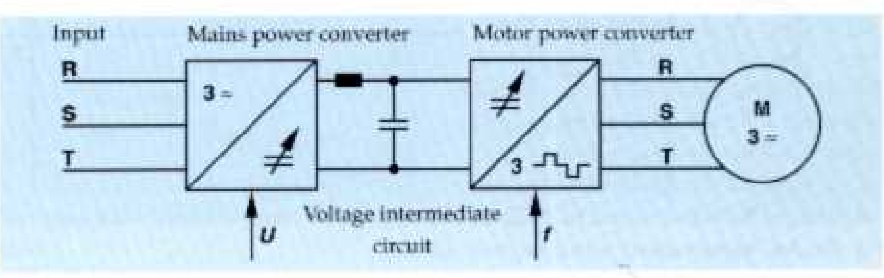


Fig. 200: Basic circuit diagram of a power converter

then reconverted into a three-phase current, but this time with constantly variable frequency or speed. The three-phase mains supply is de-coupled from the motor circuit via an intermediate circuit.

The intermediate circuit frequency converter can be designed either with impressed D.C. voltage or with impressed D.C. current (**Fig. 201**).

The term "impressed" for flow and operating pressure, which is also used in hydrostatics, is borrowed from electrotechnology.

An important feature of the frequency converter with an intermediate circuit is the inductor used for de-coupling the power converter from circuit and motor, which occurs not without losses. The inductor must also compensate for the power factor of the motor.

Frequency converters with voltage intermediate circuit are also characterised by capacitors in the intermediate circuit. In hydraulics an accumulator would be installed here.

The motor power converter distributes the impressed D.C. current or D.C. voltage in the cycle of the required frequency between the three motor lines.

In general an A.C. current is an interlinked three-phase current. With symmetrical loading, voltage and current can be shifted in opposition to each other by 120° with respect to time, the sum of the instantaneous values of voltage and current being equal to zero. The three phases are interlinked in star or star-delta connection, with or without a centre conductor.

In a D.C. circuit the power output from the current source is the product

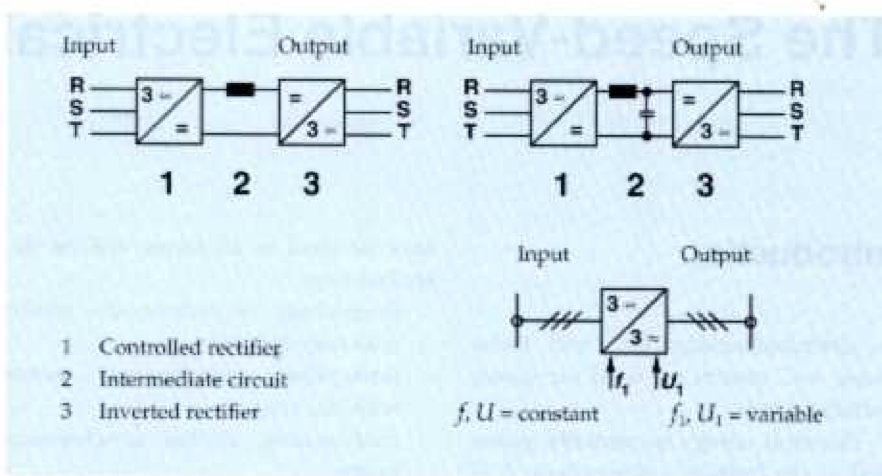


Fig. 201: Power converter for three-phase motors with impressed current (left) and impressed voltage (right)

of the current I flowing through the actuator and voltage U.

In an A.C. circuit, on the other hand, the power is dependent upon the type of actuators connected and their impedances, which result in phase shifting between voltage and current.

The angle of phase difference is called φ, cos φ being the power factor.

With a pure ohmic loading voltage and current oscillate in phase i.e. $\varphi = 0°$, $\cos\varphi = 1$ (**Fig. 202**).

With an A.C. circuit with
inductive ($\varphi = 90°$; $\cos\varphi = 0°$),
capacitive ($\varphi = -90°$; $\cos\varphi = 0°$)
and
mixed ($-90° \leq \varphi \leq 90°$; $\cos\varphi = \leq 1$)
loading is thus idle power P_q plus the active power P.

The quotient of active power and apparent power P_s represents the power factor cos φ:

$$\cos\varphi = \frac{P}{P_s}.$$

Here φ describes the angle of phase difference between voltage and current.

The generators endeavour to produce the maximum possible power factor to the actuators with appropriate support. As the generators are designed for a specific current, the active power will be reduced as the idle power increases. This causes losses in generators, transformers and the mains circuit, which in turn leads to higher system costs.

In order to maintain a high power factor, the motors should be driven with nominal loading where possible. Running motors continually with underload should be avoided if possible. With asynchronous motors the power factor is also influenced by the number of poles. Low speed motors require greater magnetisation and therefore have a lower cos φ than high speed motors.

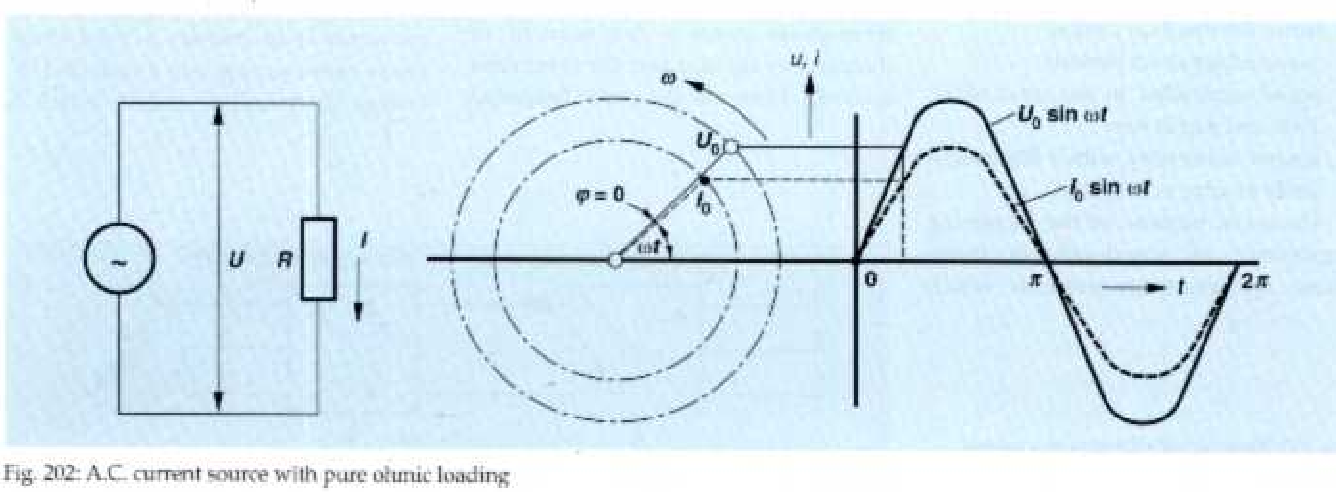


Fig. 202: A.C. current source with pure ohmic loading

In order to be able to operate a three-phase variable drive on a three-phase circuit, a static frequency converter must be used, which can convert the fixed frequency f_0 into the variable frequency f_1 that is required by the motor, as well as the associated closed loop control devices.

The power section of this frequency converter consists mainly of power semiconductors such as

- diodes,
- thyristors,
- GTOs (Gate turn-off thyristors),
- transistors,
- capacitors and
- inductors.

The semiconductors convert either the three-phase A.C. voltage into a D.C. voltage (rectifier operation) or the D.C. voltage into an A.C. voltage (inverted rectifier operation).

The closed loop control device generates peak pulses for the thyristors and influences the voltage build-up in the power section, so that the required speed is set for the motor.

The power diode

The function of a diode is comparable with that of a check valve. It permits flow of the current in one direction only from anode (A) to cathode (C) (**Fig. 203**).

The conversion of an A.C. voltage into a D.C. voltage can be seen in an example with high inductivity in the load circuit (**Fig. 204**). The valve which directs the current will always be the one of valves 1, 2 or 3, whose anode possesses the higher voltage opposed to the cathode. The valves separate themselves in the points of intersection of their phase voltages in the current-carrying wire. The D.C. voltage consists of the peaks of the sinusoidal valve voltages (**Fig. 205**). Voltage U_{di} represents the mean value during idle running, the size of the D.C. voltage for a given circuit being directly proportional to the adjacent A.C. voltage.

If the D.C. current is evened out by means of inductor coils a current block with a flow duration of 120° respectively will flow in the valves.

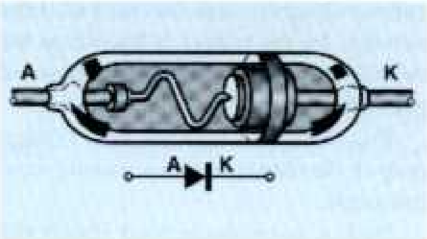


Fig. 203: Screw diode

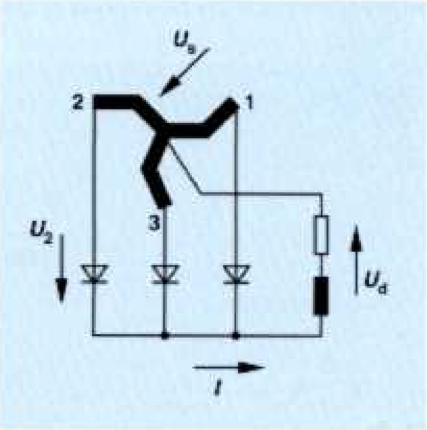


Fig. 204: Circuit diagram for rectification via three-pulse midpoint circuit

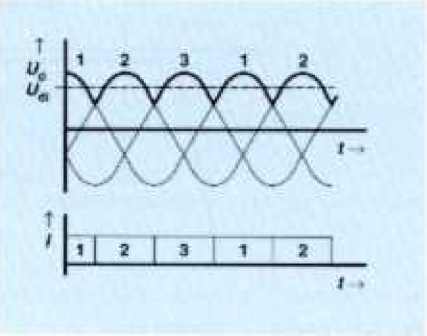


Fig. 205: Rectification to Fig. 204

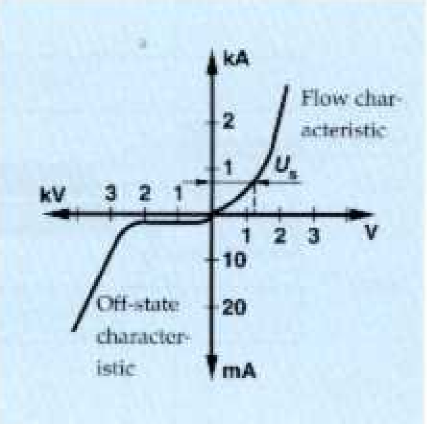


Fig. 206: Characteristic of semiconductor diode

Fig. 206 shows the characteristic of a semiconductor diode. The different current and voltage guidelines should be noted in direction of flow and the direction blocked. The on-state current is also small for low on-state voltages and increases sharply above the threshold voltage.

In the blocked direction only a small off-state current will flow due to the self-power of the crystals against the on-state current, even though the voltage is high. Silicon is used predominantly as the basic material for rectifier diodes, although germanium is sometimes used.

The thyristor

Voltage control for speed-variable drives is achieved using controllable components such as the thyristor. The thyristor blocks flow initially in both directions, only releasing flow in one direction after a control pulse. The conductive state is released when the flow is smaller than the holding current.

The thyristor is comparable with an electrically piloted check valve in hydraulics.

The physical action of a thyristor can be demonstrated by means of a simple model.

Fig. 207 shows the schema of a silicon monocrystal arranged in four different layers, so that three P-N junctions (1,2,3) are produced (P = positive, N = negative). With a negative anode-cathode voltage $U < 0$ the diode blocks P_1 - N_1 (1). This state corresponds to the off-state characteristic in **Fig. 208**.

With a positive voltage $U > 0$ and no pilot current the diode blocks N_1 - P_1 (2). We now consider the blocking characteristic in **Fig. 208**. With a positive voltage $U > 0$ and sufficiently high pilot current I_{pilot} the blocking at junction 2 at N_1 - P_2 will be lifted. The thyristor takes on a low flow resistance through current I now flowing, it ignites and behaves like a diode. This state corresponds to the flow characteristic. It will remain intact even if the pilot current is cut off, if the load current exceeds a holding value of I_H. If load current I in the load circuit is reduced to zero the thyristor will revert to the off-state blocking condition.

The thyristor characteristic shown in **Fig. 208** is for reasons of clarity not to scale.

For a load current of several 100 amps the off-state current in both directions will be 10 mA. The voltage drop in the ignited state is 1.5 volts, whereas the maximum off-state voltage in both directions can be 1 to 3 kV.

When a maximum negative off-state voltage is exceeded the off-state current will increase to such an extent that thermal destruction will occur. With a positive voltage the thyristor will switch if the so-called sweep voltage is attained with increasing pilot current and decreasing voltage. It will be conducting and will continue to be so if the pilot current is cut off, as long as the load current is above the holding value I_H.

As they are only small and have a low heat capacity highly-loaded thyristors are subject to many limiting conditions with regard to temperature, voltage and current.

The small surface area needs to be increased by means of dissipaters in order to remove heat losses (**Fig. 209**). In addition to air cooling, with large load currents oil or water cooling is usual. These can be mounted on both sides, as demonstrated in **Fig. 209**. In this way a power pack with a 30 mm diameter silicon cell is capable of controlling 500 amps by air cooling and up to 700 amps with water cooling.

The time between the natural turn-on time - the point of intersection of the valve voltages - and the turn-on time initiated by the trigger is known as the delayed turn-on time and is measured in electrical degrees.

Fig. 210 shows how the D.C. voltage output decreases with increasing control angle.

With a pure ohmic load circuit the D.C. voltage will be zero for a control angle $\alpha = 150°$.

It is a known fact that a reactive coil with high inductivity in the load circuit acts as an energy accumulator. Such a coil prevents the current from being reduced to zero when it goes over centre. The voltage built up at the coil ensures that the flow from the valve continues over centre.

As can be seen in **Fig. 211**, for a control angle of $\alpha > 90°$ the D.C. voltage will be zero.

The curve progression for an inductive load at $\alpha = 90°$ refers to an inverted rectifier i.e. the energy flows from the

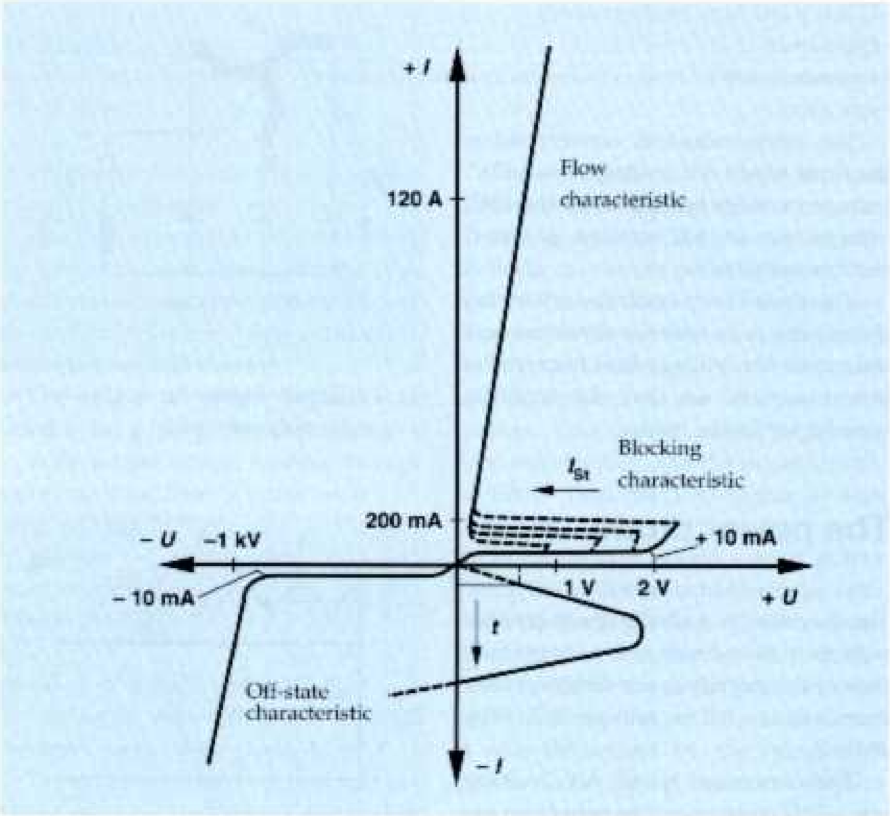


Fig. 208: Thyristor characteristic

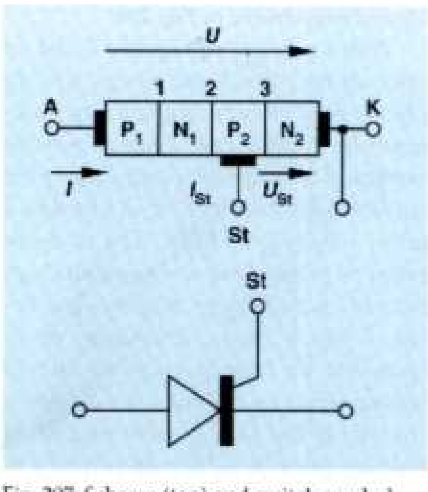


Fig. 207: Schema (top) and switch symbol (bottom) of a thyristor

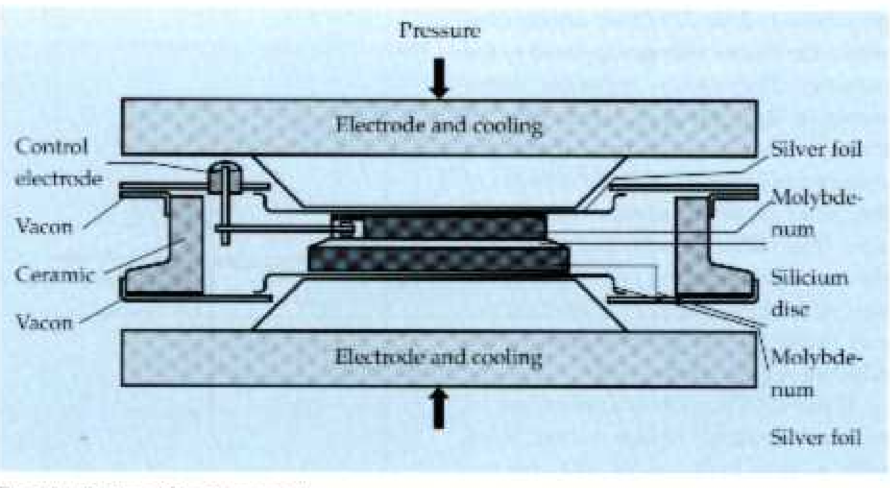


Fig. 209: Design of a power pack

coil via the power converter back into the mains circuit. The reversibility of the energy flow with inductive and ohmic inductive load is clearly visible from the control characteristic. Before the static converter gave the controlled electrical drive a new boost by means of the thyristor, we had to rely on the rotary converter with its double energy conversion. The constant mains voltage had to first be converted into a mechanical speed, which then had to be converted back into an electrical form with a variable D.C. voltage. Although this double energy conversion offered advantages in that there were separate current circuits and de-coupling via energy accumulators, there were also the associated disadvantages such as expensive machinery and foundations, poor dynamic response and high maintenance costs.

The so-called Leonard circuit functions as follows (**Fig. 212**).

Three-phase motor 1 drives at constant speed a separately excited pilot frequency generator 2, which outputs its voltage directly to the main motor 3, also separately excited.

The armature of pilot frequency generator 2 remains connected continuously to the armature of main motor 3. The electrical energy will flow through this circuit, the main current circuit, from the pilot frequency generator to the main motor, or, during deceleration, in the opposite direction.

As there is normally no available D.C. voltage, three-phase motor 1 must still drive exciter generator 4.

Speed control of the main motor can be effected by several methods:

- by changing the armature voltage of the pilot frequency generator (Leonard rule),
- by changing the excitation of the pilot frequency generator (field control) or
- by combining both these control methods.

In other words, for speed control of the main motor, only the energising fields of the pilot frequency generator or of the main motor need to be controlled, the current being monitored for torque control.

Disadvantages are the high system costs, a minimum of three motors of similar power being required, and the relatively low overall efficiency of 65 to 70 %

$$\eta_{total} = \eta_1 \cdot \eta_2 \cdot \eta_3 \cdot \eta_4.$$

It is only in recent years that the static converter has brought electrical drive systems to the forefront again with the thyristor, and it will continue to influence development in this field.

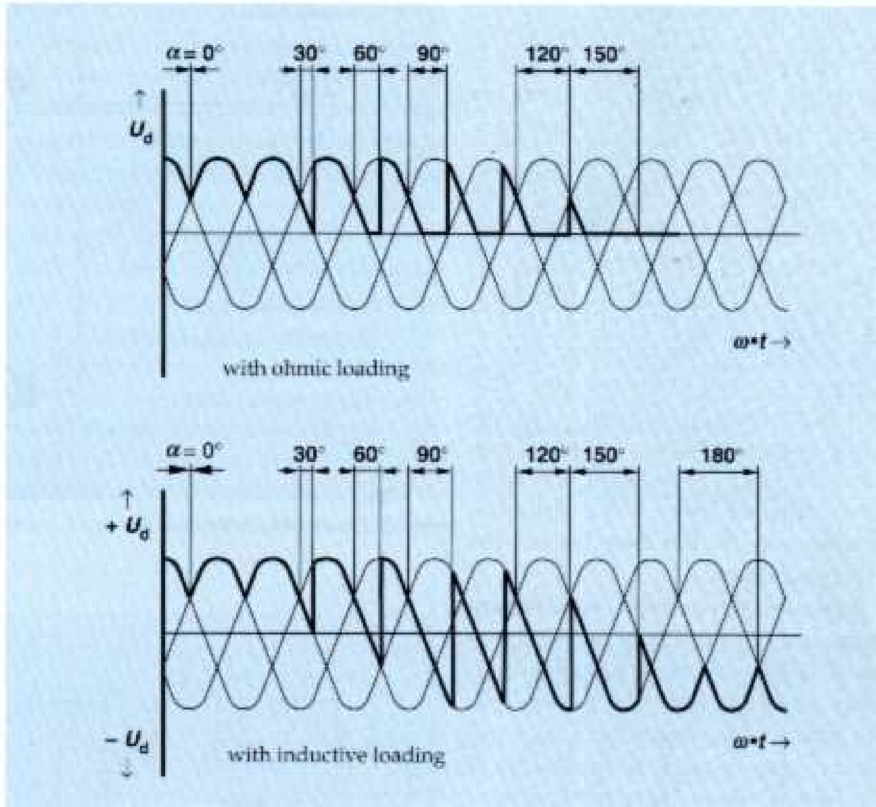


Fig. 210: Influence of control angle on the output voltage

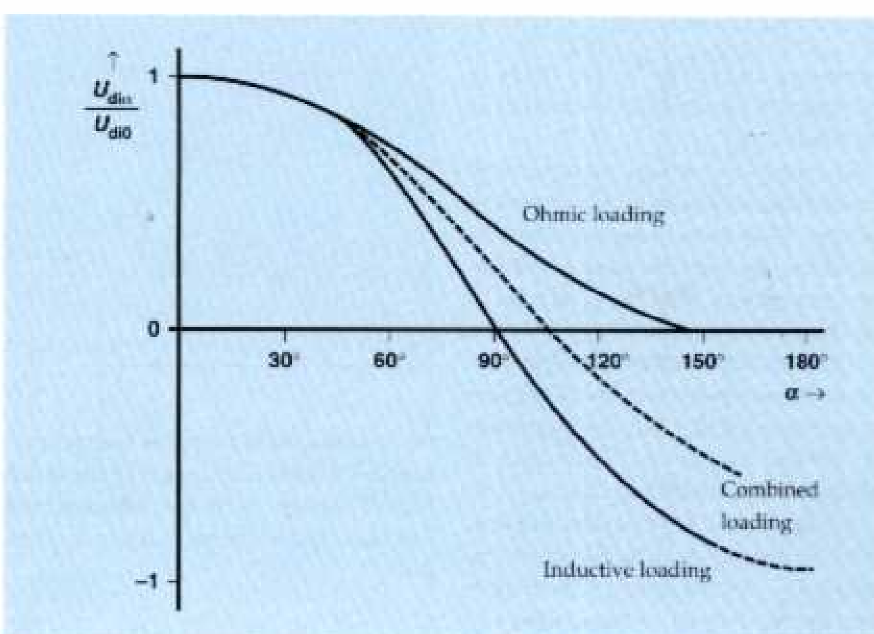


Fig. 211: Control characteristic

Gate controlled turn-off switch (GTO)

This thyristor can be turned off by means of an additional connection, the commutation being considerably reduced with converters that are self-commutating.

When this power semiconductor turn-off switch is present in the power converters the frequency thyristors are gradually compressed in the self-commutating power converter circuits.

At the same time the technical characteristics show an improvement. Due to higher clock frequencies in pulse width modulated systems - with GTO inverted rectifiers a level of 200 Hz to 1 kHz is attained - the harmonic current flows are reduced in the machine. This process results in a reduction of the harmonic torque, a higher degree of machine efficiency by reducing the additional armature losses and a reduction of noise spin-off. But these are not the only possibilities.

Power MOS transistors, bipolar transistors and, in addition to the GTO thyristors, IGBTs will shape the power converter picture in the near future. This will lead to further increase in clock frequency, with bipolar transistors to 3 kHz, with IGBTs to 5 kHz and higher.

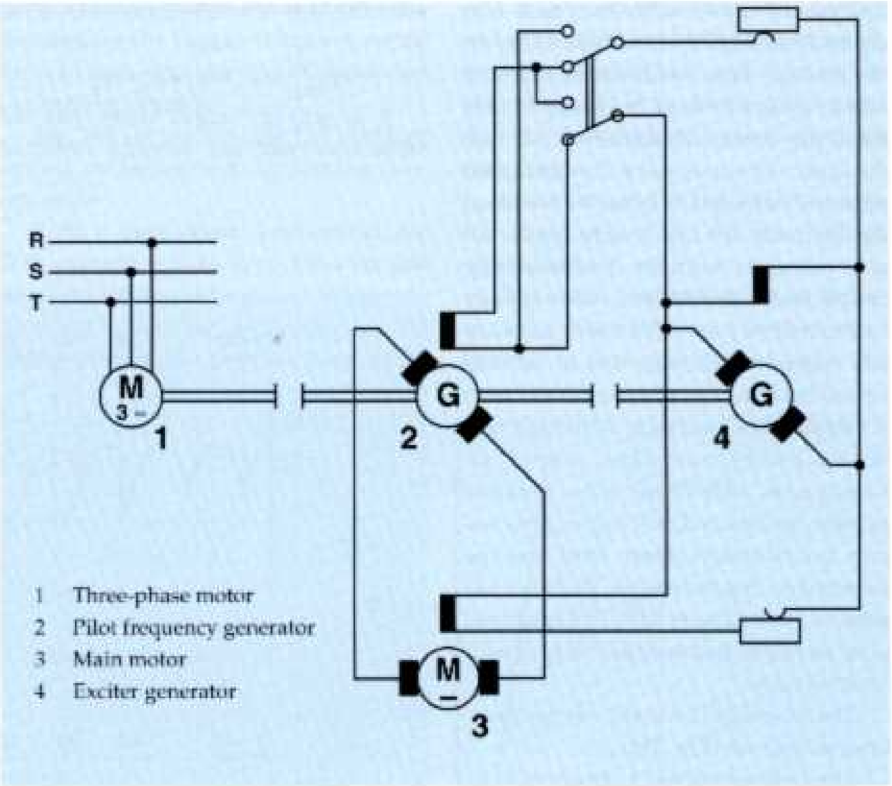


Fig. 212: Leonard circuit

Power transistor

Transistors are known in the fields of entertainment and control electronics as amplifiers.

In control technology these turn-off semiconductors are used as contactless switches. They have to a great extent replaced the electron tube as an amplification component and the relay as a switching component.

The transistor consists of a semiconductor monocrystal with two PN junctions in series (**Fig. 213**). The electrodes are labelled as follows: E (emitter), B (base) and C (collector).

In **Fig. 213** the base emitter diode is poled in the conducting direction via voltage U_1. Approximately 1 % of electrons are discharged as base current I_B via the base connection. The remaining 99 % of electrons from the emitter current are absorbed by means of the much higher voltage U_2 at the collector base junction, and collector current I_C then flows.

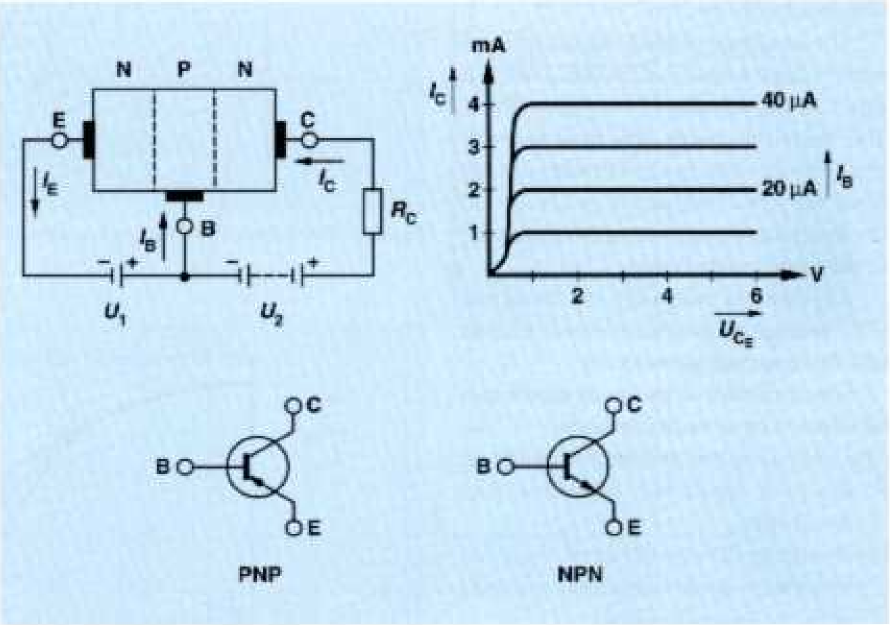


Fig. 213: Wiring diagram (top left), characteristic (top right) and symbol for a transistor

The following equation applies:

$$I_E = I_C + I_B$$

and

$$I_C = B \cdot I_B$$

where B = 20 to 500, the current amplification factor.

The characteristic field shows that, after exceeding the collector emitter voltage U_{CE}, which is identical to the diode threshold voltage, the collector current I_C is virtually constant.

Looking at the switch symbols of the thyristor the arrow indicates the flow direction of the emitter current, as we differentiate according to sequence of layers between NPN and PNP transistors.

Power capacitor

A capacitor consists of two metal electrodes insulated against each other. It offers the possibility of storing electrically small quantities of electricity from several millicoulombes upwards.

With frequency converters capacitors are mainly used as:
- filter capacitors for voltage,
- commutator capacitors for energy recovery and
- buffering or supporting capacitors

Inductor

We differentiate between air and iron inductors. They are used in power converters for:
- smoothing current,
- commutation and
- limiting current flows.

Functions

With the help of the components described above the different functions of the power converter are realised, such as those of:
- rectifiers,
- inverted rectifiers,
- D.C. chopper converters and
- antiductors.

Frequency converters for speed-variable three-phase drives normally consist of several sections of converters put together, which, according to their circuit design are either intermediate circuit converters with impressed voltage and impressed current or as direct converters.

The individual sections of the converter together with their auxiliary devices have several individual functions to fulfil during energy conversion from three-phase mains with constant voltage and frequency through to the speed-variable unit.

The rectifying function

The rectifying function is achieved either by means of uncontrolled rectifiers with diodes or by controlled rectifiers with thyristors. The preferred power converter circuit is the bridge connection, either in the form of a single-phase bridge for low-power applications or a three-phase bridge connection (six-pulse). Bridge connections do not require a middle conductor or star point and can be driven without the need for a transformer in the mains circuit. The six-pulse three-phase bridge connection, with which six valves can be activated during one cycle at regular intervals, can be used in both rectifier or inverted rectifier operation. It can be two star connected systems in series with 60° phase shift. Due to this phase shift the two three-pulse component voltages are added together to form a six-phase D.C. voltage (**Fig. 214**).

In each case the D.C. current flows via two valves connected in series, corresponding to a double valve voltage drop. The output voltage is relatively stable. With an uncontrolled converter the A.C. voltage content will amount to a mere 4.2 %. The ideal D.C. voltage U_{di} is 513 v when connecting to 3 x 380 volts. **Fig. 215** shows the circuit, the voltage progression and the control characteristic of a semi-controlled (left) and a fully-controlled (right) three-phase bridge connection. The total D.C. voltage at the semi-controlled bridge is formed from the constant component or positive voltage, and the uncontrolled section or negative component voltage. Due to the safety intervals in inverted rectifier operation of the controlled section the total D.C. voltage can only be reduced to approximately zero.

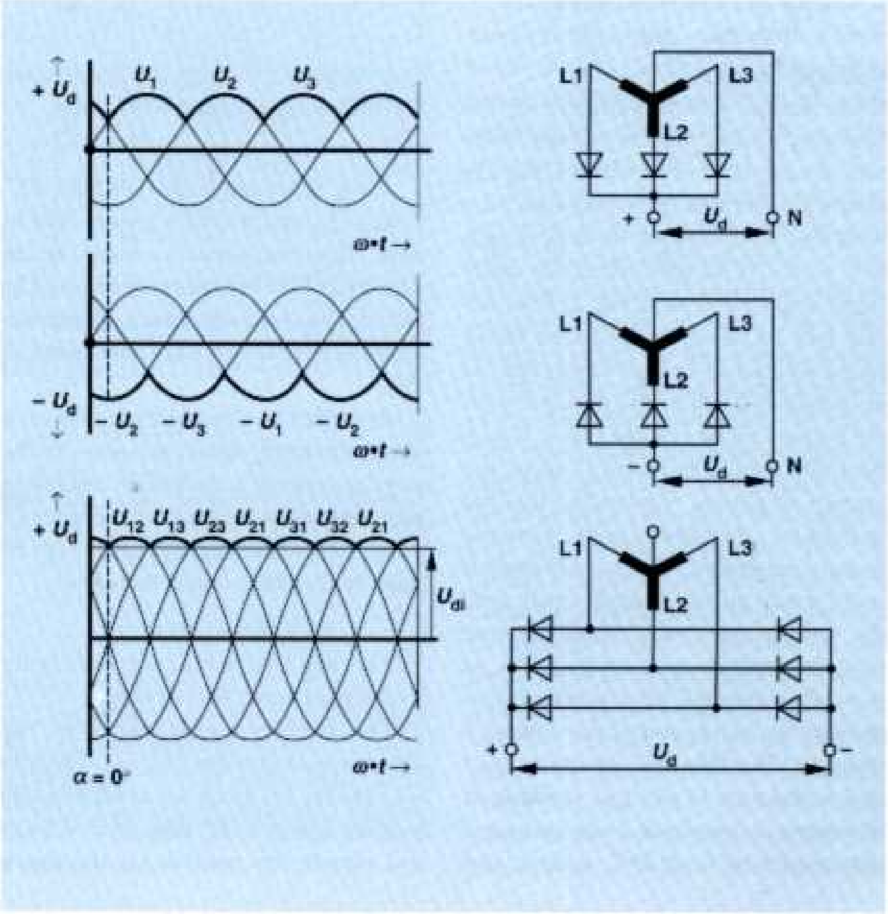


Fig. 214: Characteristic field and wiring diagram of a six-pulse three-phase bridge circuit

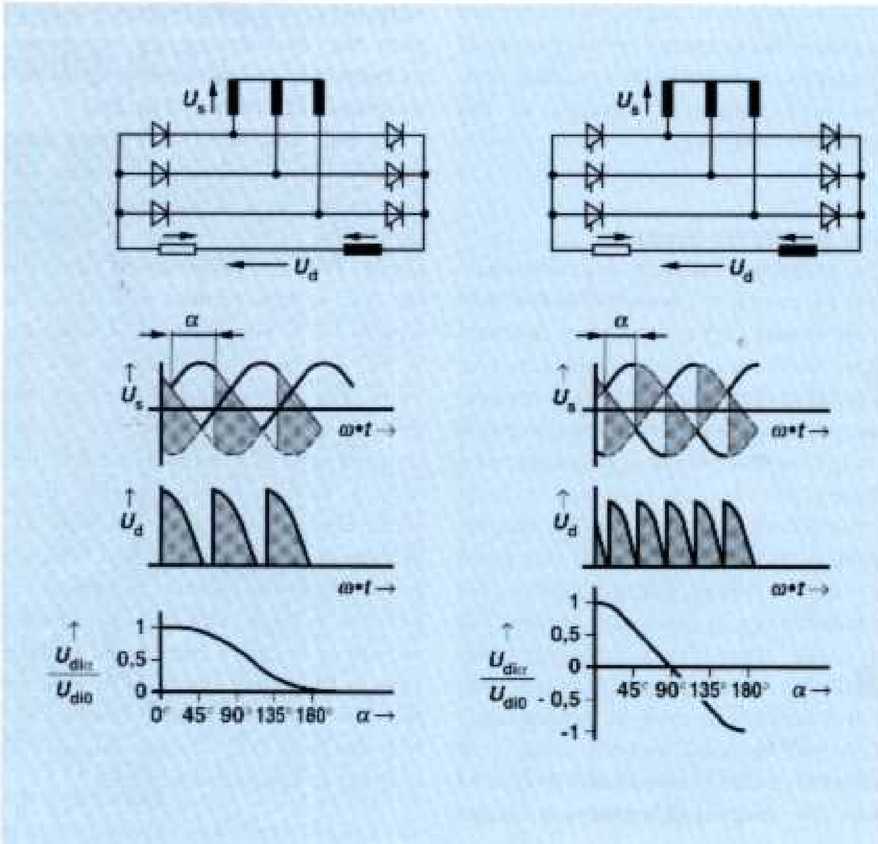


Fig. 215: Characteristic field and wiring diagram of a partially controlled (left) and completely controlled (right) three-phase bridge circuit

Voltage control

If the rectifier, which is situated on the line side, is fitted with thyristors, there will already be voltage control. **Fig 216** shows a controlled three-phase rectifier with mid-point tapping for supplying a D.C. motor. If the gate-controlled delay firing angle $\alpha = 0$ as in **Fig. 217** the circuit will act as an uncontrolled three-phase rectifier in bridge connection, as in each case it will be the thyristor with the highest anode voltage that will conduct. Smoothing inductor D keeps the current amplitude constant, so that the individual thyristors send virtually rectangular pulses. With a delay angle $\alpha = 30°$ every anode remains open until the voltage goes over centre, because only then will the next anode ignite. At [illegible] through the valve, even if the voltage is negative. The flow will be maintained due to inductor D which is working as an energy accumulator. Flow volume is determined by load, D.C. voltage and thus the power through the firing angle. The following equation will apply:

$$E_{g\alpha} = E_g \cdot \cos \alpha$$

If the firing angle $\alpha = 90°$, the D.C. voltage E_g and thus the power will be zero. The traditional Leonard circuit with 65 to 70 % efficiency, shown in **Fig. 212**, can be replaced using a system described above which has an efficiency of 95%.

Reverting to time delay, however, is not without its disadvantages. As the current in the transformer windings lags behind the voltage, in the first approximation the angle of shift will be equal to the firing angle, therefore

$$\cos \varphi = \cos \alpha.$$

This will lead to a poor power factor [illegible]

The thyristor circuit shown in **Fig. 215** can also convert D.C. power into A.C. power i.e. it can act as an inverted rectifier. For $\alpha = 30°$ (**Fig. 215**) voltage and current are positive i.e. the mains circuit supplies energy to the actuators. If the firing angle is now increased, operating conditions will arise where voltage and current have different signs. In this case energy will be fed back from the actuator to the mains.

If $\alpha = 90°$ the energy flow in both directions will be equal. If $\alpha > 120°$ the energy from the actuator will prevail over that fed back to the mains circuit. The unit will act as a generator, decelerating and supplying three-phase current energy back to the mains.

As already described, the output voltage for uncontrolled rectifiers is constant. Voltage control can then be achieved by either

- a downstream D.C. chopper converter
  or
- pulse width modulation of the power converter on the machine side, the inverted rectifier.

D.C. chopper converter

For a power of up to around 140 kW intermediate circuit frequency converters with impressed voltage are used. These frequency converters (**Fig. 218**) consist of an uncontrolled input rectifier, an intermediate circuit with impressed voltage, a downstream direct D.C. converter, the so-called chopper, a variable voltage intermediate circuit and a self-commutating inverted rectifier.

This drive works in four-quadrant operation i.e. it is suitable for both motor and generator operation in both directions.

The supply voltage with this type of frequency converter is first rectified and is then converted into a voltage of variable frequency by means of a second power converter (**Fig. 219**). At varying frequency the mean value of the voltage will reduce from left to right. If the motor voltage is set proportional to the frequency, magnetic flow and torque will remain constant. If the frequency is increased at constant voltage, the power will remain constant. Change of direc- [illegible] drops to below 5 Hz the motor will no longer rotate. Three-phase squirrel cage motors, reluctance motors or permanently excited synchronous motors can be used.

Fig. 216: Controlled three-phase rectifier in mid-point tapping

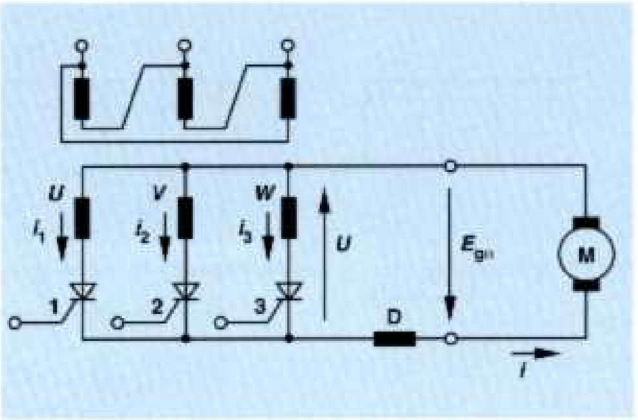
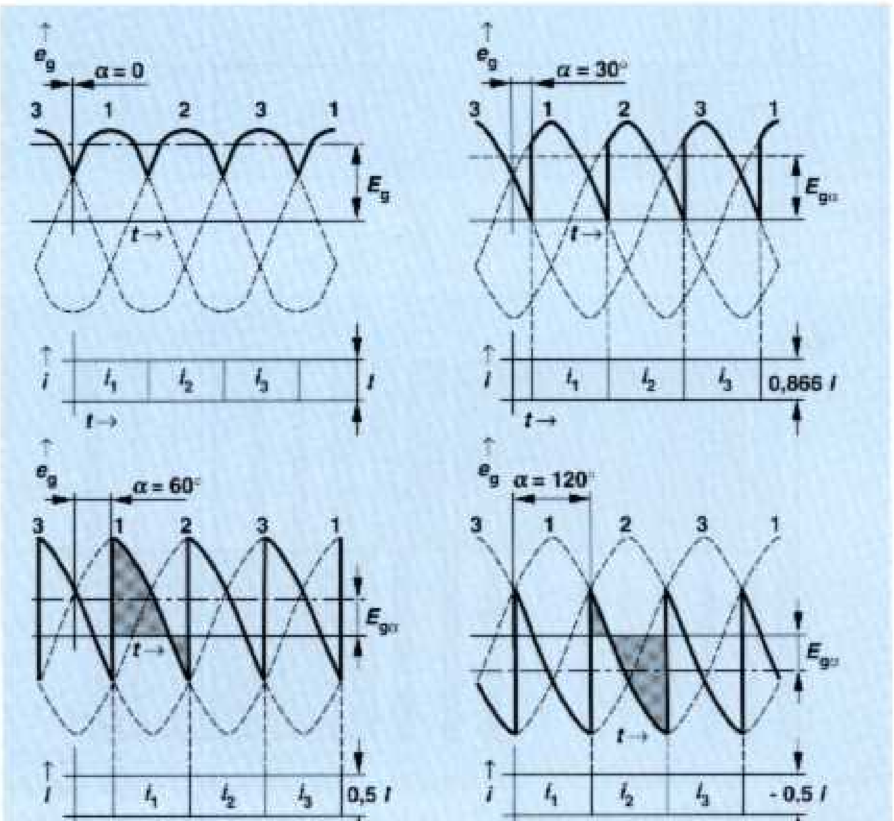
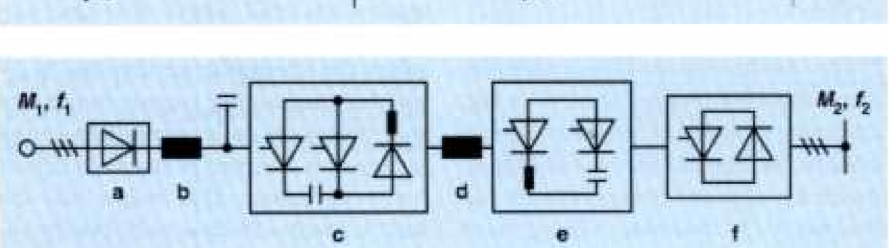


Fig. 217: Control of D.C. voltage via delayed firing





a Uncontrolled input rectifier
b Filter
c Variable intermediate circuit volatage via chopper
d Filter
e Erase device
f Self-commutating inverted rectifier by means of diodes

Fig. 218: Intermediate circuit frequency converter with impressed voltage

Pulse inverted rectifier

The pulse inverted rectifier is an intermediate circuit frequency converter with a constant, load-independent intermediate circuit D.C. voltage. The section of the converter on the machine side, the so-called pulse inverted rectifier, serves here as an actuator for both frequency and motor voltage. The voltage is adjusted during an output cycle by reversing it several times in each phase (**Fig. 220**). This produces voltage blocks at the output of each phase. The relevant mean values of these, calculated over one switching cycle, are changed such that they follow the momentary values for the required output voltages or that they supply the required motor current with the minimum harmonic oscillation possible.

Due to its special features the pulse frequency converter can be universally used for power converter-fed three-phase drives, which are able to carry out practically every task required of a drive. The introduction of turn-off semiconductor valves and the exploitation of microelectronics have made it more economical.

The pre-pulsing process applied to three-phase squirrel cage and synchronous motors with self-commutating power converters meets high technical requirements and covers a power range of up to approx. 4000 kW.

An intermediate circuit frequency converter is used in this process, with both converter sections being de-coupled via an intermediate circuit with capacitive energy accumulators. Frequency control of the output voltage occurs in the self-commutating section of the frequency converter. Its amplitude must be adjusted to be approximately proportional to frequency in order to achieve a frequency-independent flow in the unit connected. The frequency is matched by generating a fundamental supply voltage wave, variable in frequency and amplitude, as the mean value of a high frequency rectangular voltage of unequal positive and negative voltage time spans. **Fig. 221** shows the progression of the supply voltage in the stator coil of a three-phase motor with control by the pre-pulse process. U_1 represents the supply voltage, U_2 the voltage mean value. As the time when switching is to occur is freely selectable for the reversal process, the supply voltage time spans can be selected with positive or negative polarity, so that the mean value follows a pre-set command value.

Frequency and amplitude of the sinusoidal command value depend on the required speed and magnetisation of the squirrel cage motor. The motor can be put into four-quadrant operation, the positive and negative speed range being the same.

If, with an asynchronous motor, the stator current is maintained at a constant value by flow control, whereby the voltage time span of the frequency converter is influenced accordingly, and if the supply frequency is controlled so that the armature frequency remains constant with each revolution, then the torque will also remain constant. The machine can then be driven with the breakdown torque. It can be driven in this mode over the whole speed range in all four quadrants, even at low speeds and at standstill.

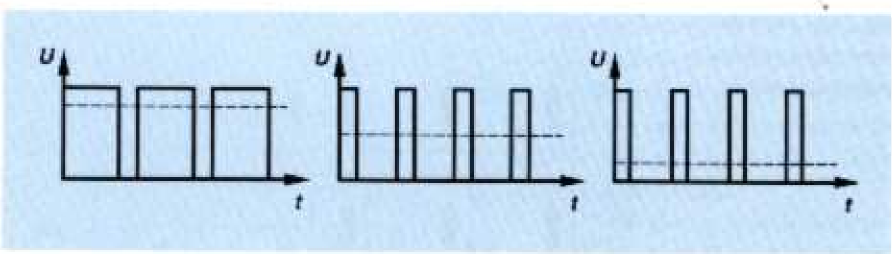


Fig. 219: Voltage mean value at variable frequency by means of chopper

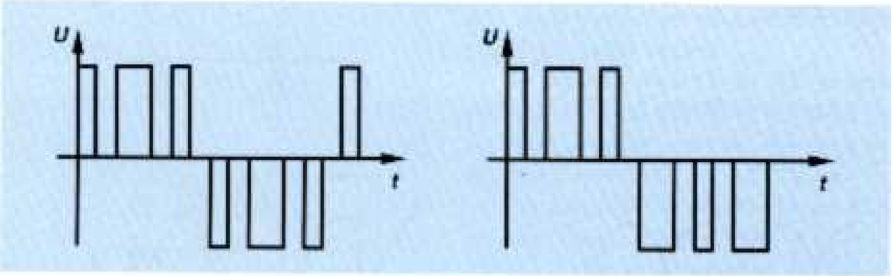


Fig. 220: Pulse inverted rectifier

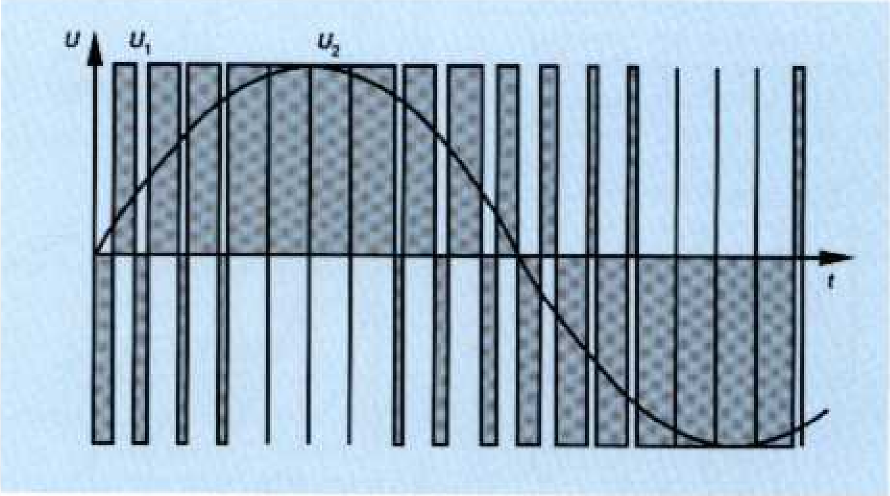


Fig. 221: Control by the prepulse process

Inverted rectifier

The inverted rectifier converts D.C. energy into A.C. energy. During the deceleration process the inverted rectifier can also assume the function of a rectifier.

Let us take the example of a simple mechanical model. Pre-requisite is a PN (positive-negative) D.C. current e.g. at the output of a rectifier.

Three reversing switches, actuated by three cam plates offset by 120°, are connected to a D.C. system (**Fig. 222**).

If the cam plate is rotated in a clockwise direction the three terminals U, V and W are connected alternately to the positive P and the negative N polarity of the D.C. system (**Fig. 222A**).

A voltage that is driving the load current can, however, only be produced if there is a potential difference between the two terminals. The interlinked voltage is shown in **Fig. 222B**.

Fig. 222C shows the star voltage, which is decisive for the flow of current in a motor coil.

As to be expected with a cam actuated switch, the output voltage of this inverted rectifier is rectangular.

The mechanical reversing switches of the inverted rectifier shown in **Fig. 222** can be replaced by power converter connections. **Fig. 223** shows a switch replaced by a power semiconductor via thyristors, and transistors being used instead of turn-off semiconductors. The circuit with transistors can be turned off and therefore makes things easier for the following reason.

As a thyristor, due to its physical characteristics, cannot erase a D.C. current, the circuit must be extended to contain an erase device, consisting of an erase thyristor and commutating capacitor. This capacitor is charged up for the commutation to such an extent that when the relevant erase thyristor is ignited a discharging current builds up, that tries to close in opposition to the conducting direction of the main thyristor to be erased. This means that the load current will flow only briefly via the capacitor and the current in the main thyristor will be reduced to zero. The main thyristor will thus perform a blocking function and the capacitor will be recharged.

If using a reversing switch with transistors or GTOs as these, unlike thyristors, can be turned off, no further components will be required for the commutation.

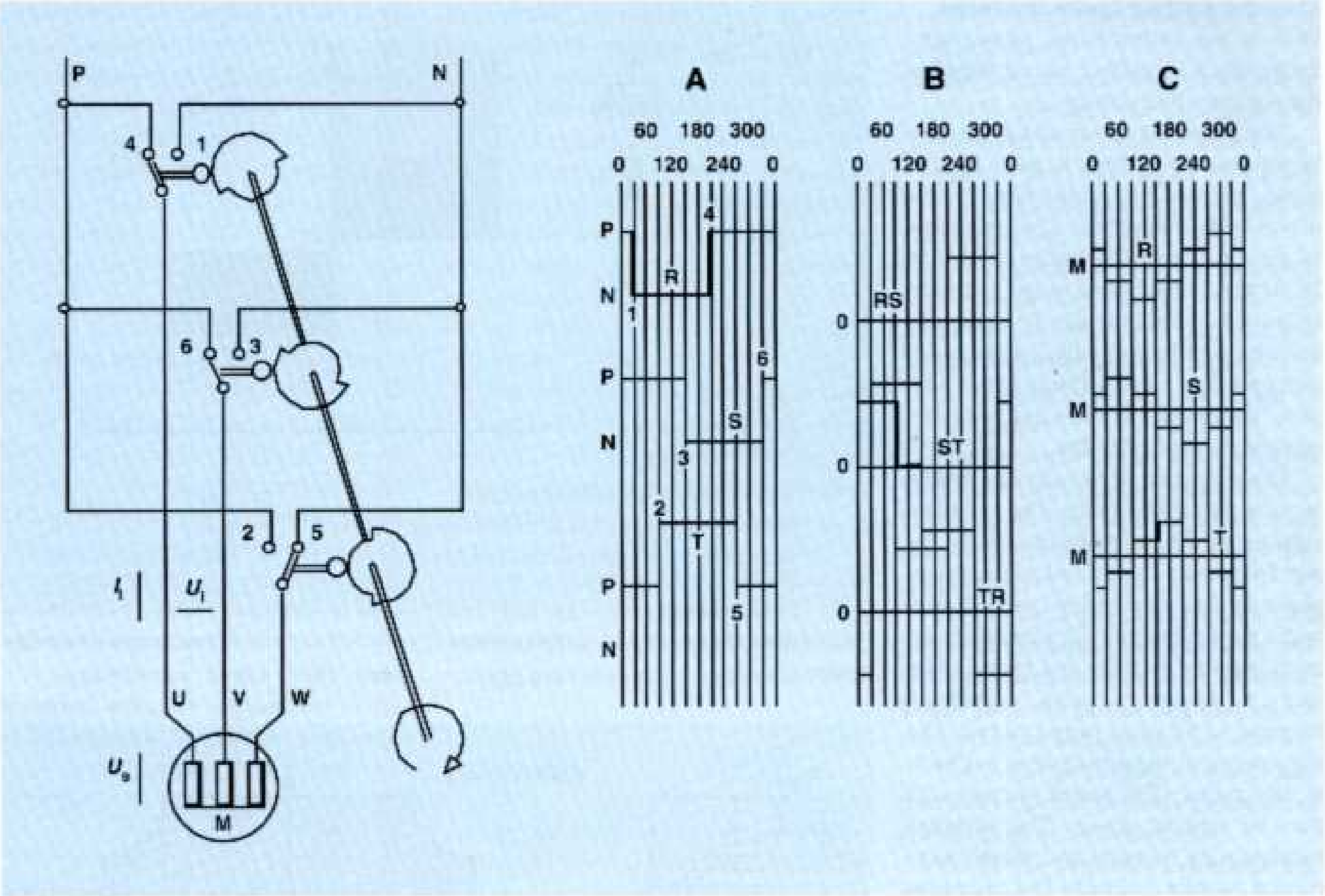


Fig. 222: Mechanical inverted rectifier

Commutation in a power converter means the transfer of a load current from one branch of the converter circuit to the next branch.

Additional reversing diodes will be required in the inverted rectifier if converting with a voltage intermediate circuit. With an ohmic inductive load, as is the case with drives, after the voltage goes over centre ($U_s = 0$), the load current will continue to flow to the output terminals of the inverted rectifier in the original direction to the centre position.

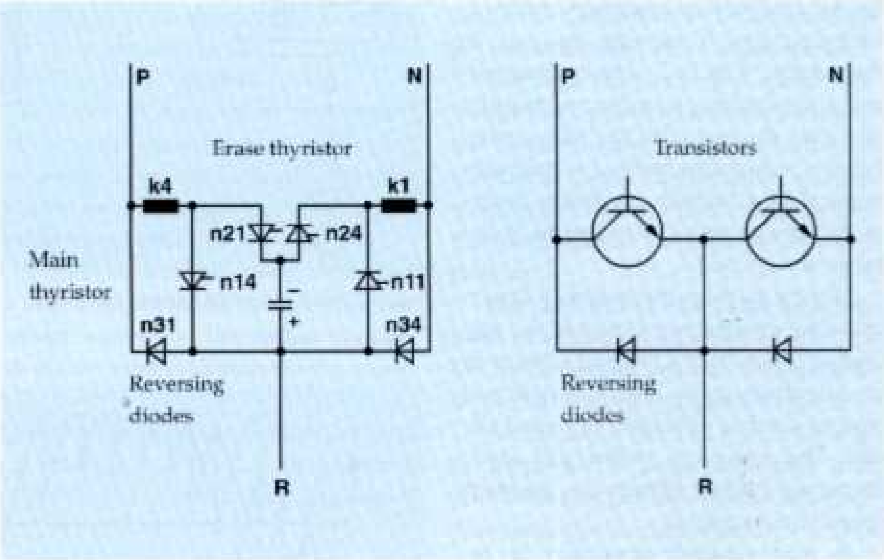


Fig. 223: Wiring diagram of a reversing switch with power semiconductor

Line commutated power converter

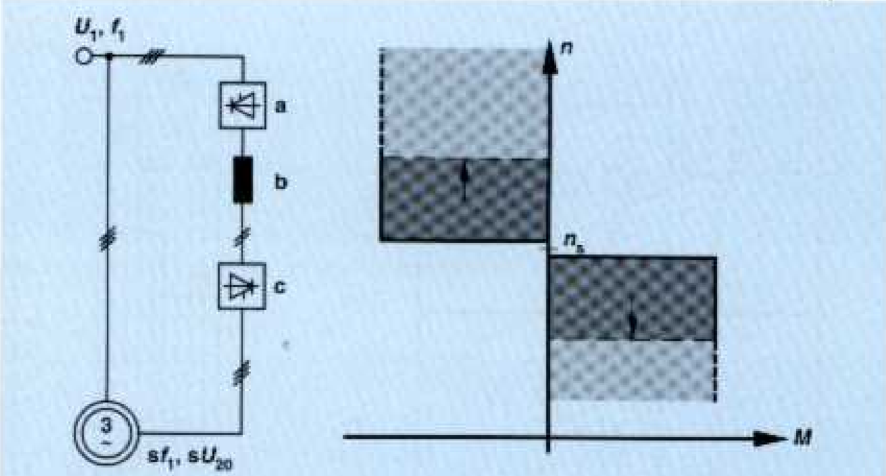
With a line commutated power converter the A.C. voltage circuit supplies the commutation voltage.

The induction motor and static converter cascade system is used for the speed control of individual motor drives with three-phase slip ring rotor motors in the power range of 100 to 25000 kW. The range of speed variation is limited; the motor can be driven in one direction (drive) only (single-quadrant operation); deceleration is not possible. The operating range and basic circuit are shown in **Fig. 224**.

The stator coil of the slip ring motor is connected to the mains. The armature voltage, which is proportional to the slip frequency, is rectified via a three-phase bridge connection with diodes and is then converted via a line commutated inverted rectifier and transformer into a frequency and amplitude matched to the main power circuit. The slip energy of the armature is fed back to the mains. The speed change will thus be without losses. The inverted rectifier with its open and closed loop control system controls the armature voltage so that the motor attains the required speed. The smoothing inductor in the D.C. intermediate circuit is essential for trouble-free operation of the inverted rectifier. It prevents any gaps in the current. The D.C. intermediate circuit and matching transformer must be designed to cope with the slip power. For this reason this method is only economical in a limited regulating range, as the power factor deteriorates if the range is increased.

During start-up the motor is driven up to the smallest revolution of the regulating range with a starting resistor in the armature circuit. Then it is switched over to cascade operation in closed circuit. The induction motor and static converter cascade system has constant speed characteristics.

If four-quadrant operation is required, for a power range of 400 to 10000 kW, line commutated frequency converters without intermediate circuit, so-called direct A.C. converters, are used. The drive motor is usually a high-pole synchronous motor, a three-phase motor with squirrel cage armature or a synchronous motor with a low mains frequency of 0 to 35 Hz. The basic



a Line commutated inverted rectifier
b Smoothing inductor in D.C. intermediate circuit
c Maschine driven uncontrolled rectifier

M Torque
n Speed
n_S Synchronous speed

Fig. 224: Circuit diagram and operating range of induction motor and static converter cascade system

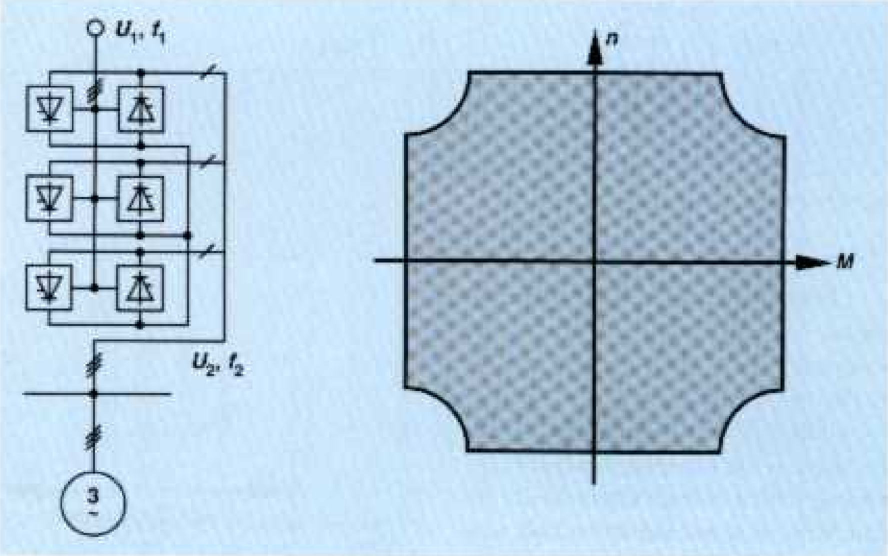


Fig. 225: Basic circuit of a direct converter

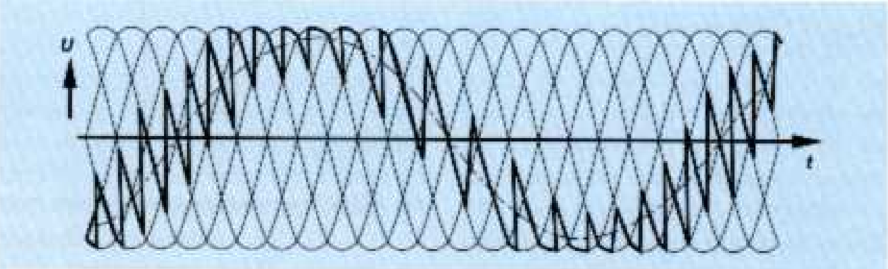


Fig. 226: Voltage progression at the output of a direct converter

circuit can be seen in **Fig. 225**. It shows three line commutated converters in antiparallel three-phase bridge connection i.e. each phase of the stator winding is connected to two power converters. The output voltage of the frequency converter is made up of sections of the mains voltage (**Fig. 226**). The output frequency must therefore be smaller than the mains frequency. As there has to be a bridge available for each semi-oscillation of the current in the motor line, a large number of thyristors are required.

The direct converter is suitable for slow-rotating, powerful drives, cement mixers for example or for common supply of several drives e.g. roller gear bed. There are, however, examples where the mains feedback causes problems. Start-up with the breakdown torque is, however, without problem.

A synchronous motor with direct converter, with the appropriate excitement, can work with a power factor of $\cos \varphi = 1$ and can thus relieve the power converter.

Self-commutating power converter

Self-commutating power converters do not require any external A.C. voltage supply source. The voltage is supplied by an inductive or capacitive energy accumulator, which is situated in the intermediate circuit and is part of the power converter equipment.

For single drives in the power range of 60 to 200 kW and without high technical requirements, converters with impressed current are installed in the intermediate circuit. **Fig. 227** shows the basic circuit.

This process is applied to three-phase squirrel cage motors. First of all the supply voltage is rectified and converted into a voltage system with variable frequency in a second power converter. Both sections of the converter are de-coupled by means of an inductor coil in the intermediate circuit.

The power converter on the machine side is self-commutating and the leakage inductance is a part of the erase circuits. For this reason the converter and drive motor must be approximately matched to each other. Motor current is almost rectangular with a fundamental oscillation of 95 %. The losses in the motor are therefore kept to within acceptable limits.

The drive can once more be driven in four-quadrant operation, the functions of the power converters being exchanged. Start-up creates no problems, although a linear move over centre is not possible, as the drive is liable to come to a standstill if the speed is too low. The self-commutating power converter is used in test stands, extruders, stirring machines and conveyor drives.

Summary

As electric motors working as power converters are fed with non-sinusoidal voltage or current, this places additional demands on them, which affects the operating characteristics.

The three-phase asynchronous motor with squirrel cage armature is the one most commonly used. This is designed for use at constant speed.

In order to limit the initial current without any auxiliary starting device, this motor has a relatively high leakage inductance, resulting in a high current displacement effect.

If such motors are speed controlled by means of power converters with impressed current or impressed voltage, this places extra demands on the motor as follows:

- an increase in operating temperature as cooling from the fan on the motor shaft reduces with decreasing speed, even though the hysteresis loss is also reduced.
- When operating at frequencies above 50 Hz, although the cooling power of the fan is increased, it cannot compensate for the higher hysteresis loss, resulting in a rise in operating temperature.
- Losses increase due to the harmonic content of the converter voltage and current, so that even at nominal speed the permissible torques have to be reduced to prevent a rise in temperature.

The top curve in **Fig. 228** shows the loading capacity against frequency due to hysteresis loss, when working with sinusoidal voltages by means of a machine converter.

The lower curve band shows the permissible torques with converter operation, taking into account harmonic losses. Here the permissible torques are reduced more with powerful machines.

There will be further torque reduction due to the number of pairs of poles and the type of converter used.

Should the frequency exceed 100 Hz the increase in hysteresis losses and associated torque reduction will lead to uneconomical operation. The required nominal torque over a large control range can normally only be achieved by the use of external air or water coolers.

The total efficiency of a speed controlled three-phase motor is made up of the motor efficiency, which contains the additional losses in the motor, and the efficiency of the converter that describes the commutation losses in the inverted rectifier.

Fig. 229 shows the three progressions of efficiency for a self-cooled 315 kW machine with quadratic characteristic plotted against frequency or speed.

As efficiency reduces proportionally to frequency it must be noted that power loss will of course also decrease. The use of power converters with three-

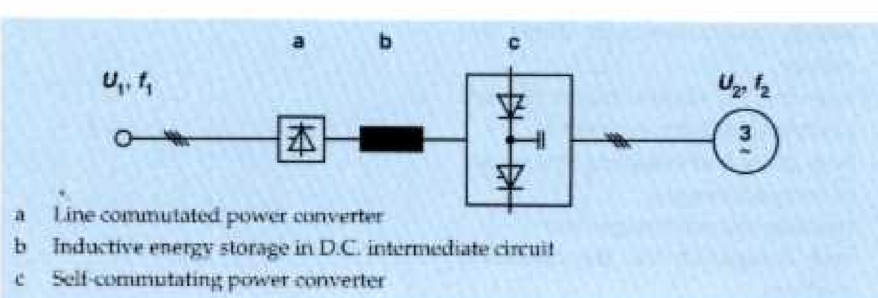


Fig. 227: Intermediate circuit power converter with impressed current

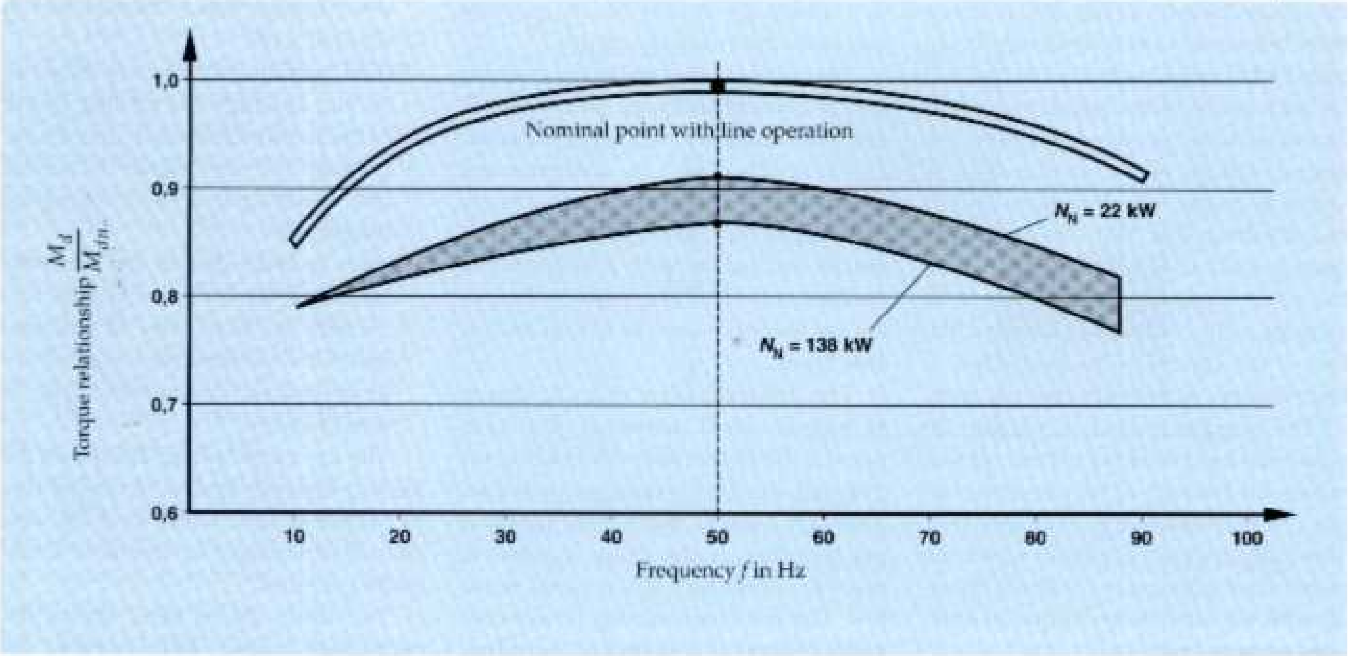


Fig. 228: Permissible torque with speed controlled three-phase motors (Source: Siemens)

phase machines will also have an effect on the power circuit.

All actuators with non-linear characteristics impress influencing variables, which can have a disturbing influence on the behaviour of actuators with linear characteristics. The degree of disturbance here will be dependent upon the impressed values and the reaction of the mains circuit. The most prominent influencing variables are power factor, harmonics of the mains current and voltage drops for commutation. There may also be non-typical components and high-frequency parts in the mains current.

In a direct comparison between speed controlled electric motors and secondary controlled drives, the hydrostatic drives fare reasonably well.

However, if we compare the market volume of the controlled electrical drive with the hydrostatic drive used mainly in industrial mechanical engineering, the hydrostatic drive appears only mediocre.

Substitution relationships are not evident to a great degree.

D.C. technology is not used at a high level. A.C. technology, on the other hand, is being increasingly implemented, and considerable new developments in recent years have opened up new areas of application for electrical drives.

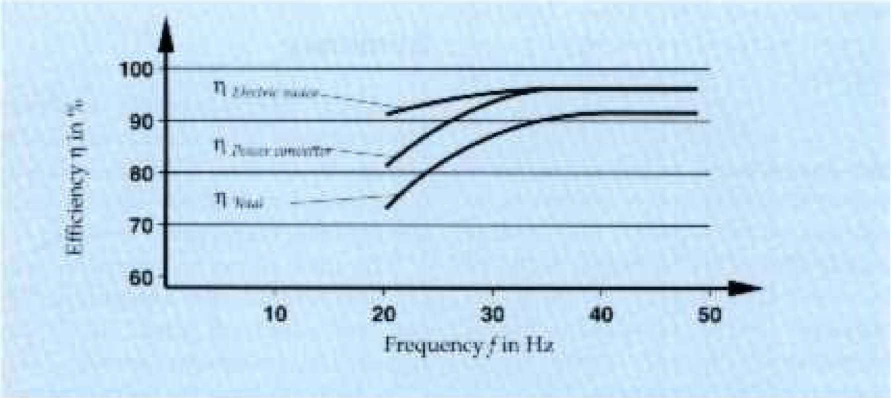


Fig. 229: Efficiency against frequency (Source: Siemens)

However, hydrostatics is still not widely accepted in spite of the obvious inherent technical advantages it has over electrotechnology, such as:

- low natural moment of inertia,
- high dynamic response, at present speed variations of 80000 rpm / sec for a 100 kW machine as opposed to 7500 rpm / sec for an electrical drive,
- unlimited four-quadrant operation,
- energy recovery electrical, via hydraulic accumulators or direct hydraulic,
- relieving the electric mains of peak loading (peaks are expensive),
- both drive and complete system are of compact design,
- possible cost advantages and
- heat dissipation via the operating medium.

Continuing automation, coupled with the necessity to improve on economic efficiency of machines and systems, demands increased implementation of speed-variable drive systems.

This opens up yet another area for secondary control, as the inherent failings of conventional hydrostatic drives do not exist with this type of drive.

Information on Project Design of Secondary Controlled Drives

Introduction

Before establishing rules to be observed when designing secondary controlled drives, we should perhaps summarise, this time from a different aspect, the basic differences between secondary control and conventional hydrostatic drive technology.

Fig. 230 shows the different characteristics of system values

where

n_2	=	speed of secondary unit in rpm,
Δp	=	pressure differential in bar,
Q_2	=	flow of secondary unit in L/min,
V_{g2}	=	displacement of secondary unit in cm³,
M_{d2}	=	torque of secondary unit in Nm and
P_2	=	power of secondary unit in kW.

of a primary controlled fixed displacement motor and a secondary controlled unit at speed stepped response (**Fig. 231**).

The conventional drive system reacts to a change in torque with a change in operating pressure differential. In addition there is a direct relationship between the swivel angle α_1 of the primary unit and the drive speed $n_2 \Rightarrow$ flow coupling.

This rule also applies to hydraulic motors with variable displacement. With flow coupled drives, speed will therefore increase as the displacement of the hydraulic motor decreases.

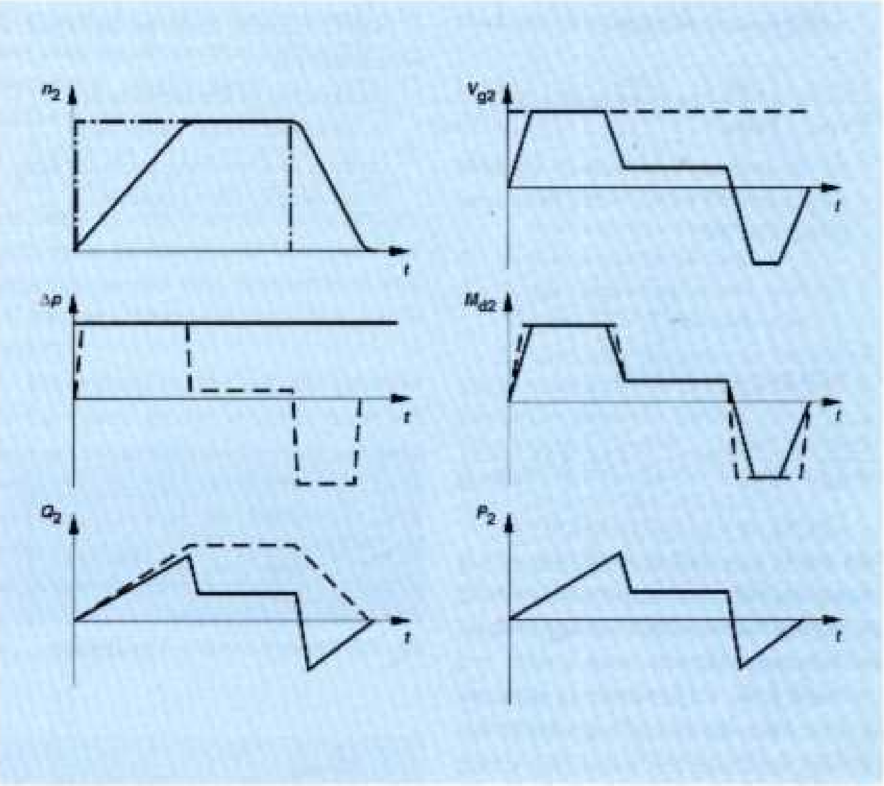
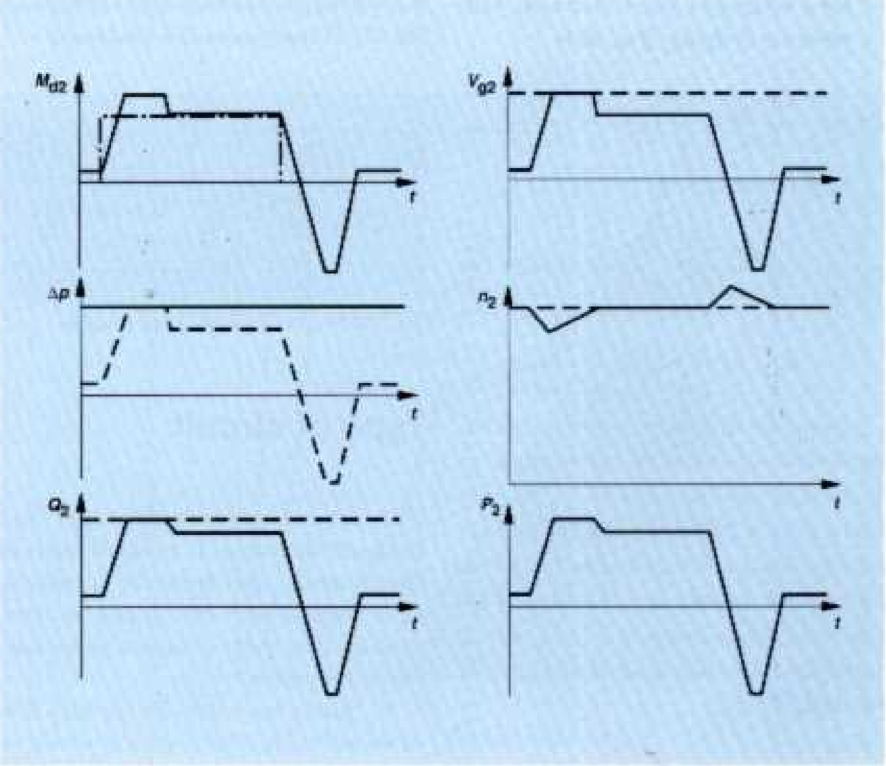


Fig. 230: System response with stepped speed



Key to Fig. 230 and Fig. 231:

- – – Primary controlled fixed motor
- —— Secondary controlled variable motor
- –·– Command value

Fig. 231: System response with stepped torque

If we convert this into mathematical form we obtain

- for a primary controlled fixed displacement motor with flow coupling (**Fig. 232**)

$$n_2 = \frac{Q}{V_{g2\max}} \cdot \alpha_1$$

with pressure reaction

$$\frac{\Delta p}{\Delta p_{\max}} = \frac{20\pi}{V_{g2\max}}\left[2\pi J_g \frac{\Delta n_2}{\Delta t} + M_l + M_r\right]\frac{1}{\Delta p_{\max}}$$

- for a primary controlled variable displacement motor with flow coupling (**Fig. 233**)

$$n_2 = \frac{Q}{V_{g2}} \cdot \frac{\alpha_1}{\alpha_2}$$

with pressure reaction

$$\frac{\Delta p}{\Delta p_{\max}} = \frac{20\pi}{V_{g2}}\left[2\pi J_g \frac{\Delta n_2}{\Delta t} + M_l + M_r\right]\frac{1}{\Delta p_{\max}}$$

The secondary controlled drive system reacts to a change in torque with a change in flow requirement i.e. as with an electric motor a high loading will result in a large current.

There is no longer a direct relationship between the swivel angle α_1 of the primary unit and the secondary speed n_2 in the stationary state.

If we convert this into mathematical form we obtain

- for a secondary controlled unit with pressure coupling (**Fig. 234**)

$$M_{d2} = \frac{\Delta p V_{g2}}{20\pi} = \frac{\Delta p V_{g2\max}}{20\pi} \cdot \frac{\alpha_2}{\alpha_{2\max}} \cdot \alpha_2$$

with flow reaction

$$\frac{\alpha_2}{\alpha_{2\max}} = \frac{20\pi}{V_{g2\max}}\left[2\pi J_g \frac{\Delta n_2}{\Delta t} + M_l + M_r\right]\frac{1}{\Delta p}$$

$$Q = V_{g2} \cdot n_2 = V_{g2\max} \cdot \frac{\alpha_2}{\alpha_{2\max}} \cdot n_2$$

However, in order to cover a sudden peak flow requirement, for example to bridge the final swivel time of the primary unit, a hydraulic accumulator is usually required. This accumulator can also be used to take up potential energy produced from lowering a load with a winch or from deceleration of a mobile machine.

Either a pressure controlled hydraulic pump or else a pressure system with impressed operating pressure is used for the energy supply on the primary side. The peak flow requirement, e.g. during acceleration, can then be covered by the hydraulic accumulators.

Safety circuits must be installed to secure against

- breakdown of the tacho-unit
  ⇒ no speed actual value and
- rupture of a working pressure line
  ⇒ no deceleration possible.

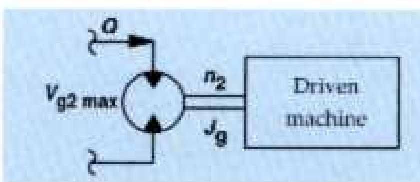


Fig. 232: Primary controlled fixed motor

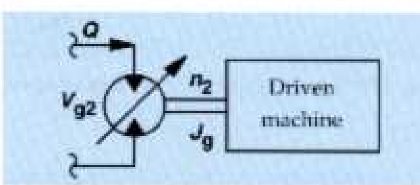


Fig. 233: Primary controlled fixed motor

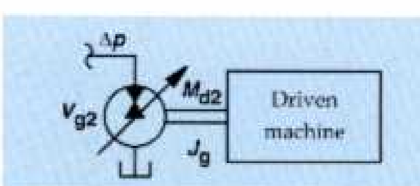


Fig. 234: Secondary controlled variable motor

Type of circuit

With secondary controlled units four quadrant operation is possible even in open circuits. The hydraulic accumulator system in this case is not so complex, as only high pressure accumulators are necessary.

It must, however, be noted, that when lowering a load with a winch or during deceleration, the units switch from motor to generator operation i.e. flow direction of the fluid is reversed.

For this reason, the permitted speeds for open circuit and pump operation must be used as opposed to those for closed circuit operation.

Short-term movements over centre and into pump operation must also be reckoned with during high response closed loop control operations. When selecting an open loop circuit therefore, it must be ensured that

- the maximum permissible speed in pump operation is not exceeded,
- the diameters of the suction line are sufficiently large and
- the suction port of the secondary unit is mounted below the oil level.

The advantages of the open circuit are still maintained if the so-called pre-fill operation is carried out. In this operation the suction line is pre-filled by means of an auxiliary pump in exceptional cases with a flow of $Q_{fill} = 1.1 \cdot Q_{max}$ at a pressure of approx. 15 bar.

Fig. 235 shows the basic circuit of a secondary unit (2) in pre-fill operation. The filling pump (1), which can be coupled to the primary unit, must always be able to generate the maximum flow of the secondary unit in generator operation, any short-term peak requirements being covered by hydraulic accumulator (3).

Due to the limitations shown it is therefore preferable for secondary controlled operation to be in closed circuit. In this case, in addition to a high pressure accumulator, the low pressure line must also be equipped with a hydraulic accumulator.

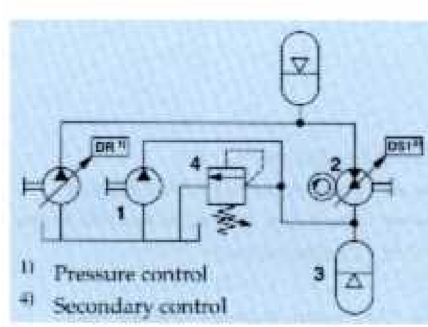


Fig. 235: Secondary unit in pre-fill operation

Pressure level

A pressure level of 280 bar has become standard in test and simulation technology as applied to the automobile industry. As constant pressures of up to 315 bar are permissible with the valve programme the operating pressure may be raised to this level without problem. This is because, with the exception of the acceleration phase, the swivel angle is always smaller than $\alpha_{2\,max}$, and the swivel angle has a greater effect on the service life of axial piston units than the operating pressure (see chapter "Axial Piston Units Designed for Use with Secondary Control").

In order to produce a higher torque required for a short time, with start-up and break-out of larger moments of inertia for example, the operating pressure with type A4VS axial piston units may be raised to 400 bar, as the servo valve in this phase does not have a control function.

A constant pressure of below 150 bar is not economical and is therefore not advisable.

Static dimensioning

(Displacement)

We can see from the acceleration phase, that a swivel angle reserve of between 10 - 20 % must be allowed for when calculating the displacement with respect to the expected disturbance torques. With motor operation the displacement of the secondary unit is calculated as follows:

$$V_{g2max} = \frac{20 \cdot \pi \cdot M_{d2max}}{\eta_{mh} \cdot \Delta p} \cdot f_{m}.$$

with $f_m = 1.1$ to 1.2.

The torque reserve is taken into account with factor f_m in the event of a disturbance torque occurring

where:

f_m = torque reserve factor,

M_{d2max} = max. torque of secondary unit in Nm,

Δp = pressure differential in bar,

V_{g2max} = max. displacement of secondary unit in cm^3,

η_{mh} = mechanical hydraulic efficiency.

For an operating pressure range of 200 to 315 bar, with a swivel angle of $\alpha_2 / \alpha_{2\,max} = 0.8$ and at maximum speed, $\eta_{mh} = 0.94$ to 0.95.

η_{mh} decreases as the swivel angle decreases. If the swivel angle is smaller than 10° calculation of the displacement by means of η_{mh} is no longer suitable. In this case the theoretical calculated displacement should have an extra 1° added to the swivel angle ($\alpha_{2\,max} = 15°$).

For drives that have to start up with a full load, as with winches for example, start-up efficiency of the secondary unit must be taken into account. This depends on the size, temperature and oil viscosity. The mean value is $\eta_A = 0.70$ to 0.73.

Depending on the angular position of the rotary group, whether there are four or five pistons in the high pressure area, η_A can vary by approx. ± 3%. If start-up efficiency has been taken into consideration when determining the displacement, then the control reserve of 10 to 20 % need not be included in the calculation.

Dynamic dimensioning

(without overshoot)

The dynamic response of a secondary controlled drive can be calculated by means of time factor t_r of the control area (see also page 48):

$$t_r = 2 \cdot \pi \sqrt{\frac{5 \cdot J_g}{\Delta p \cdot V_{g2max}}}.$$

where:

J_g = reflected moment of inertia (referred to the controlled axis) in kgm^2,

Δp = pressure differential in bar,

V_{g2max} = max. displacement of secondary unit in cm^3,

t_r = time factor of control area in sec.

The dynamic response of the swivel angle control circuit is characterised by minimum possible control time t_{swivel}. This control time t_{swivel} can be taken from the technical data of the manufacturer (RE 92055 and RE 92715).

Fig. 236 gives quick information of the speed/stepped response of the unloaded drive in idle running i.e. without load torque M_1 and frictional torque M_r but taking into account the reflected moment of inertia. This idealised consideration of the control area with a purely integral characteristic is usually the least favourable, as generating a load torque has a stabilising effect.

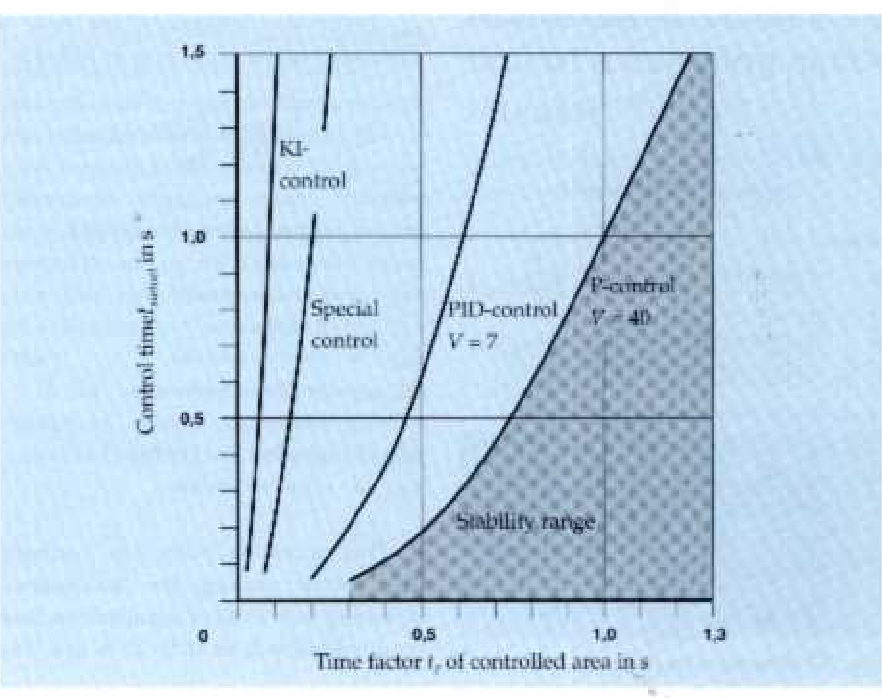


Fig. 236: Stability curve for control setting

Calculation of the reflected moment of inertia

(on the shaft of the secondary unit)

Natural moment of inertia J_{nat} of the secondary unit can be taken from the technical data of the manufacturer. The moments of inertia J_i (i = 1, 2, 3, 4) must be calculated taking into account the gear multiplication ratio (**Fig. 237**).

$$\text{Gear multiplication ratio: } i = \frac{n_{21}}{n_{22}}.$$

where:

n_{21} = input speed and

n_{22} = output speed.

$i > 1$ transfer to lower speed and

$i < 1$ transfer to faster speed.

The reflected moment of inertia is

$$J_g = J_e + J_1 + J_2 + \frac{J_3 + J_4}{i^2}.$$

Example: Calculation of the moment of inertia of a disc:

$$J_s = \frac{\pi}{32} \cdot \rho \cdot d^4 \cdot h..$$

where:

d = diameter in m,

h = width in m,

J_s = moment of inertia in kgm² and

ρ = density in kg/m³.

When converting to another unitary system the following must be considered:

$$J_s = \frac{1}{4} \cdot GD^2[\text{kpm}^2]$$

$$J_s = \frac{1}{4g} \cdot GD^2[\text{Nm}^2]$$

where:

g = acceleration due to gravity in $\frac{\text{m}}{\text{s}^2}$.

J_s = moment of inertia in kgm² and

GD^2 = flywheel effect in kpm² oder Nm².

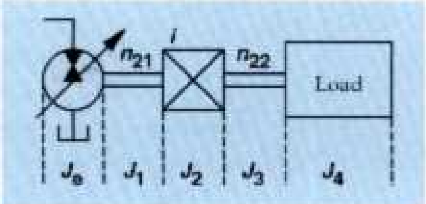


Fig. 237: Moment of inertia

Maximum permissible speeds

The maximum permissible speeds for secondary controlled units (see data sheets RE 92055 and 92715) are actually higher than the speeds that can be achieved in closed circuit with the same size conventional drives.

For example a maximum speed of 2000 rpm is permissible for a conventional hydraulic drive size 250, whereas the secondary unit has a maximum speed of 2600 rpm, although only with a swivel angle of up to 80 %. The speeds specified in the data sheet may not be exceeded, even in brief operation.

As noise emissions of an axial piston unit at constant operating pressure increase proportionally to speed, it is usually advisable to keep the design speed below the maximum speeds.

Determination of flow requirement of the secondary unit

Total flow requirement of the unit is made up as follows from the three different flows:

$$Q_{total} = Q_2 + Q_{Lservo} + Q_{pilot}.$$

With corner power, if $M_{d2\ max}$ and $n_{2\ max}$ occur simultaneously, the flow is calculated in the usual way as follows:

$$Q_2 = \frac{V_{g2max} \cdot n_{2max}}{1000 \cdot \eta_{vol}} \text{ in L/min.}$$

where:

$n_{2\ max}$ = output speed of secondary unit in rpm,

Q_2 = flow of secondary unit in L/min,

Q_{total} = total flow in L/min,

$Q_{L\ servo}$ = zero flow of servo valve in L/min,

Q_{pilot} = pilot oil flow in L/min,

η_{vol} = volumetric efficiency

This operating point can normally only occur during the acceleration phase, so in a state of equilibrium flow requirement will be 10 to 20 % less. The differential flow must then be produced by the hydraulic accumulator until swivel-back to the smaller displacement.

When several secondary controlled drives are run in parallel from a central oil supply, the additional flow requirement of the remaining actuators can be calculated from the output power P_{2i} to be generated at a given time:

$$Q_{2i} = \frac{600 \cdot P_{2i}}{\eta_t \cdot p} \text{ in L/min.}$$

where:

i = 1, 2, 3, ...

P_{2i} = output power of a secondary unit in L/min,

p = operating pressure in bar,

Q_{2i} = flow of a secondary unit in L/min and

η_t = $\eta_{hm} \cdot \eta_{vol}$ = total efficiency.

When calculating the size of units it must be remembered that, with an increase in torque, a larger flow will be demanded by the system than actually needs to be produced from the primary side, the hydraulic accumulators or actuators working as generators.

Zero flow of the servo valve $Q_{L\ servo}$ is calculated as follows:

$$Q_{Lservo} = \sqrt[4]{\frac{\Delta p}{70}} \cdot (0,8 + 0,04 \cdot Q_n) \text{in L/min.}$$

where:

p = operating pressure in bar,

Q_n = nominal flow of servo valve in L/min at 70 bar,

$Q_{L\ servo}$ = zero flow of servo valve in L/min,

Q_{pilot} = pilot oil flow in L/min.

For a nominal flow of the servo valve of 45 L/min and a pressure differential of Δ_p = 315 bar, $Q_{L\ servo}$ will be 5.5 L/min.

This flow must be produced in stationary operation

- M_{d2} = constant and
- n_2 = constant,

i.e. V_{g2} remains unchanged.

With a stepped command value for speed the secondary unit will swivel in minimum possible control time t_{swivel} to its maximum volume.

Table 5 shows the required pilot oil flow Q_{pilot} for this process for all sizes of the A4VS series (see also chapter "Ax-

Displacement in cm^3	Control time in ms	Pilot oil flow in L/min
40	30	12
71	40	16
125	50	23
180	50	23
250	60	36
355	60	36
500	80	48
750	90	70
1000	100	77

Table 5: Minimum control times and associated flow requirement for A4VS series

ial Piston Units Designed for Use with Secondary Control").

These extreme values do not normally need to be specially calculated, as hydraulic accumulators can cover peak requirements.

The actual pilot oil flow Q_{pilot} is dependent mainly on the frequency and value of the speed and torque changes.

Determination of pump size

For industrial systems with several actuators connected in parallel it makes sense to divide the required flow between several pumps. Although considerable space is required, this is outweighed by the advantages, such as

- high redundancy,
- reduction of power losses, as pumps can be disconnected when underloaded and
- shorter control times with smaller sized units, which can act beneficially on the design of hydraulic accumulators.

Note: The size of the hydraulic pump should never be greater than that of the secondary unit.

The required primary side displacement of the pump(s) can be calculated as follows:

$$V_{g1} = \frac{1000 \cdot Q_{total}}{n_1 \cdot \eta_{vol}}$$

where:

Q_{tot} = total flow requirement in L/min,
V_{g1} = displacement of primary unit in cm^3,
n_1 = drive speed of primary unit in rpm and
η_{vol} = volumetric efficiency of the pump(s).

In order to guarantee equal loading when operating several pumps in parallel, the DP pressure control for parallel operation must be used. This controls several pumps by means of a pressure relief valve, so that they set virtually equal swivel angles.

The DP pressure control is essential if mooring pumps are used. If this requirement is not fulfilled, then one pump will normally operate as a generator and the other as a motor, due to the flat p-Q characteristic of disconnected actuators. This will lead to an undesired increase in power losses. In addition to DR and DP controls other positioning devices which fulfil the requirements can be installed, such as

- DRG, remote controllable pressure control,
- HSP, hydraulic adjustment with pressure control function and subordinate positional feedback of the swivel angle.

A sufficiently large primary station is prerequisite for trouble-free operation under secondary control. Flow deficiencies will lead to pressure drops, causing the units to swivel further out, this in turn leading to an increase in flow requirement. Pressure drops that are caused by reaching the limit can be prevented by setting the limit on the motor side. (See also "Electronic Components of a Secondary Controlled Drive System").

If the secondary unit attains the maximum swivel angle the laws of pressure coupling will no longer apply.

Hydraulic accumulator dimensioning

Due to the reciprocal influences of speed and pressure control secondary controlled drive systems without hydraulic accumulators are difficult to stabilise.

One important criterion for stability is the control time of the pressure control for the hydraulic pump, which must be as short as possible. For a smaller nominal size unit the control time will also be shorter. It therefore makes sense to use two or more small pumps rather than one large one.

As the volume of hydraulic accumulator increases speed and pressure control will be de-coupled. This volume also includes the accumulator effect of the hydraulic circuit and the compression volume of the piping system. A longer control time can then be managed.

We therefore differentiate between two types of accumulators. These are:

- the damping accumulator for stabilisation and
- the energy accumulator.

The damping accumulator is used to bridge the different control times of the pressure-controlled hydraulic pump and the servo valve-controlled secondary unit.

Approximate calculation of a damping accumulator

(using the example of two 125 cm^3 displacement axial piston units)

Hydraulic pump: A4VSG125DR,
Control time 100 ms;
Secondary unit: A4VSG125DS1,
Control time 50 ms.

Due to the considerable distance, at least in part, between the primary and secondary sides the reaction speed and the resulting reaction time of the oil column must be taken into account.

It amounts to a distance of 10m:

$$t_1 = \frac{s}{v} = \frac{10}{1300} = 0,0077 \text{ s.}$$

Therefore

$$t_{total} = t_1 - t_2 + t_3 = 0{,}1 - 0{,}05 + 0{,}0077$$
$$= 0{,}058 \text{ s} = 58 \text{ ms}.$$

where:

s	=	length of oil column in m,
t_1	=	control time of hydraulic pump in s,
t_2	=	control time of secondary unit in s,
t_3	=	reaction time of oil column in s,
t_{total}	=	total time in s and
v	=	speed of pressure information in an oil column in m/s.

The hydraulic accumulator must therefore cover the flow requirement of the secondary unit for a period of 58 ms.

If the secondary unit is loaded with a torque shock of 80 % of the maximum torque in idle running at a speed of $n_2 = 2600$ rpm ($\alpha = 1°$) in a state of equilibrium, the secondary unit will swivel to the maximum 15° within 50 ms.

The flow requirement will increase proportionally:

$$\Delta Q = \frac{V_{g2} \cdot n_2}{2 \cdot 1000} = \frac{125 \cdot 2600}{2 \cdot 1000} = 163 \text{ L/min.}$$

The change in flow will then be:

$$\Delta V = \frac{\Delta Q}{60} \cdot t_{total} = \frac{163}{60} \cdot 0,058 = 0,158 \text{ L.}$$

For an assumed operating pressure of 280 bar and a permissible pressure drop of 30 bar the size of hydraulic accumulator can be calculated in accordance with the following formula:

$$V_1 = \Delta V \bullet \frac{1}{\left(1 - \frac{p_1}{p_2}\right)^{\frac{1}{n}}} = 0,158 \bullet \frac{1}{\left(1 - \frac{250}{280}\right)^{\frac{1}{1,6}}}$$

$V_1 = 2{,}3$ L.

Displacement of secondary unit in cm³	Hydraulic accumuator size in Liter
40	1
71	2,5
125	5
180	5
250	10
355	10
500	13
750	20
1000	24

Table 6: Displacement of secondary unit with respect to size of hydraulic accumulator

where:

n_2	=	speed of secondary unit in rpm,
p_1	=	minimum operating pressure in bar,
p_2	=	maximum operating pressure in bar,
ΔQ	=	flow requirement in L/min,
t_{ges}	=	total time in s,
V_1	=	hydraulic accumulator volume in L,
V_{g2}	=	displacement of secondary unit in cm³,
ΔV	=	change in volume of hydraulic accumulator in L and
n	=	polytropic exponent (1.6 at 200 bar and 60 °C).

The size of hydraulic accumulator is also dependent on the following parameters:

- system dynamic response,
- displacement of the units used,
- length of piping,
- pressure level and
- permissible pressure drop.

The hydraulic accumulator should be mounted near the secondary unit. The accumulator pre-fill pressure should amount to approx. 75 to 80 % of the operating pressure.

Low pressure accumulators in the closed circuit should be designed for gas pre-load to 7 bar at a hydraulic accumulator pressure of 15 to 16 bar.

In practice the sizes of hydraulic accumulator as shown in **Table 6** have proved correct for the relevant sizes of axial piston units.

Energy storage

The application of hydraulic drives with energy recovery and storage brings decisive advantages with mobile machinery which use their drive power mainly for short-term acceleration and deceleration of masses. These advantages are as follows:

- The drive power needs only to be designed for an average power requirement.
- The braking energy is stored in the hydraulic accumulator for use in the following acceleration process.
- The internal combustion engine with reduced drive power has a lower fuel consumption with comparable energy requirement.
- Due to virtually constant power output in a favourable area of the characteristic diagram, the specific fuel requirement can be reduced to a minimum quite easily.

With the hydraulic accumulators used, energy exchange is by means of compression and expansion of a volume of nitrogen. System pressure is therefore closely related to the relevant loading condition of the accumulator.

As the power requirement of the secondary unit at an impressed pressure is matched only by varying the flow, it is quite difficult to set the hydraulic accumulator volume by means of the speed and displacement.

With secondary controlled hydrostatic drives it is better, therefore, to use the equation of energy as follows:

for energy storage

$$E_{stor} = E_{mech} \cdot \eta_{total},$$

for energy withdrawal

$$E_{stor} = E_{mech} \cdot \frac{1}{\eta_{total}}.$$

η_{total} is the total efficiency of the whole drive path. It includes the volumetric and mechanical-hydraulic efficiency of the secondary unit, efficiency of the transmission, travel resistance and pressure losses i.e. the total losses occurring.

Depending on the type of storage we differentiate between:

potential energy

$$E_{pot} = m \cdot g \cdot h$$

and

kinetic energy

$$E_{kin} = \frac{1}{2} \cdot m \cdot v^2.$$

With these drives there are usually rotational movements for which the analogue relationships are used:

$$E_{rot} = \frac{1}{2} \cdot J \cdot \omega^2.$$

where:

E_{kin} = kinetic energy in Nm,
E_{mech} = mechanical energy in Nm,
E_{pot} = potential energy in Nm,
E_{rot} = rotation energy in Nm,
E_{stor} = energy to be stored in Nm,
g = acceleration due to gravity in m/s^2,
h = height difference in m,
J = moment of inertia in kgm^2,
m = mass in kg,
v = speed in m/s,
η_{total} = total efficiency,
ω = angular velocity in 1/s.

Calculation of accumulator volume V1

$$V_1 = \frac{E_{acc}(1-n)}{p_1 \cdot 10^2 \left[1 - \left(\frac{p_1}{p_2}\right)^{\frac{1-n}{n}}\right]}$$

where:

E_{acc} = energy to be stored in Nm,
p_1 = min. accumulator pressure in bar,
p_2 = max. accumulator pressure in bar,
V_1 = volume of gas in L,
V_0 = accumulator volume at p_0 in L,
n = polytropic exponent.

The ratio of specific heat capacity for ideal gases $\kappa = c_p/c_v$ to nitrogen at 273 K and 1 bar is $n = \kappa = 1.4$.

For the conventional pressure and temperature range of 200 bar and 60° C $n = 1.6$.

The accumulator size for p_0 will be:

$$V_0 = \frac{V_1}{0,85}.$$

p_0 = $0.9 \cdot p_1$ for membrane or bellow accumulators,
p_0 = $p_1 - 5$ bar for piston accumulators.

Calculation of accumulator volume for a secondary controlled winch drive

The following are known:

g = 9,81 m/s^2,
h = 20 m,
m = 5000 kg,
p_1 = 230 bar,
p_2 = 280 bar,
η_{drive} = 0,93.

As the size of the secondary unit and the operating point in the characteristic diagram (speed, swivel angle) are not yet known, the total efficiency is fixed at $\eta_t = 0.75$.

Operating pressure		bar				
N_2 temperature		50	100	200	300	400
°C	K	n values				
-23	250	1,540	1,677	1,835	1,842	1,801
·27	300	1,485	1,563	1,669	1,707	1,705
77	350	1,465	1,507	1,581	1,618	1,629

Table 7: Polytropic exponent n for nitrogen to K. R. Ruprecht

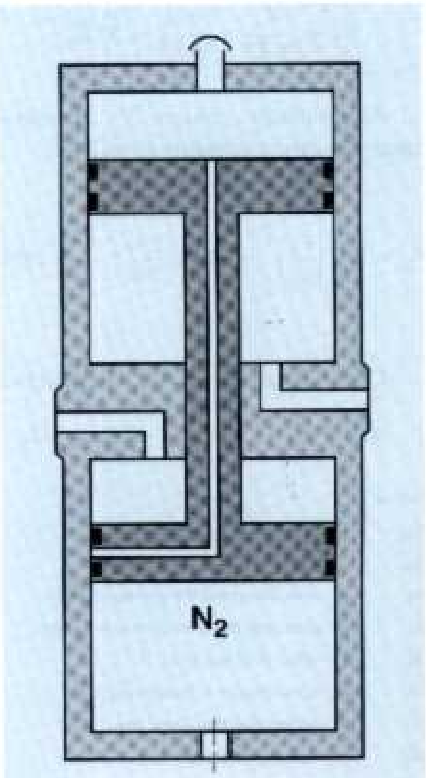


Fig. 238: Hydraulic accumulator for closed circuits

The potential energy is calculated thus:

$$E_{pot} = m \cdot g \cdot h = 5000 \cdot 9,81 \cdot 20 =$$

$$= 981 \cdot 10^3 \text{ Nm}.$$

From this we obtain the accumulator energy:

$$E_{stor} = E_{pot} \cdot \eta_t \cdot \eta_{drive},$$

$$E_{stor} = 981 \cdot 10^3 \cdot 0,75 \cdot 0,95,$$

$$E_{stor} = 699 \cdot 10^3 \text{ Nm}.$$

Accumulator volume V_1 at minimum operating pressure p_1 will be:

$$V_1 = \frac{E_{stor}(1-n)}{p_1 \cdot 10^2 \left[1 - \left(\frac{p_1}{p_2}\right)^{\frac{1-n}{n}}\right]}$$

$$V_1 = \frac{699 \cdot 10^3 \cdot (1-(1,6))}{230 \cdot 10^2 \left[1 - \left(\frac{230}{280}\right)^{\frac{1-1,6}{1,6}}\right]} = 274\text{L}.$$

$$V_0 = \frac{V_1}{0,85} = \frac{274}{0,85} = 322\text{L}.$$

Accumulator volume V_2 at maximum operating pressure p_2 will be:

$$V_2 = V_1 \cdot \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} = 274 \cdot \left(\frac{230}{280}\right)^{\frac{1}{1,6}} = 242\text{L}.$$

The volume differential will thus be:

$$\Delta V = V_1 - V_2 = 274 - 242 = 32\text{L}.$$

where:

E_{pot} = potential energy in Nm,
E_{stor} = energy to be stored in Nm,
p_1 = min. accumulator pressure in bar,
p_2 = max. accumulator pressure in bar,
V_o = volume of gas in L,
V_1 = accumulator volume at p_1 in L,
V_2 = accumulator volume at p_2 in L,
ΔV = volume difference in L,
n = polytropic exponent,
η_{drive} = drive efficiency,
η_t = total efficiency.

When operating in closed circuit this volume differential must also be generated on the boost pressure side. Due to the high costs involved the energy recovered from large flows is used preferably in open circuit.

If, however, operation has to be in closed circuit, the hydraulic accumulators shown in **Fig. 238** can be used, equal flows being moved on the high pressure and boost pressure sides.

This accumulator system was developed for use in secondary control and has been applied for the first time in a transmission circuit.

Power losses

The power losses occurring in a hydraulic system from throttle, pressure and leakage losses, are converted exclusively into heat and transferred to the operating medium. This heat then has to be removed from the circuit by taking special measures.

The procedure applied in conventional hydrostatic drives, of estimating the loss of power by means of the efficiency i.e. to set P_{loss}= 0.14 to 0.20 kW per kW installed primary power, does not work with secondary control.

There are several reasons for this:

- As several actuators are often connected together in parallel with secondary control, working as pumps or motors, and are supplied from a common primary station, this station need not be designed for the corner power of the complete system. The installed primary power is often less, sometimes considerably so, than the corner power.

The loss in power may, under certain conditions, be higher than the installed power without resulting in pressure troughs.

- The volumetric losses from a secondary unit are virtually independent of the loading, as the operating pressure remains impressed. If, for example, a unit running idly at a small swivel angle and constant speed is loaded, increasing the swivel angle, this process will have almost no effect on the losses.

Even at zero speed the lost power will remain approximately the same, unless the hydraulic isolator is switched to the blocking position. The difference occurs because the mechanical-hydraulic losses are determined by the speed.

- As there are variable displacement units on both primary and secondary sides, the control power only needs to be considered with high response drive systems.

The flow resulting from the volumetric losses, which, according to the calculation of the cooling system, is available, is virtually constant. When calculating the total power losses of a secondary controlled drive system, the losses of each individual unit must be taken into account, irrespective of whether the unit is working as a pump or motor.

The following formula can be applied to give an approximation:

$$P_{loss\ i} = \frac{V_{gmax} \cdot n_{max} \cdot \Delta p}{6 \cdot 10^5} \cdot (1 - \eta_t).$$

where:

i = 1, 2, 3, ...,
n_{max} = maximum speed in rpm,
$P_{loss\ i}$ = power loss of a unit in kW,
Δp = operating pressure differential in bar,
V_{gmax} = maximum displacement in cm^3 irrespective of swivel angle,
η_t = $\eta_{vol} \cdot \eta_{mh}$ = total efficiency

With operating ranges for sizes of up to 500 cm^3 and

- Δp = 200 to 300 bar and

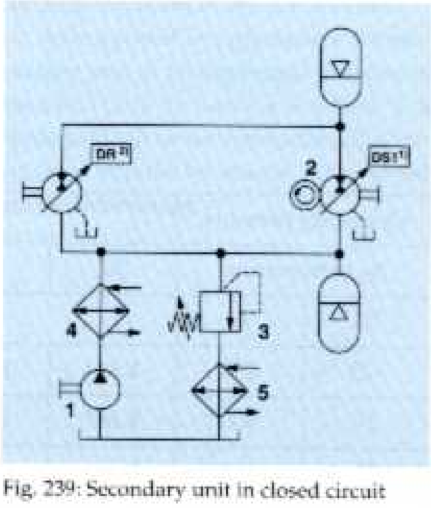


Fig. 239: Secondary unit in closed circuit

- n_{max} = 1000 to 2000 rpm

for ηt = 0.88 to 0.91 can be used (see also **Fig. 126** on page 110).

To be on the safe side the cooler should be designed using this method of calculation.

As with the conventional hydraulic drive, with secondary control the boost pump has two functions to fulfil in closed circuit operation (**Fig. 239**):

- The external leakage losses of the hydrostatic units must be replaced in order to maintain the low pressure.
- The boost pump must also generate a flow for the heat exchanger.

The heat exchanger can be installed either in the pressure line (4) or in the return line (5).

Cooling in the pressure line, which is seldom used, has the advantage that the total flow will be sent to the boost pump via the heat exchanger, although this must be maintained at a constant pressure of 25 bar. If cooling is carried out in the return line, pressure loading of the heat exchanger will be minimal. As external leakage from the hydrostatic units will remain constant irrespective of loading, these flows can also be directed via the cooler. A separate cooler / filter circuit is not normally necessary. The application of secondary controlled units on a back tensioning test stand is subject to special conditions (**Fig. 240**).

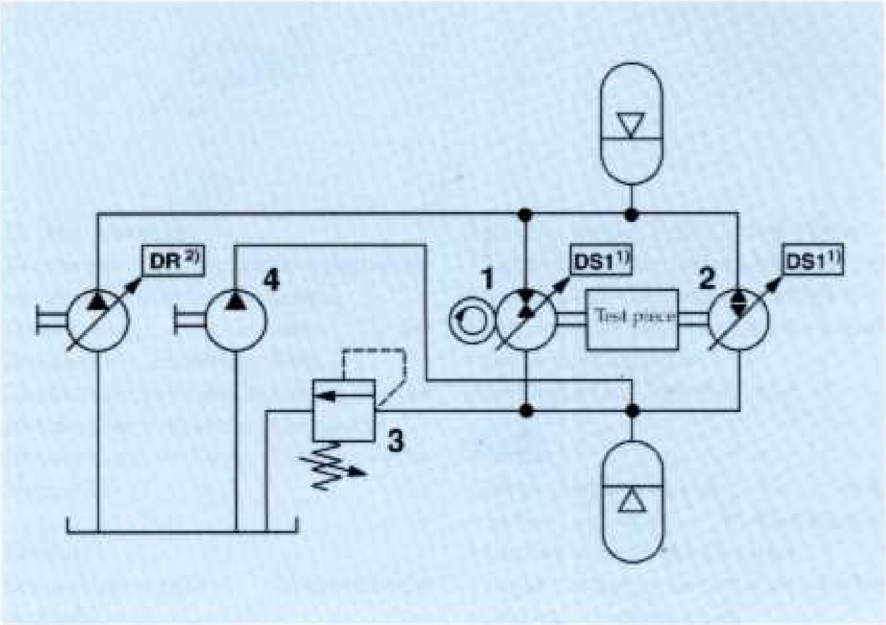


Fig. 240: Back tensioning test stand in closed circuit

Although the driving motor (1) has a greater flow requirement due to its higher losses than the driven machine (2), even with different sizes and thus increasing surplus flow pre-tensioning of the low pressure line via a pressure relief valve is not necessarily sufficient to dispense with the boost pump in the lower part of the speed range.

This boost pump (4) can, however, be kept small.

Short-term energy peaks must again be covered by an accumulator system, which must be installed near the actuators. Care must be taken, however, to ensure that in an emergency the flow requirement of the driven machine (2) is greater than that flowing to the hydraulic motor (1).

Bibliography

Prof. Dr.-Ing. Nikolaus:
Hydrostatischer Fahr- und Windenantrieb mit Energierückgewinung
Hydrostatic vehicle and winch drives with energy recovery
o+p 25 (1981) Nr. 3

Kordak:
Neuartige Antriebskonzeptionen mit sekundärgeregelten hydrostatischen Maschinen
New types of drive concepts using hydrostatic machines with secondary control
o+p 25 (1981) Nr. 5

Prof. Dr.-Ing. Nikolaus:
Dynamik sekundärgereglter Hydroeinheiten am eingeprägten Drucknetz
Dynamic secondary controlled hydrostatic units on ring mains with impressed pressure
o+p 26 (1982) Nr. 2

Kordak:
Praktische Auslegung sekundärgeregelter Antriebssysteme.
Hydraulik-Kolloquium, 13. Mai 1982.
Hochschule der Bundeswehr, Hamburg, Tagungsunterlagen
The practical design of secondary control systems
Hydraulik Symposium, 13. May 1982.
Military College, Hamburg (Seminar notes)

Kordak:
Einsatzbeispiele sekundärgereglter Antriebe für hochdynamische Vorgänge mit der Möglichkeit der Energierückgewinnung.
Fachbeitrag zum 2. Hamburger Hydraulik-Kolloquium, April 1985
Application examples of drives with secondary control for high response processes with the possibility of energy recovery
Presentation notes for the 2nd Hamburg Symposium, April 1985

Dr.-Ing. Metzner:
Kennwerte der Dynamik sekundärgeregelter Axialkolbeneinheiten.
Dissertation, Universität der Bundeswehr, Hamburg, 1985.
Characteristics of the dynamics of axial piston units with secondary control.
Dissertation, Military College, Hamburg, 1985

Kordak:
Sekundärgeregelte hydrostatische Antriebe
Hydrostatic drives with secondary control
o+p 29 (1985) Nr. 9

Kern, Reyer, Schober, Weiger:
Hochdynamischer, hydraulischer Motorenprüfstand.
High response hydraulic engine test stand
o+p 29 (1985) Nr. 4

Mannesmann Rexroth GmbH:
The Hydraulic Trainer, Volume 6, 1st edition
RE 00293/08.89

Dr.-Ing. Lindemann, Kugler:
Prüfsysteme mit Leistungsrückspeisung und Verspannung – Beispiele für Hydrostatiklösungen.
VDI-Berichte 681
Test systems with power feedback and back tensioning - examples of hydrostatic solutions
VDI Report 681

Weiblein, Kötzle, Schober, Schaub:
Radführungsprüfstand zur Simulation der thermisch-mechanischen Belastungen am Achskopf einer angetriebenen PKW-Achse.
VDI-Berichte 681
Axle test stand for the simulation of thermal-mechanical loading at the axle head of a driven car axle
VDI Report 681

Folckert, Menne, Prof. Dr.-Ing. Waller:
Hochdynamischer Versuchsstand zur Identifikation von Antriebssträngen.
High response test stand for the identification of drive paths
Antriebstechnik 33 (1993) Nr. 2

Kordak:
Drehmomentsteuerung bei elektrischen und hydrostatischen Maschinen mit hoher Dynamik
Torque control in electrical and hydrostatic high response units
o+p 38 (1994) Nr. 1 und 2

Prof. Dr.-Ing. Backé:
Servohydraulik
Umdruck zur Vorlesung
Servo hydraulics
Lecture paper

Fa. Siemens:
Drehzahlveränderbare Antriebe in der Praxis
Speed-variable drives in practice

Prof. Dr.-Ing. Leonhard:
Regelung in der elektrischen Antriebstechnik
Closed loop control in electrical drive technology
Teubner Studienbücher

Flegel, Prof. Dr.-Ing. Birnstiel
Elektrotechnik für den Maschinenbauer
Electrotechnology for the mechanical engineer
Carl Hanser Verlag, München/Wien

Kordak:
Der sekundärgeregelte hydrostatische Antrieb in mobilen Arbeitsgeräten.
Secondary controlled hydrostatic drives in mobile machinery
o+p 29 (1995) Nr. 12

Beyer, Illian, Liebler:
Sekundärgeregelter Fahrantrieb für automatisiertes Transportsystem CT40
RIQ Ausgabe 4 / 1991
Secondary controlled drives for CT40 automated transport system
RIQ, No. 4 / 1991

Dissertationsschriften am Institut für fluidtechnische Antriebe und Steuerungen der Rheinisch-Westfälischen Technischen Hochschule Aachen zum Thema Sekundärregelung
Dissertations on the theme of secondary control at the Institute for Fluid Technical Drives and Controls of the Rhineland-Westphalian Technical University, Aachen

Murrenhoff:
Regelung von verstellbaren Verdrängereinheiten am Konstant-Drucknetz
The closed loop control of variable displacement units in a constant pressure system
RWTH Aachen, 1983

Haas:
Sekundärgeregelte hydrostatische Antriebe im Drehzahl- und Drehwinkelregelkreis
Secondary controlled hydrostatic drives in speed and swivel angle closed circuits
RWTH Aachen, 1989

Weishaupt:
Adaptive Regelungskonzepte für eine hydraulische Verstelleinheit am Netz mit aufgeprägtem Versorgungsdruck im Drehzahl- und Drehwinkelregelkreis
Adaptive closed loop control concepts for a hydraulic displacement unit in a mains circuit with impressed supply pressure in speed and swivel angle closed circuits
RWTH Aachen, 1995

Kögl:
Drehmomentenregelkreis am Netz mit angepaßtem Versorgungsdruck
Torque closed control circuit in a mains circuit with matched supply pressure
RWTH Aachen, 1995

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