韓國精密工學會 '94年度 秋季學術大會論文集 pp.286～292

An Analytical Study on the Performance Improvement for Both Electric and Elecro-hydraulic Type High Power Drive System through Damping Factor

Dae Ok Lee, Tai Young Ahn
(Agency for Defense Development, Korea)
Landers Phillip
(Dundee College, England)

Abstract

The classical way to improve the control performance is studied on the aspects of gun/turret drive systems. Two ways are discussed compara ively ; electrical case and electro-hydraulic case. System parameters are analytically studied in terms of resonance frequencies, and damping and gear train ratio effects are checked in relation to resonance frequency increase. Benefit of the feedback is discussed to increase the damping of the natural frequency leading to bandwidth increase.

Introduction

This paper covers the servo control basics on both electric and electro-hydraulic drive systems. Though development work is based on the already well established principles and equations, the step by step writing is quite inquiry of physical interpretations. Working is largely consisted of two parts. The first part describes the physical properties and the mathematical models which are characterizing the electric and electro-hydraulic servo control systems comparatively. The second part is rather implementation oriented toward a typical plant. Electric and electro-hydraulic basic equations are derived and dynamic characteristics are interpreted using typical traverse axis parameters emphasizing natural frequencies for both cases. Gear train ratio effects leading to the worst and the best are studied, which are apparent to both cases. Transfer function are derived for both cases in case of motor, gear train, load condition combination. Damping is noted to improve the performance firstly with the system parameters, then feedbacks are considered and evaluated for the engineering solution.

Basic Equations

Electric case

$$V = L\frac{dI}{dt} + RI + K_m\omega_m$$

$$T_m = K_m I$$

$$T_l = T_m - J_m\frac{d\omega_m}{dt} - D_m\omega_m$$

$$T_L = \zeta N T_l \quad and \quad \omega_2 = \frac{\omega_m}{N}$$

$$T_L = K_S\int(\omega_2 - \omega_L)dt$$

$$T_L = J_L\frac{d\omega_L}{dt} + D_L\omega_L$$

Hydraulic case

$$Q = \frac{V}{B}\frac{dP}{dt} + LP + d_m\omega_m$$

$$T_m = d_m P$$

$$T_l = T_m - J_m\frac{d\omega_m}{dt} - D_m\omega_m$$

$$T_L = \zeta N T_l \quad and \quad \omega_2 = \frac{\omega_m}{N}$$

$$T_L = K_S\int(\omega_2 - \omega_L)dt$$

$$T_L = J_L\frac{d\omega_L}{dt} + D_L\omega_L$$

These equations are identical except for the first two, and even in the case of the first two equations the structure is the same. However the parameter definitely are not. In the electrical case $L/R = 0.0015$ seconds and $K_m^2/J_mR \cong 10$, whilst in the hydraulic case $\frac{V}{B}/L \cong 0.01$ and $d_m^2/J_mL \cong 3\times10^4$. Notice motor torque constant is equal to emf constant. More simply, in the hydraulic case J_m is negligible whilst in the electric case L is negligible as we shall see this results in very different dynamics.

where	V	the motor input voltage and the fluid volume respectively
	L	the motor armature inductance and the flow-pressure gain
	R	the motor armature resistance
	I	the motor armature current
	K_m	the motor torque or e.m.f. constant
	ω_m, ω_L	the motor speed and the load speed respectively
	B	the bulk modulus of fluid
	P	the hydraulic pressure
	d_m	the volumetric displacement of hydraulic motor
	T_m, T_i, T_L	the torque developed by the motor, the torque at the gearbox input shaft and the torque applied to the load respectively
	J_m, J_L	the motor inertia and the load inertia respectively
	D_m, D_L	the motor friction coefficient and the load friction coefficient respectively
	ζ	the gearbox efficiency
	N	the gearbox ratio
	K_s	the stiffness of the output shaft

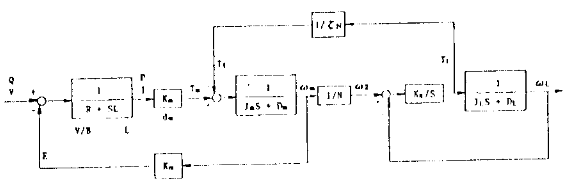

Fig. 1 Electrical servo System Block Diagram

Table 1. Electrical and hydraulic Characteristics for Traverse Direction only

Electric	*Hydraulic*
$R = 1\Omega$	$L = 0.032 \times 10^{-10} m^5 / Ns$
$L = 0.0015H$	$V / B = 3.33 \times 10^{-14} m^5 / N$
$K_m = 0.25$	$d_m = 13 \times 10^{-6} m^3 / rad$
$J_m = 0.007 kg - m^2$	$J_m = 0.0017 kg - m^2$
$D_m = 0.003$	$D_m = .003$
$N = 800$	$N = 1000$
$\zeta = 0.7$	$\zeta = 0.7$
$K_s = 12 \times 10^6$	$K_s = 12 \times 10^6$
$J_L = 20 \times 10^3 kg - m^2$	$J_L = 20 \times 10^3 kg - m^2$
$D_L = 5500$	$D_L = 5500$
$T_a = 0.002 sec$	$T_v = 0.002 sec$

Transfer Functions

The basic system is fourth order. This is because we have a transfer function relating angular velocity to either terminal voltage(in the electrical case), or flow rate(in the hydraulic case). Such a system would be second order, it having two energy storage elements, there being inertia and either armature inductance(in the electrical case) or oil compressibility(in the hydraulic case). However the finite stiffness of the shaft raises the order of any system by two, so the basic system is fourth order. If we considered angular displacement instead of angular velocity then the order would be raised by unity, whilst if we include power amplifier dynamics the order would be raised still further.

Consider for now angular velocity versus terminal voltage. We then have

$$\frac{\omega_L}{V} = \frac{K}{as^4 + bs^3 + cs^2 + ds + e}$$

The coefficients are tabulated on Table 2.

Table 2. Coefficients of the Generalized Transfer Function for Electric and Hydraulic Cases

Order	*Electric*	*Hydraulic*
s^4	$\zeta N^2 J_L J_m L$	$\zeta N^2 J_L J_m V / B$
s^3	$\zeta N^2 J_L J_m R$	$\zeta N^2 J_L J_m L$
s^3	$\zeta N^2 J_m D_L L$	$\zeta N^2 J_m D_L V / B$
s^3	$\zeta N^2 J_L D_m L$	$\zeta N^2 J_L D_m V / B$
s^2	$J_L K_s L$	$J_L K_s V / B$
s^2	$\zeta N^2 J_L K_m^2$	$\zeta N^2 J_L D_m^2$
s^2	$\zeta N^2 J_m D_L R$	$\zeta N^2 J_m D_L L$
s^2	$\zeta N^2 J_L D_m R$	$\zeta N^2 J_L D_m L$
s^2	$\zeta N^2 J_m K_s L$	$\zeta N^2 J_m K_s V / B$
s^2	$\zeta N^2 D_L D_m L$	$\zeta N^2 D_L D_m V / B$
s^1	$D_L K_s L$	$D_L K_s V / B$
s^1	$J_L K_s R$	$J_L K_s L$
s^1	$\zeta N^2 D_L K_m^2$	$\zeta N^2 D_L d_m^2$
s^1	$\zeta N^2 J_m K_s R$	$\zeta N^2 J_m K_s L$
s^1	$\zeta N^2 D_L D_m R$	$\zeta N^2 D_L D_m L$
s^1	$\zeta N^2 D_m K_s L$	$\zeta N^2 D_m K_s V / B$

The numerator is

$$K_s K_m \zeta N = 1680 \times 10^{12}, \quad K_s d_m \zeta N = 101.4 \times 10^3$$

The difference in the numerical value of the terms K is striking but not important. For instance, consider the 'dc' term

$$\frac{\omega_L}{V}\Big|_{ss} = \frac{K_s K_m \zeta N}{K_s K_m^2 \zeta N^2} = \frac{1}{K_m N} = 0.005$$

$$\frac{\omega_L}{Q}\Big|_{ss} = \frac{K_s d_m \zeta N}{K_s d_m^2 \zeta N^2} = \frac{1}{d_m N} = 76.9$$

But we are not comparing like with like. Unity terminal voltage would be regarded as quite a small voltage to apply to an electric motor, whilst a flow rate of $1 in^3/sec$ would be an enormous input to a hydraulic motor.

Consider now the possible causes of resonances, first from an intuitive point of view.

In the case of the electrical system we have two inertias (J_L and J_m) and the shaft stiffness. We disregarded the armature inductance (L) partly because we suspect that we can, and partly because we know that if a printed circuit motor were used the inductance would be even less. We would then have

$$\omega_n^2 = \frac{K_s}{\dfrac{J_L \zeta N^2 J_m}{J_L + \zeta N^2 J_m}} = 4426.5$$

Therefore $\omega_n = 66.5\, rad/sec, \quad f_n = 10.6\, Hz$

For the hydraulic case we have the load inertia and two stiffnesses. The motor inertia is regarded as negligible. One of the stiffnesses is the shaft stiffness, whilst the other is V/B, the compressed volume divided by the bulk modulus of the working fluid. The stiffness resulting from non-zero V/B is

$$\frac{d_m \zeta N^2}{V/B} = d_m \zeta N \frac{B}{V} = 33 \times 10^8 = K'$$

$$K_s = 12 \times 10^6$$

Therefore

$$\frac{K_s K'}{K_s + K'} = 11.69 \times 10^6$$

$$\omega_n^2 = \frac{K_s K'}{K_s + K'} \frac{1}{J_L} = 598$$

$$\omega_n = 24.45\, rad/sec, \quad f_n = 3.89\, Hz$$

It is reassuring to note that resonant frequencies of approximately 10Hz and 4Hz are just what occurs in practice for the electric and hydraulic traverse system respectively. As it is desirable to make the resonant frequency of a servo as high as possible, so that the gain of the system is low at the resonant frequency, the electrical servo appear to have a distinct advantage. Further, the mechanical resonance given by

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{K_s}{J_L}}$$

which is common to both appear to be the worst case in the electrical system whilst being the best case in the hydraulic system.

Then $\dfrac{K_s K'}{K_s + K'} \cong \dfrac{K_s K'}{K'} = K_s$

and $f_n = \dfrac{1}{2\pi}\sqrt{\dfrac{K_s}{J_L}} = 3.898\, Hz$

We see that the hydraulic system almost attains its best case but as this is significantly worse than the electric system, this observation is not much consolation.

A further point to be made here is that we can at least hope to raise the electrical system's resonant frequency still further by reducing the gearbox ratios (N). Whilst, of course, suffering a concomitant (occurring concurrently) increase in the inertia of the motor rotor J_m. We have

$$\zeta N T_m = J \dot{\omega}_L = (J_L + \zeta N^2 J_m)\dot{\omega}_L$$

$$N\downarrow \Rightarrow T_m \uparrow \quad \textit{but not by so much}$$

$$T_m = K_m I \overset{ss}{=} \frac{P\Phi Z}{2\pi a}\frac{V}{R}$$

For a lap winding a=P, whilst for a wave winding a=1. Consider the lap winding for simplicity.

$$T_m \overset{ss}{=} \frac{\Phi Z}{2\pi}\frac{V}{R}$$

V is limited by the supply capacity.

$$T_m \propto \Phi Z = \beta \times \textit{pole area} \times Z$$

B is limited by type of magnetic material.

Therefore $T_m \propto \textit{pole area} \times \dfrac{Z}{R}$

Increase pole area $\rightarrow$ increase in length or circumference. But length increase best as Jm increase very quickly with circumference.

Therefore $T_m \propto \textit{length of rotor}$

Now $J_m = Mk^2 = Pd \times vol \times k^2 = Pd \times \pi r^2 l \times k^2$

Therefore $J_m \propto \textit{length of rotor}$ and $\dot{\omega}_L = \dfrac{\zeta N J_m}{J_L + \zeta N^2 J_m}$

If the rotor length is variable

$$\dot{\omega}_L = \frac{\zeta(\frac{N}{m}) m T_m}{J_L + \zeta(\frac{N}{m})^2 m J_m}$$

where in above case m=1.

As length increases over variable m increase.

As $m \rightarrow \infty \quad \dot{\omega}_L \rightarrow \lim_{m\rightarrow\infty} \dfrac{\zeta N T_m}{J_L + \zeta N^2 \frac{J_m}{m}} = \dfrac{\zeta N T_m}{J_L}$

Clearly $\dot{\omega}_L$ actually become greater as we reduce the gearbox ratio. The limitations to this process are two. Firstly we will have the problem of designing and installing a very big size motor. Secondly as we reduce the gearbox ratio, the motor speed, for a given load speed, becomes less. Motors do not perform well at low speed(cogging). Nevertheless, the designer does have some latitude regarding the resonant frequency of the electrical system. There is no such latitude(extent) with its hydraulic counter part. Reducing N to 400, and thus doubling J_m (to 0.014 $kg-m^2$) results is

$$\frac{J_L \zeta N^2 J_m}{J_L + \zeta N^2 J_m} = 1454$$

Therefore $\omega_n^2 = 8253$, $\omega_n = 90.846\, rad/sec$, $f_n = 14.46\, Hz$

This is a quite significant increase in the resonant frequency.

Let us now look more carefully at the resonances by analyzing the terms in the transfer function, already given, for the motor, gearbox, load combination. Consider the electric case. We have

$$\frac{K_s K_m \zeta N}{\zeta N^2 J_L J_m R s^3 + \zeta N^2 J_L K_m^2 s^2 + (J_L K_s R + \zeta N^2 J_m K_s R)s + \zeta N^2 K_s K_m^2}$$

if we let the amature inductance(L) equal zero. Considering just the s^2, s^1, s^0 terms we have

$$s^2 + \frac{K_s R(J_L + \zeta N^2 J_m)s}{\zeta N^2 J_L K_m^2} + \frac{\zeta N^2 K_s K_m^2}{\zeta N^2 J_L K_m^2}$$

Clearly $\omega_n^2 = \frac{K_s}{J_L}$ and $2\zeta\omega_n = \frac{K_s R(J_L + \zeta N^2 J_m)}{\zeta N^2 J_L K_m^2}$

Therefore $\omega_n = \sqrt{\frac{K_s}{J_L}}$ and $\zeta = \frac{1}{2}\sqrt{\frac{J_L}{K_s}}\frac{K_s R(J_L + \zeta N^2 J_m)}{\zeta N^2 J_L K_m^2}$

Finally $\zeta = \frac{1}{2}\frac{R(J_L + \zeta N^2 J_m)}{\zeta N^2 K_m^2}\sqrt{\frac{K_s}{J_L}}$

As before $f_n = 3.89\, Hz$, but it is known that in an electrical servo this resonance does not appear since $\zeta = 27.47$. With this damping factor it is hardly surprising that this resonance is not apparent. Consider now the s^3, s^2, s^1 terms. Then we have

$$s^2 + \frac{\zeta N^2 J_L K_m^2}{\zeta N^2 J_L J_m R}s + \frac{K_s R(J_L + \zeta N^2 J_m)}{\zeta N^2 J_L J_m R} \quad or \quad s^2 + \frac{K_m^2}{J_m R}s + \frac{K_s(J_L + \zeta N^2 J_m)}{\zeta N^2 J_L J_m}$$

Therefore $\omega_n^2 = \frac{K_s(J_L + \zeta N^2 J_m)}{\zeta N^2 J_L J_m} = K_s \Big/ \frac{\zeta N^2 J_L K_m^2}{(J_L + \zeta N^2 J_m)}$

This is exactly the resonance already inferred intuitively. But we can now calculate the damping ratio.

$$2\zeta\omega_n = \frac{K_m^2}{J_m R}, \quad and \quad \zeta = \frac{K_m^2}{2 J_m R}\frac{1}{\omega_n}$$

$$\omega_n = 66.5\, rad/sec, \quad and \quad \zeta = 0.067$$

The damping factor is very low and this is just what we would expect for a resonance which is, in practice, very noticeable.

Now let us consider the hydraulic case. Here we have

$$\frac{K_s d_m \zeta N}{(J_L K_s \frac{V}{B} + \zeta N^2 J_L d_m^2)s^2 + J_L K_s L s + \zeta N^2 K_s d_m^2}$$

Allowing $J_m = 0$, then we have

$$s^2 + \frac{J_L K_s L}{(J_L K_s \frac{V}{B} + \zeta N^2 J_L d_m^2)}s + \frac{\zeta N^2 K_s d_m^2}{(J_L K_s \frac{V}{B} + \zeta N^2 J_L d_m^2)}$$

$$and \quad \omega_n^2 = \frac{\zeta N^2 K_s d_m^2}{J_L K_s \frac{V}{B} + \zeta N^2 J_L d_m^2} = \frac{d_m^2 \zeta N^2 K_s}{K_s V + d_m^2 B \zeta N^2} / J_L$$

$$and \quad \omega_n^2 = \frac{d_m^2 \zeta N^2 \frac{B}{V} K_s}{d_m^2 \zeta N^2 \frac{B}{V} + K_s} / J_L$$

This is the resonance already obtained from intuitive ideas and it has already been seen to comply with practice. Again however, we can now calculate the damping factor.

$$2\zeta\omega_n = \frac{K_s R(J_L + \zeta N^2 J_m)}{J_L K_s \frac{V}{B} + \zeta N^2 J_L d_m^2} = \frac{0.768}{2.205}$$

$$Then \quad \zeta = 0.0071$$

Again ζ is so low that we would expect to observe this resonance. Notice that ζ is almost 10 times less than for the dominant resonance in the electrical case. Again the electrical system appears to be better, although it should be admitted that, in either case, we expect these resonance to cause problems.

What can be done to raise the damping factor in either case?

Let us consider motor torque feedback. In the hydraulic case we can use differential pressure feedback instead as torque is very difficult to sense satisfactorily. Ignoring the valve dynamics we have the valve flow in Fig. 2.

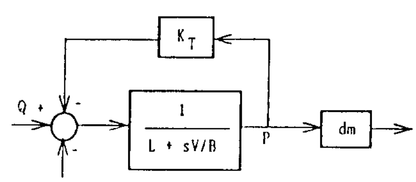

Fig. 2 Valve Flow Ignoring Dynamics

The closed loop gain

$$\frac{1/(L + s\frac{V}{B})}{1 + K_T/(L + s\frac{V}{B})} = \frac{1}{L + s\frac{V}{B} + K_T}$$

$L \rightarrow L + K_T$

As increase in the apparent value of the motor losses(L) has no effect on ω_n, but ζ is directly proportional to L and is indeed very sensitive to L.

$$2\zeta\omega_n \cong \frac{J_L K_s L}{\zeta N^2 J_L d_m^2} = \frac{K_s L}{\zeta N^2 d_m^2}$$

Then $\zeta = 2.23 \times 10^9 L$

In this case $\zeta = 2.23 \times 10^9 (L + K_T)$. Clearly ζ can be raised almost arbitrarily. It should however be remembered that feedback is applied via the valve which is a saturating device. When the valve is saturated, is under large signal conditions, feedback is ineffective and the resonance is, once more, underdamped.

Consider now the electrical case.

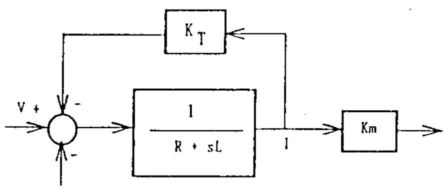

Fig. 3 Electric Motor Ignoring Dynamics

This time we sense armature current instead of torque. We then have

$$\frac{1/(R+sL)}{1 + K_T/(R+sL)} = \frac{1}{R + sL + K_T}$$

$$R \rightarrow R + K_T$$

An increase in the apparent value of amature resistance further damps a resonance which is already well damped, but it reduce the damping on the resonance which causes the problems.

$$\zeta = \frac{K_m^2}{2J_m R}\frac{1}{\omega_n}$$

Clearly, torque feedback is not the solution in the electrical case. We could consider for torque feedback, but this is a poor engineering solution because of the extreme sensitivity of the system to such feedback. In the electrical case motor velocity feedback has been found to solve the problem. Just considering up to the s^3 term again we have

$$\frac{K_s K_m \zeta N}{\zeta N^2 J_L R s^3 + (\zeta N^2 J_L K_m^2 + \zeta N^2 K_m J_L c)s^2 + F's + G'}$$

$$= \frac{26.8}{s^3 + 35.7(0.25+c)s^2 + 4426s + 21.4 \times 10^3 (0.25+c)}$$

$$F' = \zeta N^2 K_s J_m R + K_s J_L R \quad and \quad G' = \zeta N^2 K_m^2 + \zeta N^2 K_s K_m c$$

$\omega_n^2 = 4426$ *as before and* $\omega_n = 66.5\ rad/sec$

and $f_n = 10.6\ Hz$

$2\zeta\omega_n = 35.7(0.25+c)$ *Therefore* $\zeta = 0.067 + 0.268c$

Suppose we again ζ=1, then c=3.476. This is reasonable enough, but what happens to the lower resonance which previously we found to be overdamped.

$\omega_n^2 = 599$ *Therefore* $\omega_n = 24.46\ rad/sec$

and $f_n = 3.9\ Hz$ *Therefore* $\zeta = 0.067 + 0.268c$

as before $2\zeta\omega_n = \frac{4426}{35.7(0.25+c)}$ *Therefore* $\zeta = 0.698$

This lower resonance is now underdamped.

From our knowledge of the hydraulic case a solution is to include torque feedback as well as motor velocity feedback. However, it is clear that damping on resonance by feedback may result in some other resonance becoming underdamped. It is also far from clear whether we would gain anything by having both torque and motor velocity feedback. Perhaps each will tend to deflect the effects of the other. It should be borne in mind that we don't mind resonances as long as they are overdamped. Of course, feedback surely damps resonances when the system is not saturated, so if we rely on feedback for damping we would still wish our resonant frequencies to be high.

To find out whether feedback helps us, we consider a general 5th order transfer function. The transfer function is 5th order to allow for power amplifier dynamics. Consider

$$\frac{K}{s^5 + as^4 + bs^3 + cs^2 + ds + e}$$

We always wish to raise e(coefficient e in generalized transfer function) to improve the overall performance. This parameter is raised by increasing the load velocity feedback. The question is can we obtain a situation when all possible resonances are overdamped regardless of increase in e. From above transfer function we have the following four damping ratios.

$$\zeta_1 = \frac{1}{2}\frac{d}{c}\sqrt{\frac{c}{e}} = \frac{1}{2}\frac{d}{\sqrt{ce}}, \quad \zeta_2 = \frac{1}{2}\frac{c}{b}\sqrt{\frac{b}{d}} = \frac{1}{2}\frac{c}{\sqrt{bd}}$$

$$\zeta_3 = \frac{1}{2}\frac{b}{a}\sqrt{\frac{a}{c}} = \frac{1}{2}\frac{1}{\sqrt{ac}}, \quad \zeta_4 = \frac{1}{2}\frac{a}{\sqrt{b}}$$

Put $\frac{d}{\sqrt{ce}} = \frac{c}{\sqrt{bd}} = \frac{b}{\sqrt{ac}} = \frac{a}{\sqrt{b}}$, and require d, c, b in terms of a and e, also these ζ_i in terms of a and e, then we have. $d = ae^{3/5}$, $c = a^{1/2}e^{3/10}$, $b = a^{1/2}e^{1/10}$.

Then

$$\zeta_1 = \frac{1}{2}\frac{d}{\sqrt{ce}} = \frac{1}{2}\frac{a^{1/4}}{e^{1/20}}, \quad \zeta_2 = \frac{1}{2}\frac{c}{\sqrt{bd}} = \frac{1}{2}\frac{a^{1/4}}{e^{1/20}},$$

$$\zeta_3 = \frac{1}{2}\frac{b}{\sqrt{ac}} = \frac{1}{2}\frac{a^{1/4}}{e^{1/20}}, \quad \zeta_4 = \frac{1}{2}\frac{a}{\sqrt{b}} = \frac{1}{2}\frac{a^{1/4}}{e^{1/20}}$$

To damp resonances resulting from rise in e, make a as large as possible and then calculate d, c, b. Tendency is to make ζ large or also to keep poles on same diagonal?

$$tan\,\theta = \frac{\omega_n\sqrt{1-\zeta^2}}{\zeta\omega_n} = \frac{\sqrt{1-\zeta^2}}{\zeta}$$

If all ζ are the same then all poles are on same diagonal. But we are not really leading with poles.

$$\frac{K}{s^5+as^4+bs^3+cs^2+ds+e}$$
$$=\frac{K}{s^5+as^4+a^{1/2}e^{1/10}s^3+a^{1/2}e^{1/10}s^2+ae^{1/5}s+e}$$

$$\omega_1^2=\frac{e}{c}=\frac{e}{a^{1/2}e^{1/10}}=\frac{e^{1/12}}{a^{1/2}},\quad \omega_2^2=\frac{d}{b}=\frac{ae^{1/5}}{a^{1/2}e^{1/10}}=\frac{e^{1/2}}{a},$$
$$\omega_3^2=\frac{c}{a}=\frac{a^{1/2}e^{1/10}}{a}=a^{1/2}e^{1/10},\quad \omega_4^2=b=a^{1/2}e^{1/10}$$

What feedbacks can we use for this? From Fig. 4 we can derive next equations.

$$e=V_i-AV-BI-C\omega_m-FT_L-G\omega_L$$

Leave out D_m or D_L

$$V=\frac{K_a}{1+sT_a}e,\quad I=\frac{1}{R+Ls}(V-K_m\omega_m),\quad T_m=K_mI,$$
$$\omega_m=\frac{1}{J_ms}(T_m-T_1),\quad T_1=\frac{T_L}{\zeta N},\quad \omega_2=\frac{1}{N}\omega_m,$$
$$T_L=\frac{K_s}{s}(\omega_2-\omega_L),\quad \omega_L=\frac{T_L}{J_Ls}$$

Require $\frac{\omega_L}{V_i}$ and we have $\frac{\omega_L}{V_i}=\frac{1}{Next\ equations}$

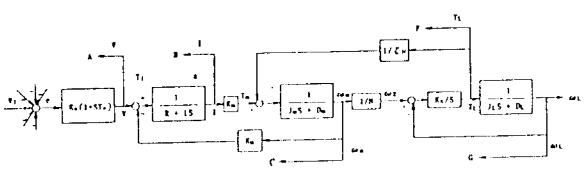

Fig. 4 Servo Control System with Feedbacks

$$s^5)\quad \frac{T_a}{K_a}\frac{J_mJ_LNL}{K_sK_m}=\frac{0.112\times10^{-6}}{K_a}$$

$$s^4)\quad \frac{T_a}{K_a}\frac{J_mJ_LNR}{K_sK_m}+\frac{1+K_aA}{K_a}\frac{J_mJ_LNL}{K_sK_m}$$
$$=\frac{74.67\times10^{-6}}{K_a}+56\times10^{-6}\frac{1+K_aA}{K_a}$$

$$s^3)\quad \frac{1+K_aA}{K_a}\frac{J_mJ_LNR}{K_sK_m}+\frac{T_a}{K_a}\frac{LJ_mN}{K_m}+\frac{T_a}{K_a}\frac{LJ_L}{\zeta NK_m}+\frac{T_a}{K_a}\frac{K_mNJ_L}{K_s}$$
$$+\frac{BJ_mJ_LN}{K_mK_s}=37.3\times10^{-3}\frac{1+K_aA}{K_a}+\frac{6.72\times10^{-5}}{K_a}$$
$$+\frac{4\times10^{-4}}{K_a}+\frac{6.67\times10^{-4}}{K_a}+0.3733B$$

$$s^2)\quad \frac{T_a}{K_a}\frac{RJ_mN}{K_m}+\frac{1+K_aA}{K_a}\frac{LJ_mA}{K_m}+\frac{T_a}{K_a}\frac{RJ_L}{\zeta NK_m}+\frac{1+K_aA}{K_a}\frac{LJ_L}{\zeta NK_m}$$
$$+\frac{1+K_aA}{K_a}\frac{K_mNJ_L}{K_s}+\frac{CNJ_L}{K_s}=\frac{0.0448}{K_a}+0.0336\frac{1+K_aA}{K_a}$$
$$+\frac{0.2667}{K_a}+0.2\frac{1+K_aA}{K_a}+0.333\frac{1+K_aA}{K_a}+1.333C$$

$$s^1)\quad \frac{1+K_aA}{K_a}\frac{BJ_mN}{K_m}+\frac{1+K_aA}{K_a}\frac{RJ_L}{\zeta NK_m}+\frac{T_aK_mN}{K_a}+\frac{BJ_mN}{K_m}$$
$$+\frac{BJ_L}{K_m\zeta N}+FJ_L=22.4\frac{1+K_aA}{K_a}+133.3\frac{1+K_aA}{K_a}+\frac{0.4}{K_a}$$
$$+22.4B+133.3B+20\times10^3F$$

$$s^0)\quad \frac{1+K_aA}{K_a}K_mN+CN+G=20\frac{1+K_aA}{K_a}+800C+G$$

Coefficients of s^4 affected by A. So adjust A(voltage). Coefficients of s^3 affected by A or B. So adjust B(current). Coefficients of s^2 affected by A or C. So adjust C(motor velocity). Coefficients of s^1 affected by A or B or F. So adjust F(load torque). Coefficients of s^0 affected by A or C or G. So adjust G(load velocity).

We are effectively increasing G in any case to obtain the benefit of negative velocity feedback. Consider time delay effects when adjusting variables. Refer control system feedbacks block diagram. One typical result in each case is shown below.

Electric case ; $-180°$ at 65 rad/sec

State	K_a	K_{ω_m}	LF Gain	1Hz Gain	0 dB ω	Resonance
unstable	2300	0	11.49	2.15	14	60~72
stable	2300	2.5×10^{-4}	4.55	4.4	55	none

Hydraulic case ; $-180°$ at 31 rad/sec($K_T=0$)
$-180°$ at 104 rad/sec($K_T=10^{-8}$)

State	K_a	K_T	LF Gain	1Hz Gain	0 dB ω	Resonance
unstable	0.25	0	9.61	10.3	78	24.5
stable	0.25	10^{-8}	9.61	5.3	50	none

Conclusions

The damping control problems of the electric and electro-hydraulic drive systems to drive large loads have been studied in relation to the turret driving system for military vehicles. The available parameters are identified and the fourth order transfer functions to analyze the effects of feedback parameters are derived and compared with both systems. The fifth order servo control systems with various state feedbacks are modeled and analyzed to identify the effects of control gains and feedback loop gains.

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