Compensating Structural Dynamics For Servo Driven Industrial Machines with Acceleration Feedback

George W. Younkin, Life Fellow, IEEE
Industrial Controls Consulting Division
Bull's Eye Research, Inc.
N7614 HWY 149
Fond du Lac, Wi 54935
T: 920-929-6544 F: 920-929-9344
E: george@bullseyenet.com

I. INTRODUCTION

Industrial servo drives, over a period of time, have become very reliable with mean time between failure in years. Industrial machines, referred to as the servo plant, have fewer problems with mechanical nonlinearities. Such things as backlash and stiction in machine axis slides have been minimized. On the other hand, structural dynamics of industrial machines continue to be a problem with mechanical resonances occurring inside the servo position loop resulting in unstable servo drives. There are some techniques to compensate for existing structural resonances. The ideal situation is to design an industrial machine without dynamic problems. In reality this situation does not always exist. Some of the compensating techniques for machine resonances are notch filters, frequency selective feedback, and acceleration feedback. The subject of this paper is the application of acceleration feedback compensation in an actual industrial machine servo drive.

II. DISCUSSION

For this discussion a worst case condition for a large industrial servo axis will be used. The following parameters are assumed from this industrial machine servo application:

Motor - Kollmorgen motor - M607B
Machine slide weight - 50,000 lbs
Ball screw: Length - 70 inches
Diameter - 3 inches
Lead - 0.375 inches/revolution
Pulley ratio - 3.333
J_T = Total inertia at the motor= 0.3511 lb-in-sec^2
t_e = Electrical time constant = 0.02 second = 50 rad/sec
$t_1 = t_e$
K_e = Motor voltage constant = 0.646 volt-sec/radian
K_T = Motor torque constant = 9.9 lb-in/amp
K_G = Amplifier gain = 20 volts/volt
K_{ie} = Current loop feedback constant = 3 volts/40A= 0.075 volt/amp
R_a = Motor armature circuit resistance = 0.189 ohm
K_i = Integral current gain = 735 amp/sec/radian/sec

The block diagram of fig. 1 represents dc and brushless dc motors. All commercial industrial servo drives make use of a current loop for torque regulation requirements. fig. 1 includes the current loop for the servo drive with PI compensation. Since the block diagram of fig. 1 is not solvable, block diagram algebra separates the servo loops to an inner and outer servo loop of
fig 2.

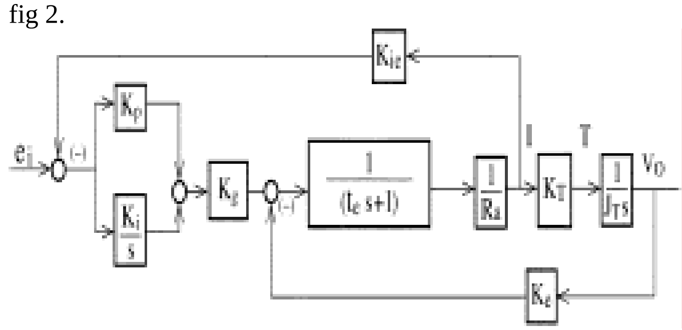

Figure1. Motor and current loop block diagram

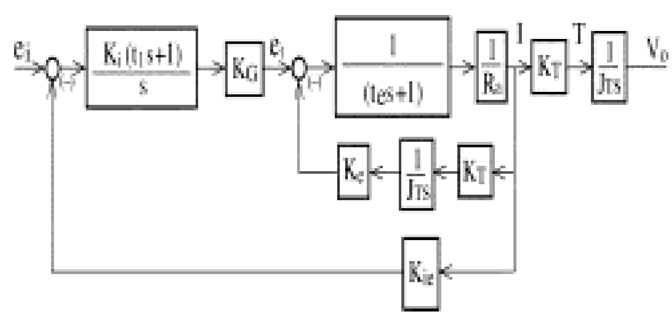

Figure 2. Equivalent motor and current loop
Block diagram

The first step in the analysis is to solve the inner loop of fig. 2. The closed loop response I/e_1= G/1+GH where:

$$G_{(s)} = \frac{1}{R_a(t_e s+1)} = \frac{5.29}{(t_e s+1)} \quad (5.29=14.4 \text{ dB}) \qquad (1)$$

$$GH_{(s)} = \frac{0.646x9.9}{0.189x0.3511(t_e s+1)s} \quad (2)$$

$$GH_{(s)} = \frac{96}{s(t_e s+1)} \quad (96 = 39dB) \quad (3)$$

$$\frac{1}{H_{(s)}} = \frac{J_T s}{K_e K_T} = \frac{0.3511s}{0.646x9.9} \quad (4)$$

$$\frac{1}{H_{(s)}} = 0.054s \quad (0.054 = -25\ dB) \quad (5)$$

Using the rules of Bode, the resulting closed loop Bode plot for I/e_1 is shown in fig. 3. Solving the closed loop mathematically :

$$\frac{I_{(s)}}{e_1(s)} = \frac{G}{1+GH} = \frac{1}{R_a(t_e s+1) + K_e K_T / J_T s} =$$

$$\frac{J_T s}{J_T R_a s(t_e s+1) + K_e K_T} \quad (6)$$

$$\frac{I_{(s)}}{e_{1(s)}} \frac{J_T s}{J_T R_a t_e s^2 + J_T R_a s + K_e K_T} =$$

$$\frac{J_T s / K_e K_T}{[(J_T R_a / K_e K_T) t_e s^2 + (J_T R_a / K_e K_T)s + 1]} \quad (7)$$

$$\frac{I_{(s)}}{e_{i(s)}} = \frac{(0.3511/(0.646x9.9)s}{t_m t_e s^2 + t_m + 1} =$$

$$\frac{0.054s}{0.01x0.02s^2 + 0.01s + 1} \quad (8)$$

where: $t_m = \dfrac{J_T R_a}{K_e K_T} = \dfrac{0.3511x0.189}{0.646x9.9} =$

$$0.01\text{sec},\ w_m = \frac{1}{t_m} = 100 \ \frac{rad}{\text{sec}} \quad (9)$$

$$t_e = 0.02\text{sec},\ w_e = \frac{1}{t_e} = 50 \ \frac{rad}{\text{sec}} \quad (10)$$

For a general quadratic-

$$\frac{s^2}{w_r{}^2} + \frac{2\delta s}{w_r} + 1 \quad (11)$$

$w_r = [w_m\ w_e]^{1/2} = [100 \times 50]^{1/2} = 70$ rad/sec (12)

$$\frac{I_{(s)}}{e_{1(s)}} = \frac{0.054s}{s^2/70^2 + (2\delta/70)s + 1} \quad (13)$$

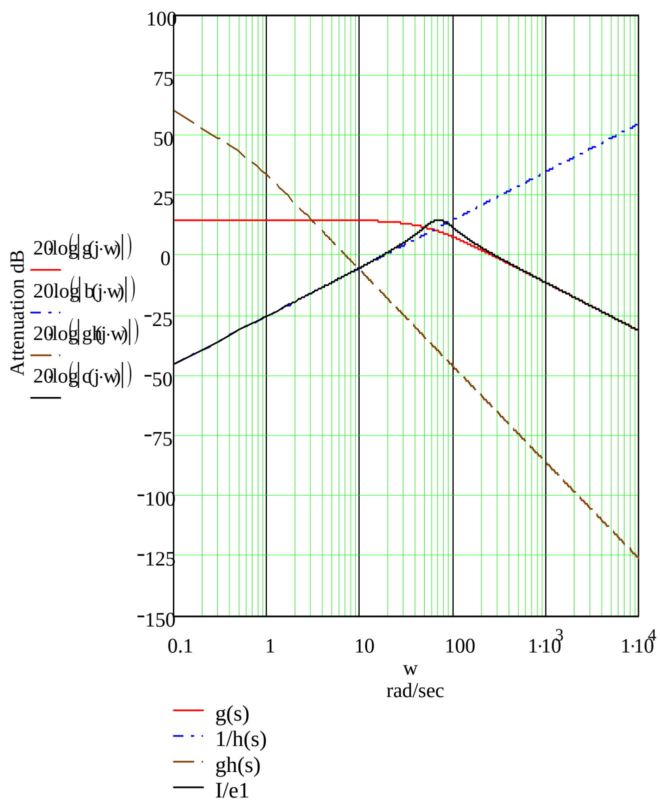

Figure 3. Current inner loop

Having solved the inner servo loop it is now required to solve the outer current loop. The inner servo loop is shown as part of the current loop in fig. 4.

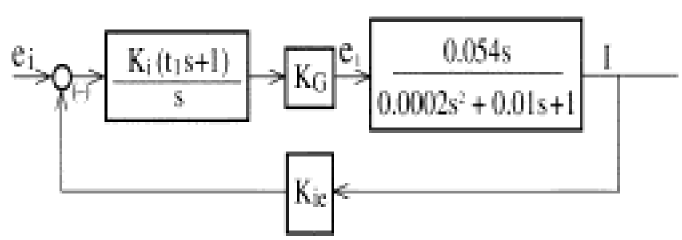

Figure 4. Current loop

In solving the current loop, the forward loop, open loop, and feedback loop must be identified as follows:

The forward servo loop-

$$G_{(s)} = \frac{K_i K_G x0.054(0.02s+1)}{0.0002s^2+0.01s+1} =$$

$$\frac{735x20x0.054(0.02s+1)}{0.0002s^2+0.01s+1} \qquad (14)$$

$$G_{(s)} = \frac{794(0.02s+1)}{0.0002s^2+0.01s+1} =$$

$$\frac{15.88s+794}{0.0002s^2+0.01s+1} \qquad (15)$$

Where: K_G =20 volt/volt

K_{ie} = 3/40 = 0.075 volt/amp

$K_i K_G$ x 0.054 794 (58 dB)

K_i = 794/(20 x 0.054)=735

$$G_{(s)} = \frac{79{,}400s+3{,}970{,}000}{s^2+50s+5000} \qquad (16)$$

The open loop-

$$GH_{(s)} = 0.075\ x\ \frac{79{,}400s+3{,}970{,}000}{s^2+50s+5000} \qquad (17)$$

$$GH_{(s)} = \frac{5955s+297{,}750}{s^2+50s+5000} \qquad (18)$$

The feedback current scaling is-

$H_{(s)}$ = 3 volts/40 amps = 0.075 volts/amp

$1/H_{(s)}$ = 13.33 = 22.4 dB (19)

$$\frac{I_{(s)}}{e_{i(s)}} = \frac{794(0.02s+1)}{0.0002s^2+1.215s+60} =$$

$$\frac{13.3(0.02s+1)}{(0.0000033s^2+0.02s+1)} =$$

$$\frac{13.3(0.02s+1)}{(0.000166S+1)(0.02S+1)} = \frac{13.3}{\left(\frac{J\omega}{6000}+1\right)} \qquad (20)$$

The Bode plot frequency response is shown in fig. 5. The current loop bandwidth is 6000 radians/second or about 1000 Hz, which is realistic for commercial industrial servo drives.

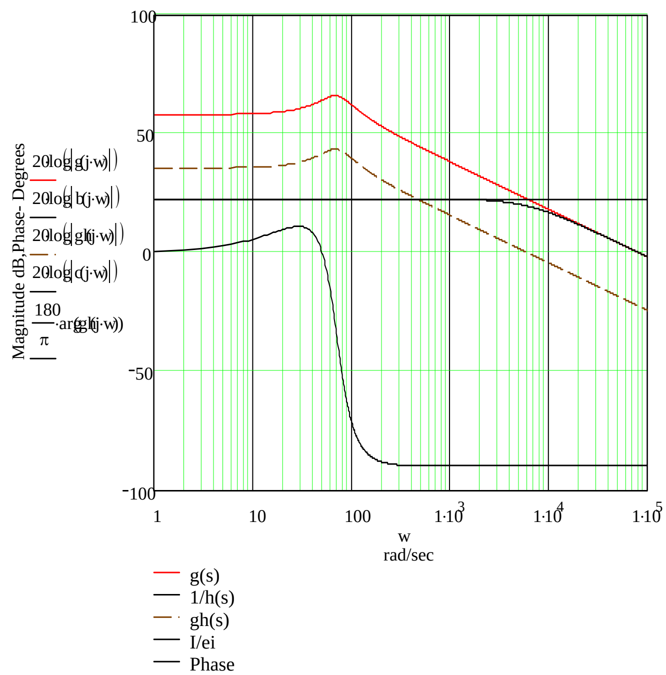

Figure 5. Current loop response

The current loop as shown in fig. 5 can now be included in the motor servo loop with reference to fig. 2 and reduces to the motor servo loop block diagram of fig. 6.

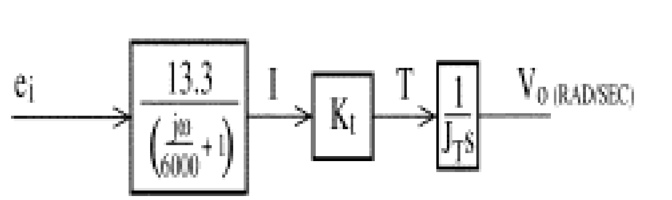

Figure 6. Motor and current loop

The completed motor servo loop has a forward loop only (as shown in fig. 6) where:

J_T = Total inertia at the motor = 0.3511 lb-in-sec^2

K_T = Motor torque constant = 9.9 lb-in/amp

$$G_{(s)} = \frac{13.3x9.9}{0.3511s((jw/6000)+1)} =$$

$$\frac{375}{s(0.000166s+1)} \quad (51.5\ dB) \qquad (21)$$

$$G_{(s)} = \frac{375}{0.000166s^2+s+0} = \frac{2,250,090}{s^2+6000s+0} \qquad (22)$$

$$\frac{V_{o(s)}}{e_{i(s)}} = \frac{K_T}{J_T s} \ x \ \frac{[I]}{[e_i]} = \frac{9.9}{0.3511s}$$

$$x \frac{13.3(0.02s+1)}{0.00000331s^2+0.02s+1} \qquad (23)$$

$$\frac{V_{o(s)}}{e_{i(s)}} = \frac{375(0.02s+1)}{0.00000331s(s+50)(s+5991)} \qquad (24)$$

$$\frac{V_{o(s)}}{e_{i(s)}} = \frac{375(0.02s+1)}{s(0.02s+1)(0.000166s+1)} \qquad (25)$$

$$\frac{V_{o(s)}}{e_{i(s)}} = \frac{375}{s((jw/5991)+1)} \qquad (26)$$

Since the current loop bandwidth (6000 rad/sec) is very high, it can be neglected. Thus $\frac{V_{o(s)}}{e_{i(s)}}$

Becomes-

$$\frac{V_{o(s)}}{e_{i(s)}} = \frac{375}{s} \qquad (27)$$

The Bode frequency response for the motor and current loop is shown in fig. 7. The motor and current closed loop frequency response, indicate that the response is an integration which includes the 6000 rad/sec bandwidth of the current loop. This is a realistic bandwidth for commercial industrial servo drives.

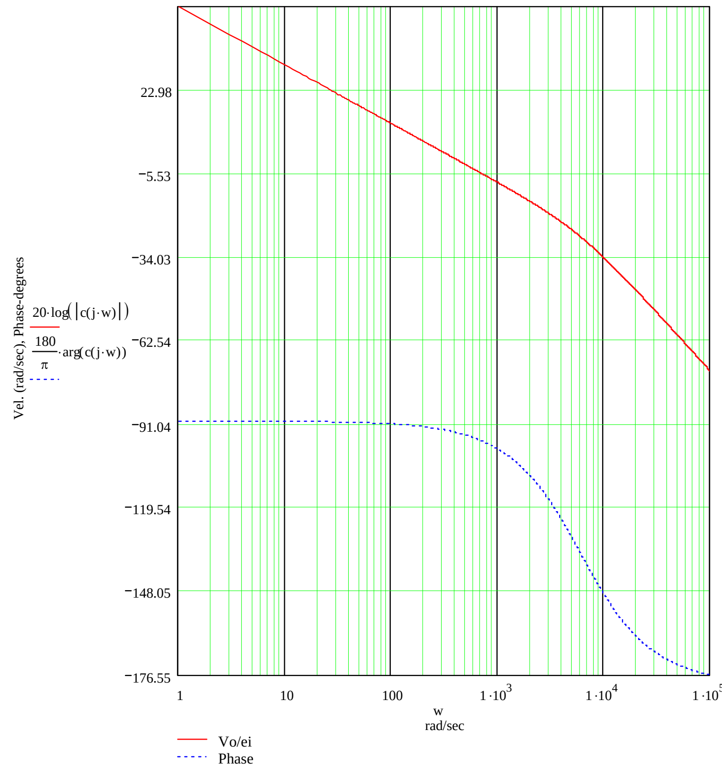

Figure 7. Motor and I loop frequency Response

For the purposes of this discussion it will be assumed that the motor and current loop are enclosed in a velocity servo loop. Such an arrangement is shown in fig. 8.

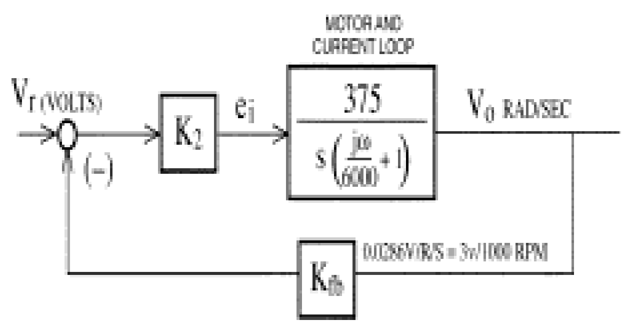

Figure 8. Velocity loop

The servo compensation and amplifier gain are part of the block identified as K_2. Most industrial servo drives use proportional plus integral (PI) compensation. The amplifier and PI compensation can be represented as in figure 9[1].

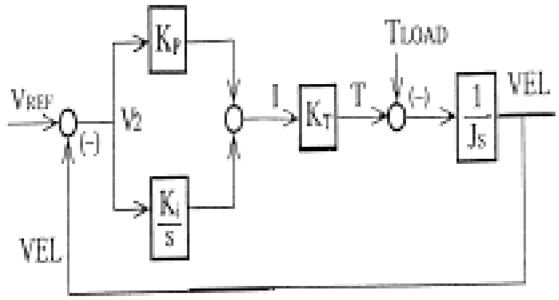

Figure 9. PI compensation

$$\frac{I}{V_2} = \left[K_p + \frac{K_i}{s}\right] = \frac{K_p s + K_i}{s} = \frac{K_i\left[\frac{K_p}{K_i}s + 1\right]}{s} = \frac{K_2\left[t_2 s + 1\right]}{s} \tag{28}$$

$$t_2 = \frac{K_p}{K_i} \qquad w_2 = \frac{K_i}{K_p} \quad \text{(Corner frequency)}$$

The adjustment of the PI compensation is suggested as-

1. For the uncompensated servo Bode plot, set the amplifier gain to a value just below the level of instability.
2. From the Bode plot for PI compensation of fig. 10, the corner frequency $w_2 = K_i/K_p$ should be lower than the 45° phase margin (-135 degrees) of fig. 7. The reason for this is that the attenuation characteristic of the PI controller has a phase lag that is detrimental to the servo phase margin. Thus the corner frequency of the PI compensation should be lowered about one decade or more from the –135 degree phase shift point (w_g) of the open loop Bode plot for the servo drive being compensated. For the servo drive being considered, w_g occurs at 6000 rad/sec.

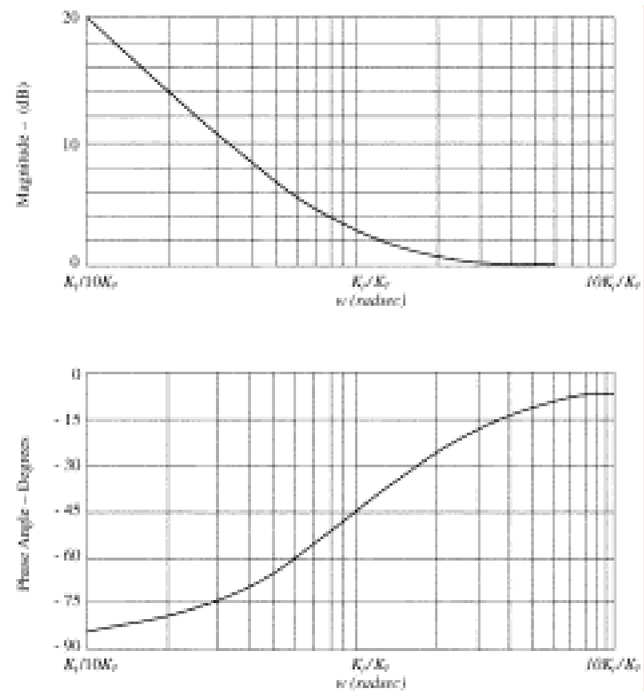

Figure 10. PI compensation response

Applying the PI compensation of fig. 9, to the velocity servo drive is shown if fig. 11.

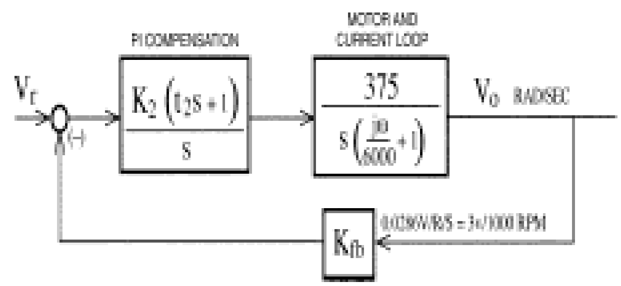

Figure 11. Velocity loop with PI compensation

In general the accepted rule for setting the servo compensation begins by removing the integral and/or differential compensation. The proportional gain is then adjusted to a level where the velocity servo response is just stable. The proportional gain is then reduced slightly further for a margin of safety.

At this point the PI compensation is added as shown in fig. 11. The index of performance for the PI compensation is that the corner frequency $w_2 = K_i/K_p$, should be a decade or more lower than the –135 degree phase shift (45 degree phase margin) frequency (w_g) of the forward loop Bode plot (fig. 7) for the industrial servo drive being considered[1].

By lowering the PI compensation corner frequency ($w_2 = \frac{K_i}{K_p}$) to 20 rad/sec (0.05 sec), a stable velocity servo drive results. The forward loop and open loop are defined as follows:

$H_{(s)}$ = 0.0286 v/rad/sec 1/$H_{(s)}$ = 34.9 (30.8 dB)

Gain @ w=1 rad/sec = 100 dB = 100,000

$$K_2 = 100{,}000/375 = 266 \qquad (29)$$

$$G_{(s)} = \frac{K_2 x375((j\omega/20)+1)}{s^2((j\omega/6000)+1)} =$$

$$\frac{100{,}000((j\omega/20)+1}{s^2((j\omega/6000)+1)} \qquad (30)$$

$$G_{(s)} = \frac{100{,}000(0.05s+1)}{s^2(0.000166s+1)} = \frac{5000s+100{,}000}{s^2(0.000166s+1)} \qquad (31)$$

$$G_{(s)} = \frac{30{,}120{,}481s+602{,}409{,}638}{s^3+6024s^2+0s+0} \qquad (32)$$

$$GH_{(s)} = 0.0286xG = \frac{2860((j\omega/20)+1)}{s^2((j\omega/6000)+1)} 69dB \qquad (33)$$

The Bode plot for the velocity loop with PI compensation is shown in fig. 12, having a typical industrial velocity servo bandwidth of 30 Hz (188 rad/sec).

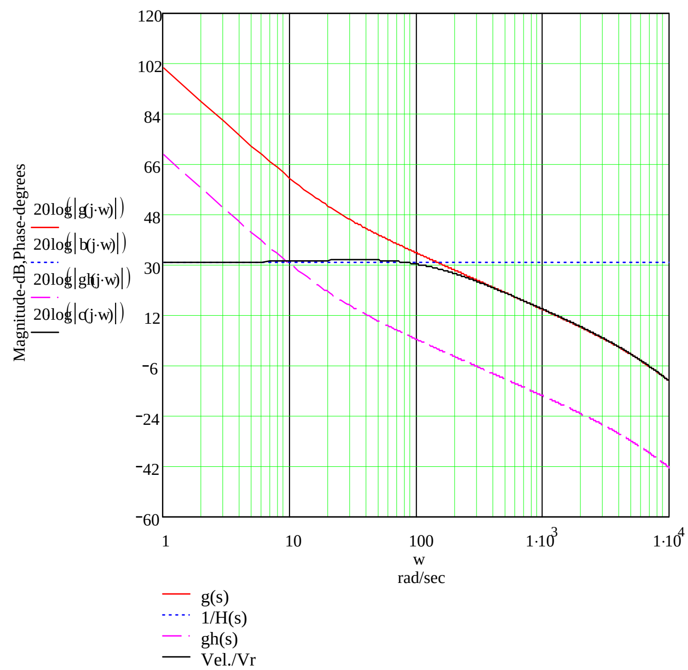

Figure 12. Velocity servo response

POSITION SERVO LOOP COMPENSATION

Having compensated the velocity servo, it remains to close the position servo around the velocity servo. Commercial industrial positioning servos do not normally use any form of integral compensation in the position loop. This is referred to as a “naked” position servo loop. However, for type 2 positioning drives, PI compensation would be used in the forward position loop. There are also some indexes of performance rules for the separation of inner servo loops by their respective bandwidths[3]. The first index of performance is known as the 3 to 1 rule for the separation of a machine resonance from the inner velocity servo. All industrial machines have some dynamic characteristics, which include a multiplicity of machine resonances. It is usually the lowest mechanical resonance that is considered; and the index of performance is that the inner velocity servo bandwidth should be 1/3 of the predominant machine structural resonance.

A second index of performance is that the position servo velocity constant (K_v) or position loop gain, should be ½ the velocity servo bandwidth[3]. These indexes of performance are guides for separating servo loop bandwidths to maintain some phase margin and overall system stability. Industrial machine servo drives usually require low position loop gains to minimize the possibility of exciting machine resonances. In general for large industrial machines, the position loop gain (K_v) is set about 1 ipm/mil (16.66/sec). The example being studied in this discussion has a machine slide weight of 50,000 lbs., which can be considered a large machine that could have detrimental machine dynamics. There are numerous small machine applications where the position loop gain can be increased several orders of magnitude. The technique of using a low position loop gain is referred to as the “soft servo”. A low position loop gain can be detrimental to such things as servo drive stiffness and accuracy. The “soft servo” technique also requires a high-performance inner velocity servo loop. This inner velocity servo loop with its high-gain forward loop, overcomes the problem of low stiffness. For example, as the machine servo drive encounters a load disturbance the velocity will instantaneously try to reduce, increasing the velocity servo error. However the high velocity servo forward loop gain will cause the machine axis to drive right through the load disturbance. This action is an inherent part of the drive stiffness[3].

For this discussion it will be assumed that the industrial machine servo drive being considered has a structural mechanical spring/mass resonance inside the position loop. The machine as connected to the velocity servo drive is often referred to as the “servo plant”. The total machine/servo system can be simulated quite accurately to include the various force or torque feedback loops for the total system[4]. For expediency in this discussion, a predominant spring/mass resonance will be added to the output of the velocity servo drive. Thus the total servo system is shown in the block diagram of fig. 13. Position feedback is measured at the machine slide to attain the best position accuracy.

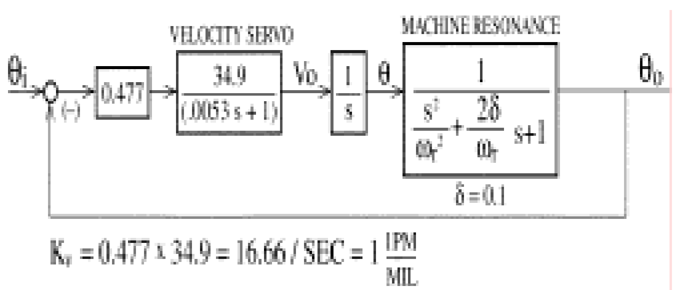

Figure13. Position loop block diagram

In reality a machine axis weighing 25 tons will have low frequency structural resonances. Machine axes of this magnitude in size will characteristically have structural resonances of about 10Hz to 20Hz. Using the same position servo block diagram of fig. 13 with the same position loop gain of 1 ipm/mil (16.66/sec), and a machine resonance of 10 Hz; the servo frequency response is shown in fig. 14 with the transient response shown in fig. 15. The position servo frequency response shows an 8 dB resonant (62.8 rad/sec) peak over zero dB, which will certainly be unstable as observed in the transient response of fig. 15

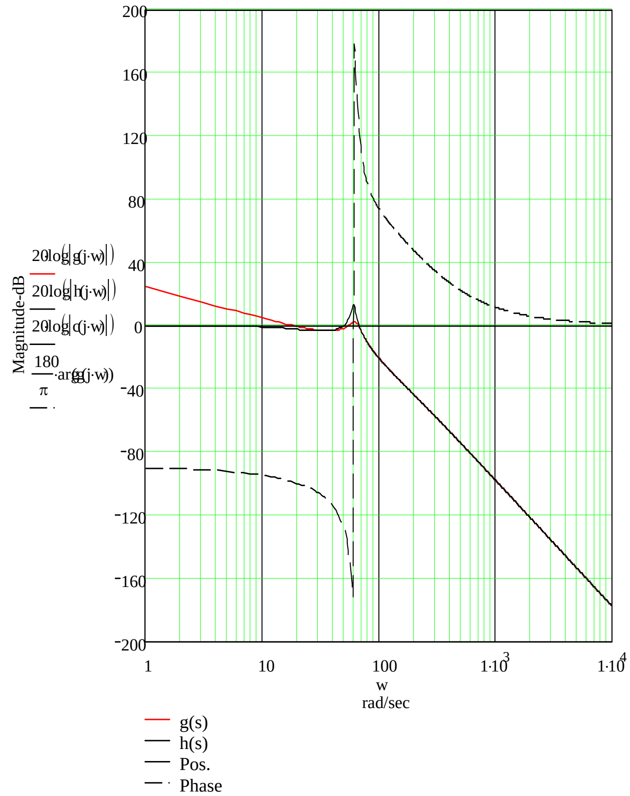

Figure 14. Position loop response

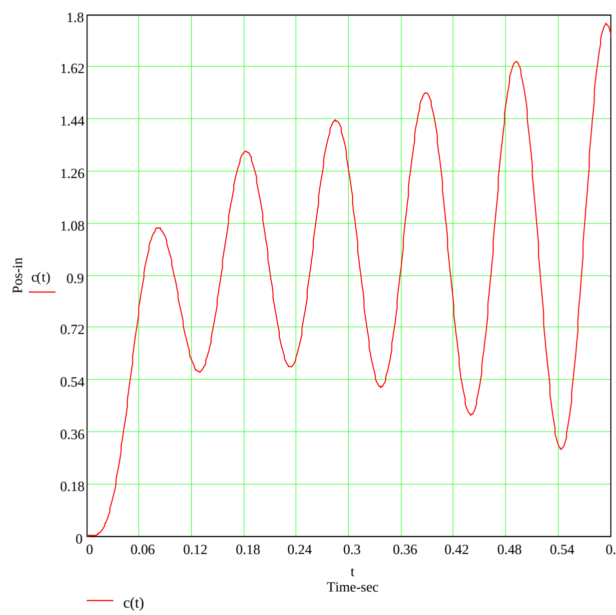

Figure 15. Position transient response

One of the most significant problems with industrial machines is in the area of machine dynamics. Servomotors and their associated amplifiers have very long mean time between failure characteristics. It is quite common to have an industrial velocity servo drive with 20 Hz to 30 Hz bandwidths mounted on a machine axis having structural dynamics (resonances) near or much lower than the internal velocity servo bandwidth. There must be some control concept to compensate for these situations. There are a number of control techniques that can be applied to compensate for machine structural resonances that are both low in frequency and inside the position servo loop. The first control technique is to lower the position loop gain (velocity constant). Depending on how low the machine resonance is, the position loop gain may have to be lowered to about 0.5 ipm/mil (8.33/sec.). This solution has been used in numerous industrial positioning servo drives. However, such a solution also degrades servo performance. For very large machines this may not be acceptable. The index of performance that the position loop gain (velocity constant) should be lower than the velocity servo bandwidth by a factor of two, will be compromised in these circumstances.

A very useful control technique to compensate for a machine resonance is the use of wien bridge notch filters[3]. These notch filters are most effective when placed in cascade with the position forward servo loop, such as at the input to the velocity servo drive. These notch filters should have a tunable range from approximately 5 Hz to a couple of decades higher in frequency. The notch filters are effective to compensate for fixed machine structural resonances. If the resonance varies due to such things as load changes, the notch filter will not be effective. There are commercial control suppliers that incorporate digital versions of a notch filter in the control; with a future goal to sense a resonant frequency and tune the notch filter to compensate for it. This control technique can be described as an adaptive process.

Another technique that has been very successful with industrial machines having low machine resonances, is known as "frequency selective feedback[5,6,7]". This control technique is the subject of another discussion. In abbreviated form it requires that the position feedback be located at the servo motor eliminating the mechanical resonances from the position servo loop, resulting in a stable servo drive but with significant position errors. These position errors are compensated for by measuring the machine slide position through a low pass filter; taking the position difference between the servo motor position and the machine slide position; and making a correction to the position loop; which is primarily closed at the servo motor.

A third machine dynamics compensation technique is referred to as acceleration feedback[8]. With this control technique the output position of the machine slide is differentiated twice to obtain acceleration. This can be accomplished with an accelerometer located at the machine slide. The output signal will require some filtering to eliminate electrical noise. The output of the accelerometer represents the acceleration of the servo-driven machine slide, and will be multiplied by a scale factor K_f=10. This acceleration signal is used as a feedback and summed at the input (e_i) of the motor and current loop as shown in fig. 16. The result is two interacting servo loops. To continue further analysis, block diagram algebra is used to separate the servo loops as shown in fig. 17.

The motor and current loops have an integral transfer function with a current loop bandwidth of 6000 rad/sec. This bandwidth is normal for a current loop and is sufficiently high in frequency to be neglected since the frequency is much higher than the servo bandwidths of the servo drive. The motor and current loop appear with the transfer function of-375/s, which is correct since a motor is an integrator.

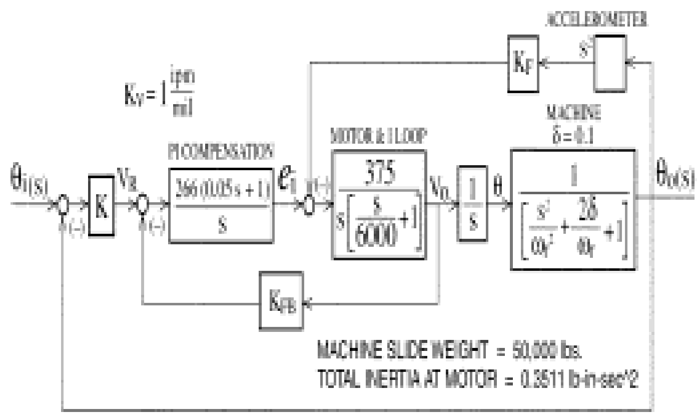

Figure 16. Industrial BLDC servo drive with a 10 Hz resonance in the machine structure and acceleration feedback

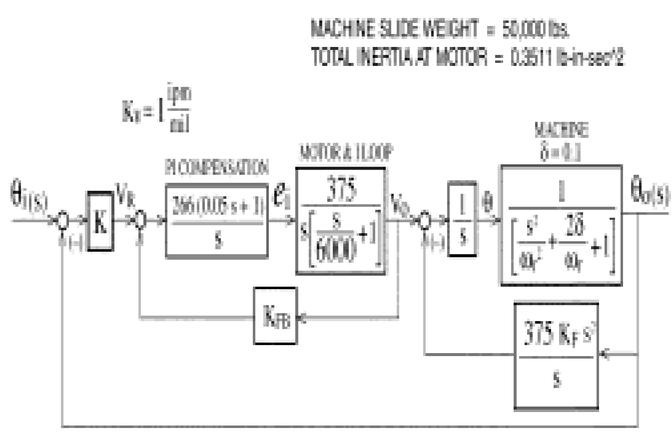

Figure 17. Equivalent Industrial BLDC servo drive with a 10 Hz resonance in the machine
structure and acceleration feedback

The next step is to close the servo plant loop $\frac{\theta_{0(s)}}{V_{o(s)}}$. The machine resonant transfer function for a 10 Hz resonance is-

$$\frac{\theta_{o(s)}}{\theta_{(s)}} = \frac{1}{0.000253s^2 + 0.00138s + 1} \quad (34)$$

The servo plant forward loop transfer function of $\frac{\theta_{o(s)}}{V_{o(s)}}$ is-

$$G_{(s)} = \frac{1}{s(0.000253s^2 + 0.00138s + 1)} \quad (35)$$

The feedback term for the servo plant transfer function $\frac{\theta_{0(s)}}{V_{o(s)}}$ for a gain K_f=10 is-

$$H_{(s)} = 3750s \quad (36)$$

The closed loop transfer function for the machine resonant servo plant is-

$$\frac{\theta_{o(s)}}{V_{o(s)}} = \frac{1}{s(0.000253s^2 + 0.00138s + 3750)} \quad (37)$$

The position forward loop transfer function for a K_v of 1 ipm/mil becomes-

3750 x 16.66=62000

$$G_{(s)} = \frac{62000}{s(0.0053s+1)}\, x$$

$$\frac{1}{(0.000253s^2 + 0.00138s + 3750)} \quad (38)$$

The closed position loop frequency response $\frac{\theta_{o(s)}}{\theta_{i(s)}}$ is shown in fig. 18. The system resonance has been shifted to 4000 rad/sec with the peak of the resonance about 20 dB below zero dB. The stable transient response of the servo drive is shown in fig. 19.

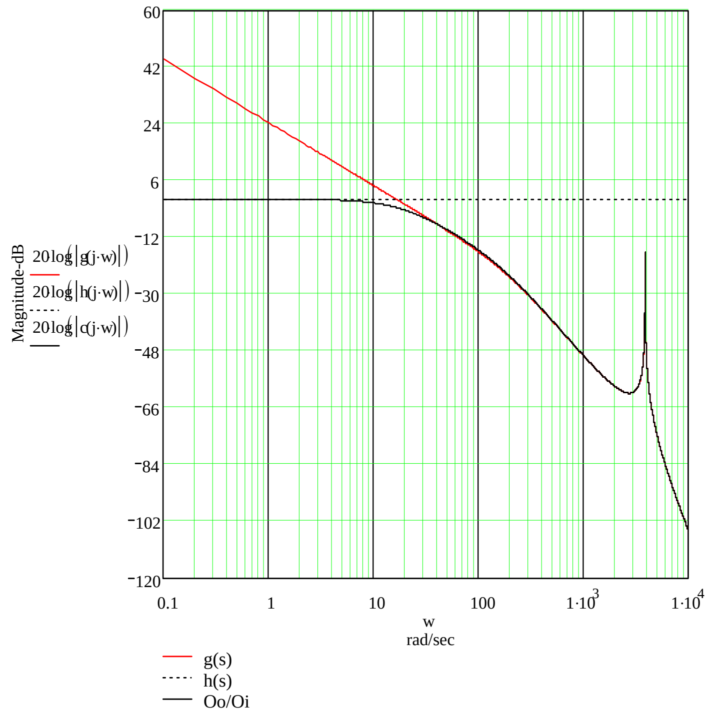

Figure 18. Position frequency response with acceleration feedback

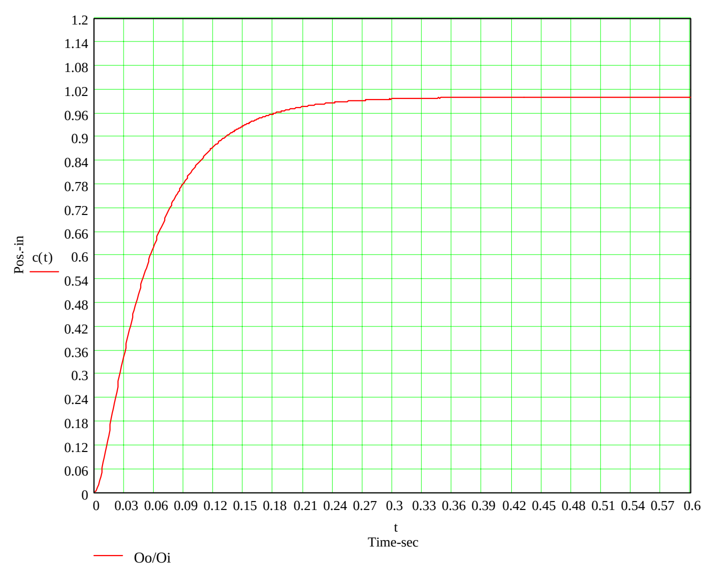

Figure 19. Transient response with Acceleration feedback

III. CONCLUSIONS

Commercial industrial electric brushless DC servo drives use an inner current/torque loop to provide adequate servo stiffness. The servo loop bandwidth for the current loop is usually about

1000 Hz. In analysis this servo loop is often neglected because of its wide bandwidth. Including the current loop as in figure 2, results in a motor and current loop response (fig. 7) that is an integration with the current loop response. A classical servo technique is to enclose the motor/current loop in a velocity servo loop. Since most commercial industrial brushless DC servo motors have position feedback from the motor armature for the purpose of current commutation; this signal is differentiated to produce a synthetic velocity loop. Additionally, commercial industrial servo drives use proportional plus integral (PI) servo compensation to stabilize the synthetic velocity loop.

The PI type of compensation has a corner frequency that must be a decade or more lower in frequency than the 45-degree phase shift frequency of the uncompensated open loop Bode plot. This requirement is needed to avoid excessive phase lag from the PI compensation where the open loop 45-degree phase shift frequency occurs. Commercial industrial electric servo drives have a very long mean time between failures, and therefore are very reliable. The servo plant (the machine that the servo drive is connected to) has ongoing problems with structural dynamics. When these mechanical resonances have low frequencies that occur within the servo bandwidths, unstable servo drives can result. There are some solutions to stabilize these unstable servo drives. Reducing the position loop gain has been used in the past with a net degradation in performance.

Two other possible solutions to stabilize these servo drives with unacceptable machine dynamics are the use of notch filters to tune out fixed frequency resonances and using a control technique referred to as "frequency selective feedback[5,6,7]". As discussed in this paper, the use of a control technique referred to as "acceleration feedback[8]" is another option in compensating for undesirable machine resonances.

References

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