3 Comparison of various types of speed controlled motors

Diagram 1 shows a comparison of various types of controllable electrical motors and hydrostatic units under secondary control with special reference to the maximum possible rate of change of speed per second as a function of the corner power.

The criteria for this evaluation was taken from the manufacturers literature.

The rate of speed change at present obtainable is of the order of 40 000 rpm/s. The technical possibilities are thus not exhausted. Development in the next few years must be concerned with the question of how curve B1 can be pushed towards the theoretical limit of curve A.

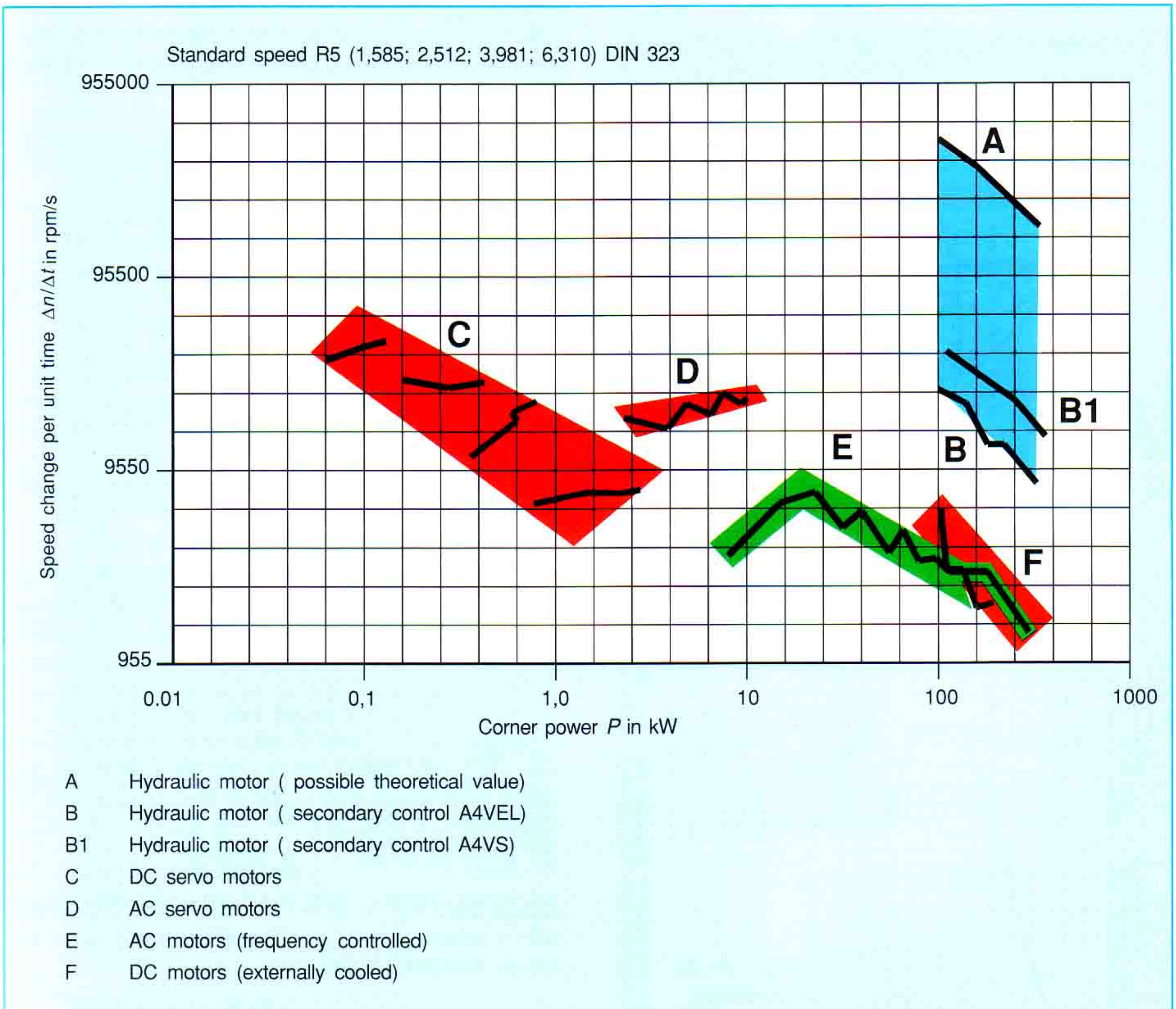

Diagram 1: *Maximum rate of change of speed of various motors as a function of the corner power.*

The actual values at present achievable with secondary control are shown in the ranges B/B1 (on a logarithmic scale).

To achieve this, a number of different steps are possible:

1. The production of a special rotat ing group designed for this system with the object of reducing the control times of the units.
2. An improvement of the electronic signal processing in the tachometer, swivel angle feedback and at the servo valve.
3. The development of advanced digital closed loop control concepts with specially adapted control algorithms for secondary control. The aim of this development must be an adaptive digital controller which recognizes changes in parameters and automatically optimizes itself to suit these changes.

These changes can only be made by changing over to digital signal processing. The advantage of digital closed loop control lies in the fact that the control algorithm can be processed by means of a microprocessor. A further advantage is the high resolution of the determination of rotary angle which is made possible as it is only dependent on the pulse rate of the generator per revolution. Furthermore, great flexibility is then available in the implementation of complex and new versions of closed loop controls and also the easy access to changes and adaptations of existing algorithms by means of changing the programmes.

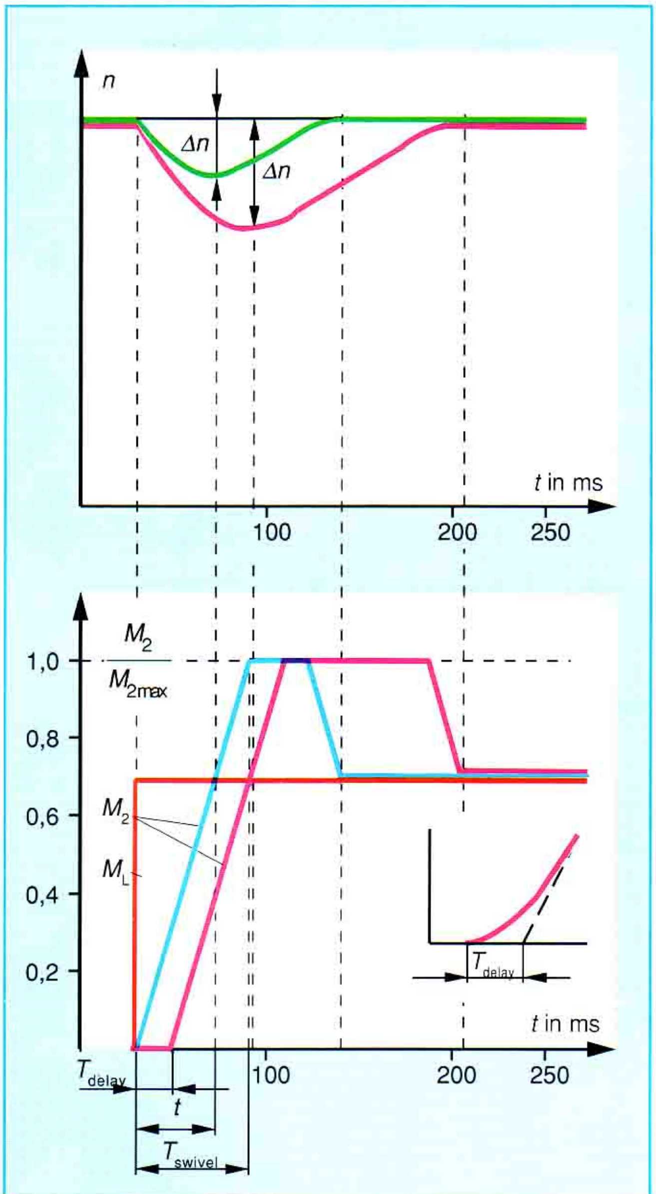

Diagram 2: *Minimum magnitude of maximum speed variation under a stepped load change.*

The area of application of secondary controls is not, however, solely dependent on the electronic control but it also primarily dependent on the control times of the units (definition: from 0 ° to α_{max}).

Diagram 2 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 80% of the maximum possible torque.

The derivation of the formula for Δn is based on the following mathematical relationships:

$$t = X \cdot T_{swivel} + T_{delay}$$

$$\Delta \dot{n} = \frac{M_L(t) - M_2(t)}{2 \cdot \pi \cdot J_g}$$

$$\Delta n = \int_0^t \frac{M_L(t) - M_2(t)}{2 \cdot \pi \cdot J_g} \cdot dt$$

$$X = \frac{M_L}{M_{2max}} \quad \text{(torque relationship)}$$

As soon as the change in load torque is recognized, and without a servo valve in the control, the unit swivels immediately towards maximum displacement dependent only upon the swivel time (T_{swivel}). The hydraulic torque, which is dependent on the swivel angle α, is smaller than the load of 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 delayed time T_{delay}. This is dependent upon the natural frequency of the valve.

$$T_{delay} \approx \frac{1}{2 \cdot \pi \cdot f_E}$$

Due to this, the speed variation is 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 can be calculated as follows:

– Without the time delay of the servo valve

$$\Delta n = \frac{3\,X^2 \cdot \Delta p \cdot V_{2\,max}}{4 \cdot \pi^2 \cdot J_g} \cdot T_{swivel} \quad \text{in rpm}$$

– With the delay of the servo valve.

$$\Delta n = \frac{3\,X^2 \cdot \Delta p \cdot V_{2\,max}}{4 \cdot \pi^2 \cdot J_g} \cdot T_{swivel} \cdot \left(1 + \frac{2 \cdot T_{delay}}{X \cdot T_{swivel}}\right) \quad \text{in rpm}$$

$V_{2\,max}$	Displacement	in cm^3
Δp	Operating pressure	in bar
I_{total}	Reflected moment of inertia	in kgm^2
T_{swivel}	Control time	in s
T_{delay}	Time delay	in s
f_{nat}	Natural frequency of servo valve	in Hz
$M_{2\,max}$	Maximum torque of secondary unit	in Nm
M_L	Load torque	in Nm

As can be seen from the formula for Δ*n*, 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. The electric motor is indeed capable of increasing the torque within the air gap of the motor within 15-20 ms. Unfortunately, this has no influence on the dynamic response of the unit due to its high internal moment of inertia.

Step functions such as we have shown here in *diagram 2* for the load change are well favoured in simulation calculations as they can be clearly defined. In practice, however, they do not exist, as the acceleration must then be infinitely great. Practical conditions are therefore more as we have illustrated in *diagram 3* where the load torque rises along a ramp. 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. 2*.

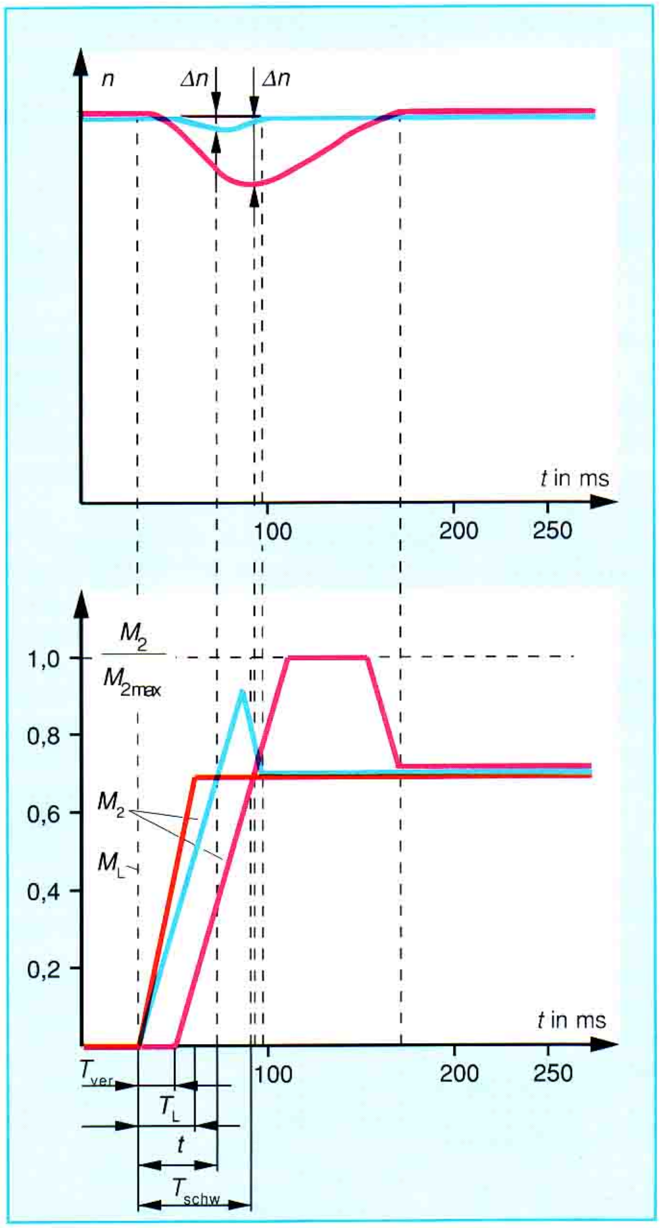

Diagram 3: *Speed variation with a ramped load change.*

Speed variation without the time delay of the servo valve.

$$\Delta n = \frac{3 \cdot X^2 \cdot \Delta p \cdot V_{2\,max}^2}{4 \cdot \pi^2 \cdot J_g} \cdot T_{swivel} \left(1 - \frac{T_L}{X \cdot T_{swivel}}\right) \text{ in rpm}$$

Speed variation with time delay in servo valve.

$$\Delta n = \frac{3 \cdot X^2 \cdot \Delta p \cdot V_{2\,max}^2}{4 \cdot \pi^2 \cdot J_g} \cdot T_{swivel} \left(1 - \frac{T_L}{X \cdot T_{swivel}} + \frac{2 \cdot T_{delay}}{X \cdot T_{swivel}}\right) \text{ in rpm}$$

$V_{2\,max}$	Displacement	in cm^3
Δp	Operating pressure	in bar
I_{total}	Reflected moment of inertia	in kgm^2
T_{swivel}	Control time	in s
T_L	ramp time	in s
T_{delay}	Time delay	in s

Utilising the above formula for speed variation Δn, a numerical example has been calculated and the results shown in *table 2*. In this case, an axial piston unit type A4VSO 250DS has been installed and a torque variation of X = 70% considered. Columns 1 & 2 cover a load step without and with time delay. Columns 3 & 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 rotary group. On stability grounds, this is the value specified in publication API 160.

The time delay is:

$$T_{delay} \approx \frac{1}{2 \cdot \pi\, f_E} = \frac{1}{2 \cdot \pi \cdot 45} = 3{,}54 \text{ ms}$$

The results calculated in column 2 have been compared with a simulation calculation in the bottom two lines of the table. The two values agree closely considering that the simulation calculation embodied a realistic gain in the speed control loop in order to achieve stability and also the starting friction was included in the calculation process.

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 virtue of better design) has a greater influence on speed accuracy than any improvement in the mechanical area.

	1	2	3	4	
T_{swivel}	60	60	60	60	ms
T_L	—	—	40	40	ms
T_{delay}	—	3,54	—	3,54	ms
$J_g = 10 \cdot J_{nat}$	1	1	1	1	kgm^2
Δp	250	250	250	250	bar
M_{max}	995	995	995	995	Nm
X	0,7	0,7	0,7	0,7	%
q	250	250	250	250	cm^3
Δn	140	163	7	30	rpm
Δn (Simulation)		184			rpm

Table 2: *Sample calculation for Δn from diagrams 2 and 3*

J_{nat} = internal moment of inertia of secondary unit.

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 the control time T_{swivel}. The dynamics of the overall control loop and the regulator transfer characteristics are also important.

In order to assist the reader to assess the influence of these values on the dynamics of the drive and the weighting of the parameters on the characteristics of the system, *diagram 4* shows the graphic relationship between them.

The dynamics of the system under control are defined by the **time factor of the control loop** T_R as follows:

$$T_R = 2 \cdot \pi \cdot \sqrt{\frac{\pi \cdot J_{tot}}{\frac{p \cdot V_{tot}}{10} - 2\pi\,(M_L + M_R)}} \quad \text{in s}$$

J_{tot}	Moment of inertia reflected onto the shaft of the axial piston unit	in kgm^2
p	Operating pressure	in bar
V_{tot}	Displacement	in cm^3
M_L	Load torque	in Nm
M_R	Frictional torque	in Nm

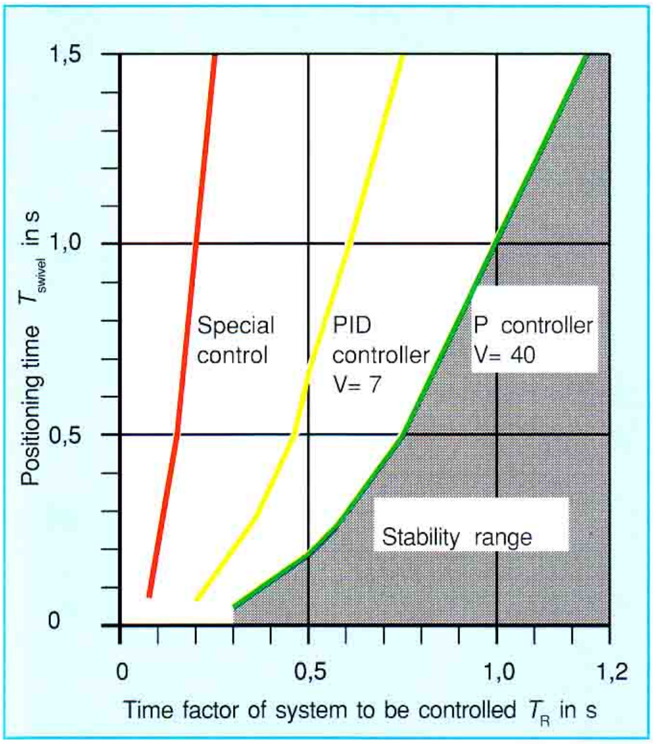

Diagram 4: Stability curves for closed loop control design

Time factor T_R is proportional to the moment of inertia and is a parameter of the drive which eliminates the influence of the size factor, i.e. the dynamics of a complete series of units may be determined from this single stability diagram *(diagram 4)*.

The area below the curves forms the stability zone. Above the curves instability can be expected.

The reader 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 value of control time T_{swivel} is greater than the curve for the pre-determined limiting positioning time, the secondary speed control of the given series operating under conditions of imposed pressure can only achieve the pre-selected command speed after a period of oscillation (a settling period). 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 has a certain amount of dynamic reserve. The value of this reserve depends upon the value of the difference between the two control times.

When designing a system, it must be noted, that the drive must lie 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 increases the stability of the drive.

It is also possible to see from the curves that if a better regulator with PID transfer characteristics is employed 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 for 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 come into play where, for example, the reflected inertia is reduced due to the square of the gear reduction coming into play.

With a view to control times, all problems are also reduced, so that in conclusion it can almost be stated that:

**The control time can be as short as possible.
It is always too long.**