MAGELLAN PROJECT

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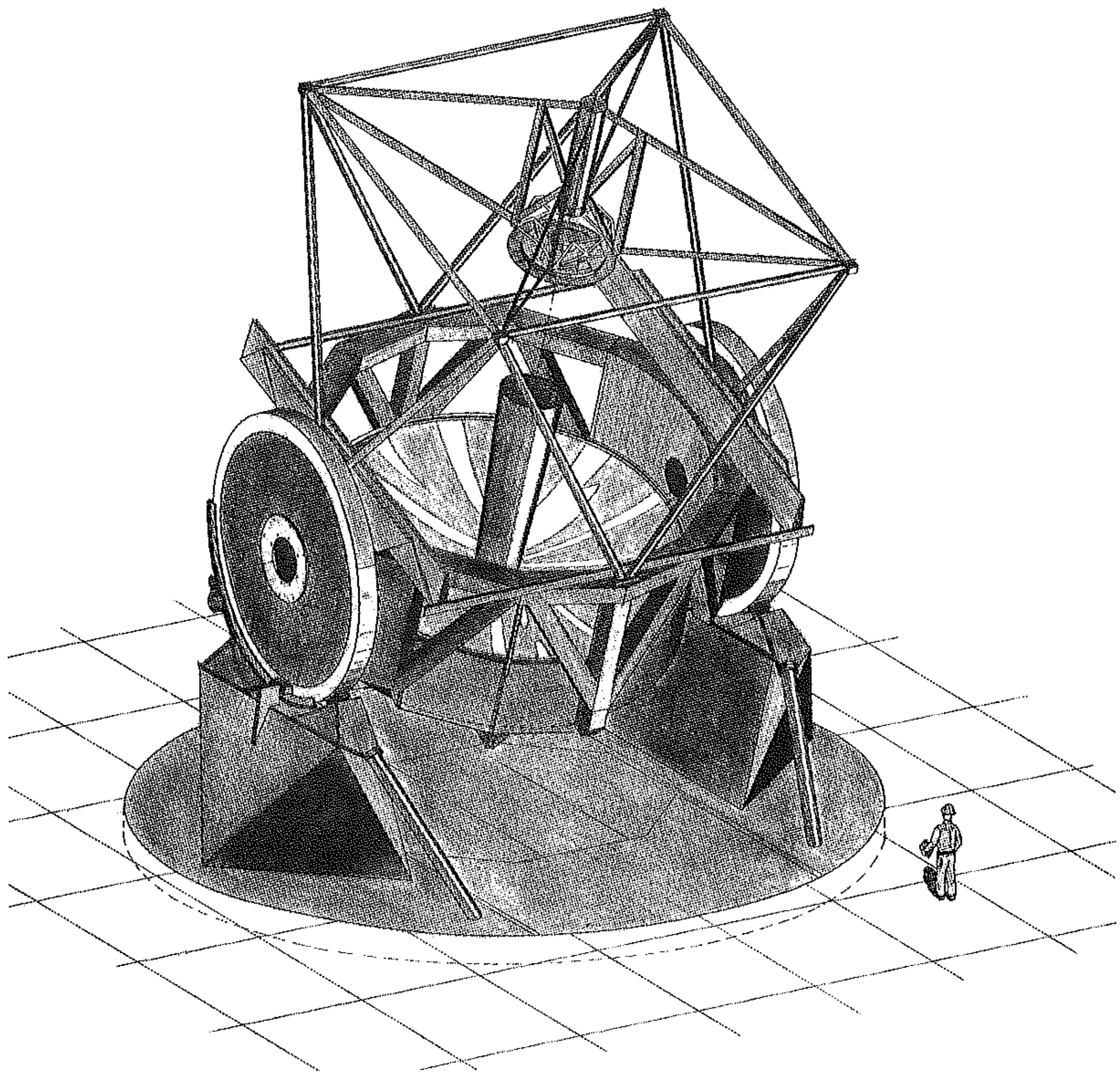

The Trouble with Tach Feedback

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CAVEAT: Throughout this report I consistently use the word "bandwidth" to mean the frequency at which the response of the system passes the -3 dB point (half-power) for the last time. This is not the same as saying that the response never goes below -3 dB until the bandwidth has been exceeded; i.e., there may be a temporary dip in the response at a frequency below the bandwidth. The result is that the term "bandwidth" gives an indication of the frequency up to which the closed loop servo "does good". The full performance of the system, however, can only be judged by looking at the full response.

If you find all this baffling, you may disregard it for now. It is important for the more rigorous aspects of this report, but it is not vital to getting the gist of the material.

INTRODUCTION

When using a motor to drive a load, it is common practice to close an inner servo loop around a tachometer signal generated directly from the motor shaft. Frequently, an outer loop is then closed around the position of the driven load. The key to the popularity of this strategy is that each of the two loops can be implemented with very simple dynamic compensators, usually consisting of nothing more than a proportional gain. Furthermore, the inner tachometer loop can be unconditionally stable up to very wide bandwidths, making its implementation somewhat foolproof.

This sort of two-loop strategy will provide satisfactory performance only under certain conditions. To work well, it is necessary for the motor shaft to be comparatively stiff, otherwise the resonance of the load on the drive shaft (often called the "locked-rotor" resonance) will prove to be a limiting factor in the system bandwidth. In many small mechanical systems, it is possible to make this resonant frequency sufficiently high. In other mechanical systems such as the main drives of large telescopes, this may not be possible. Fortunately, it is possible to avoid this difficulty through the use of a different--and in some ways simpler--control scheme which causes the system to display more favorable dynamics.

This report illustrates the potential problems in a system with tachometer feedback and compares it to system in which no tachometer is required.

ROAD MAP

This report uses the deceptively simple mechanical model shown in Figure 1. One inertia represents the rotor of a driving motor, the other inertia represents the driven load, and the intervening spring represents the compliance of the motor shaft. With proper bookkeeping of the gear ratios and inertias, this model could also represent a system with a gear train; up to the first resonance, the dynamics are identical.

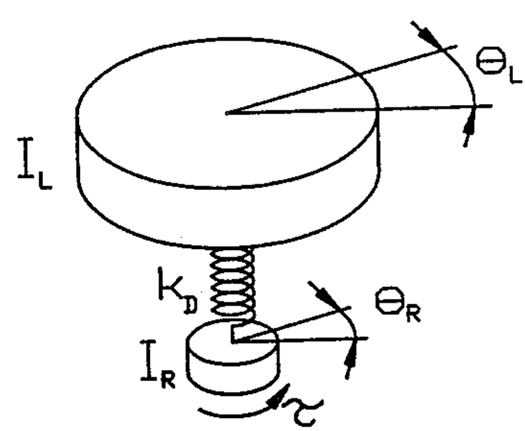


Figure 1.
The model system consists of two inertias and a spring. Torque is applied at the lower inertia (the "rotor") and positions and rates can be measured at both inertias.

The model parameters--listed in Appendix A--have been selected to make the illustrations in this report clear and to represent a general system. The selected parameters do not represent the mass properties of any particular telescope. For this analysis, the fundamental features of interest are the frequencies of the resonances and antiresonances, and not so much the masses and spring rate.

The model permits position or rate measurements at both the rotor and load inertias. Rate measurement at the rotor represents the use of a tachometer, and measurements at the load are the feedback used to close an outer loop. The input to the system is the torque applied by the motor at the rotor, and the output of primary interest is the position of the load inertia.

The analysis will proceed in three steps. First, we will show that wideband rate (tachometer) feedback is possible for an inner loop around the motor rotor. In fact, we will show that in our simplified model, the inner-loop bandwidth can be made arbitrarily large using nothing more than a proportional gain.

In the second step, we will show that with the inner loop closed, the outer loop around the load shows a resonance at the frequency of the load on the shaft compliance (the "locked-rotor" resonance), and that this resonance will limit the outer loop bandwidth in a practical system.

The third and last step of the analysis considers the dynamics of the system when motor torque is used as the system input and feedback is taken only from the load. This configuration will also show a bandwidth limiting resonance, but this time it will be at the "free-boundary" resonance of the system--a more favorable condition.

STEP ONE.
TACHOMETER FEEDBACK: WIDEBAND CONTROL IS POSSIBLE.

Figure 2 shows a sketch of the model we are analyzing with feedback on the motor rate, and Figure 3 shows the corresponding block diagram. The transfer functions corresponding to the blocks in Figure 3 are derived in Appendix A.

We first turn our attention to the transfer function from torque to rate at the motor rotor. In the block diagram of Figure 3, this is the block labeled "G". Figure 4 shows the magnitude and phase response of this transfer function, and it reveals an important fact: the phase of this transfer function never makes a steep transition to large negative phase angles. This fact is also apparent in the magnitude plot in that the resonance peak is preceded by the antiresonance valley. Note that the antiresonance appears at the relatively low locked-rotor frequency.

Since the phase of this system never goes below -90 degrees, we conclude that it is possible to close the loop using only a proportional gain, and that in principle, this gain could be arbitrarily high. In Figure 3, the gain is represented by block "A". (In any physical system there would be other elements that would contribute to the phase delay, and the phase would in fact eventually go below -180 degrees. It is however possible for this transition to be in the many-kilohertz range, which is almost infinity for our considerations here.)

Figure 5 shows now the closed-inner-loop magnitude and phase response for increasing values of the gain A. Note that the antiresonance valley continues to appear in the closed-loop response and will degrade the servo performance until A--and the bandwidth--is made extremely large. This means that in a practical system we should expect poor disturbance rejection near the locked-rotor frequency. Neither the antiresonance nor the resonance shown in the open-loop response (Figure 4) has limited the bandwidth in this case.

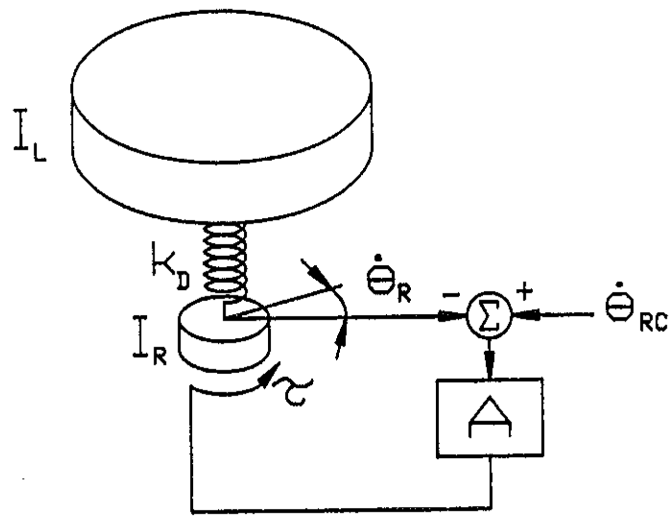


Figure 2.
Sketch of the model system showing a closed tachometer loop.

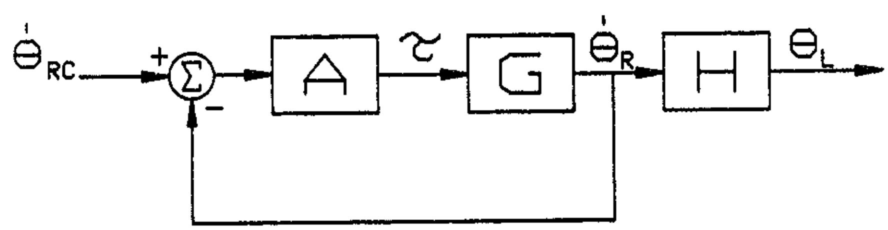


Figure 3.
Block diagram corresponding to Figure 2 showing a closed tachometer loop.

<u>STEP TWO.</u>
<u>OUTER-LOOP DYNAMICS WITH INNER-LOOP TACHOMETER FEEDBACK:</u>
<u>THE MOTOR SHAFT HAD BETTER BE STIFF OR ELSE...</u>

Figure 6 shows a sketch of the system which we will now analyze. This system includes the rate (tachometer) feedback from the previous step and also includes position feedback from the load. Figure 7 shows the corresponding block diagram. This block diagram is the same as that in Figure 3 with the addition of block "B" in the outer loop.

We turn our attention now to the open loop transfer function that takes the rate input to

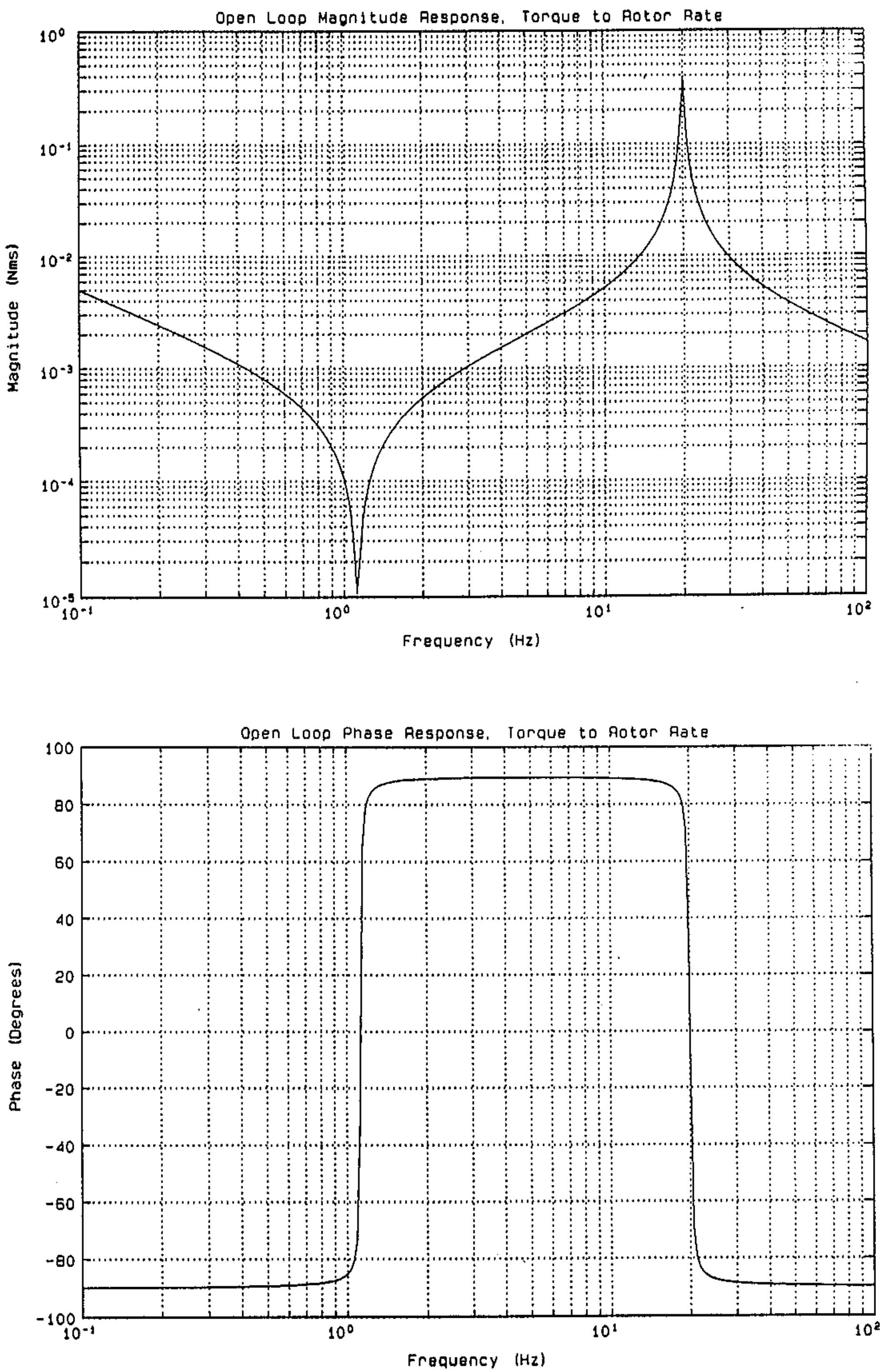
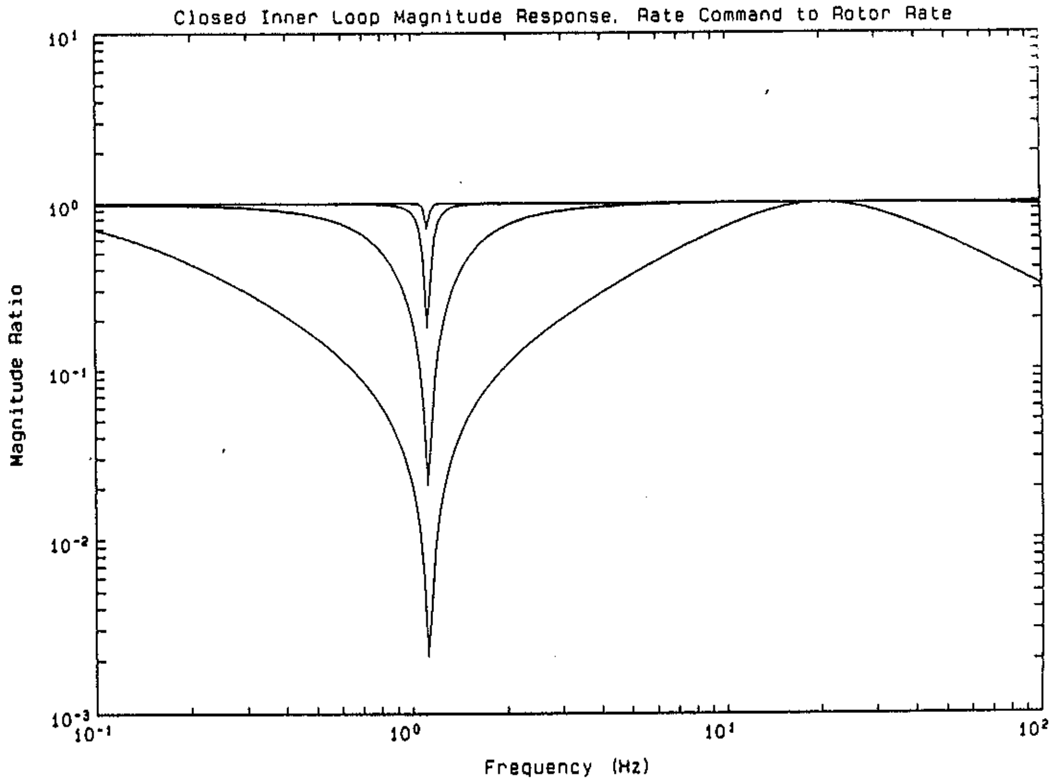


Figure 4.
Open-loop magnitude and phase response of the model system with torque at the rotor as the input and the rotor rate as the output. Note that there is an antiresonance near 1 Hz, the locked-rotor frequency, and a resonance near 20 Hz, the free-boundary frequency. Notice also that the phase response shows a region of phase lead and shows no sharp transitions to large negative phase angles.



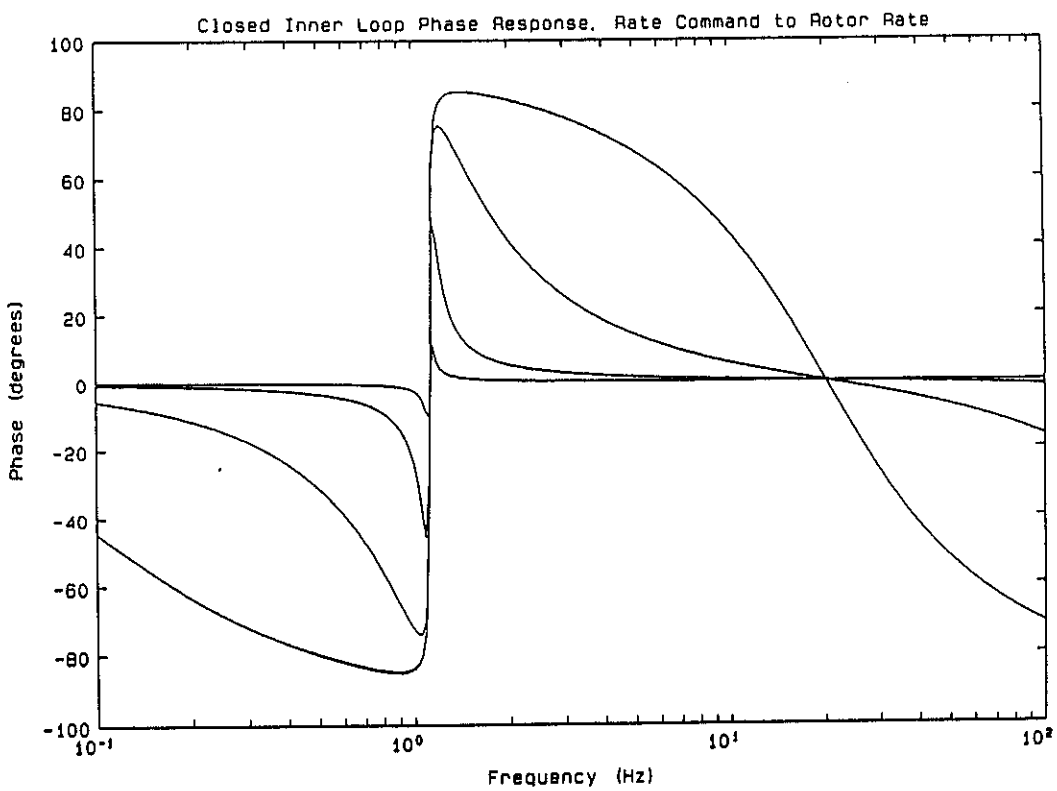


Figure 5.
Magnitude and phase response of the model system with the tachometer loop closed and with varying values of the loop gain. The input is the rotor rate command and the output is the rotor rate. Note that as the loop gain increases, both the magnitude and phase responses become flatter. The antiresonance that was present in the open-loop response (Figure 4) is also present here and will degrade the system disturbance rejection. At the highest gain, the antiresonance is nearly gone, but the closed-loop bandwidth is well beyond 100 Hz. This is unrealistic for most practical systems.

the second summing junction in Figure 7 to the resulting position at the load. This is the open-loop response of the outer loop with B=1, and is derived in Appendix A. The magnitude and phase response of this transfer function are shown in Figure 8. The notable--and annoying--features in these responses are the resonance peak at the locked-rotor frequency, and the accompanying steep phase transition from -90 to -270 degrees. Short of rather extreme measures, the frequency of the resonance is a strong limit on the bandwidth of the outer loop.

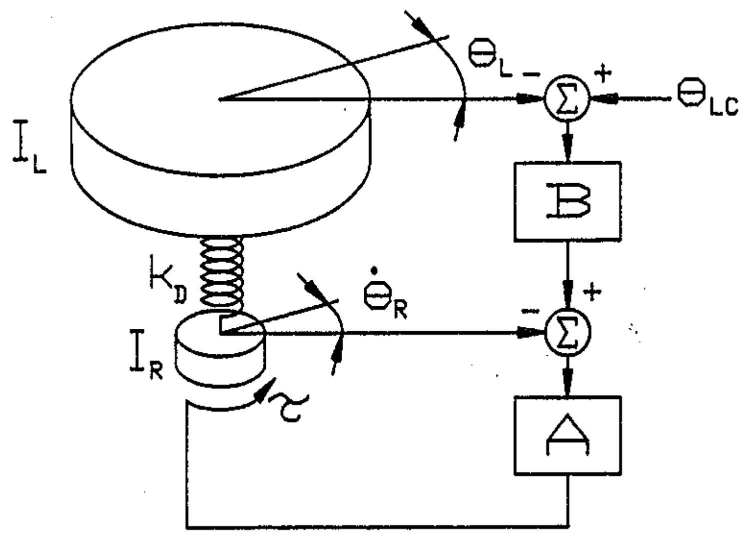


Figure 6.
Sketch of the model system with loops around both the rotor rate (the inner tachometer loop) and the load position (the outer loop).

We can estimate that the closed-loop bandwidth with a simple controller could be about a factor of ten below the frequency of the resonance. In reality, this value is dependent on the damping in the system (here assumed to be rather small, a conservative assumption). Referring to Figure 8, we see that the resonance is near 1 Hz, and so the closed loop bandwidth might be 0.1 Hz. It is probably possible to do better than this by maybe a factor of 2 with a more sophisticated controller, but a factor of 10 would be achievable only with much kicking and screaming.

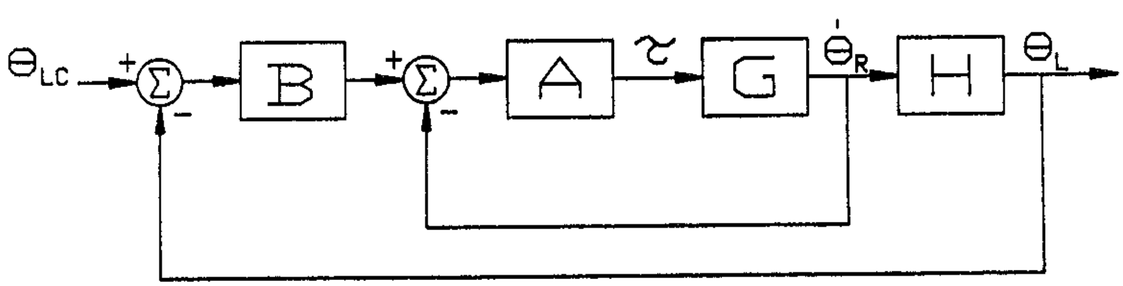
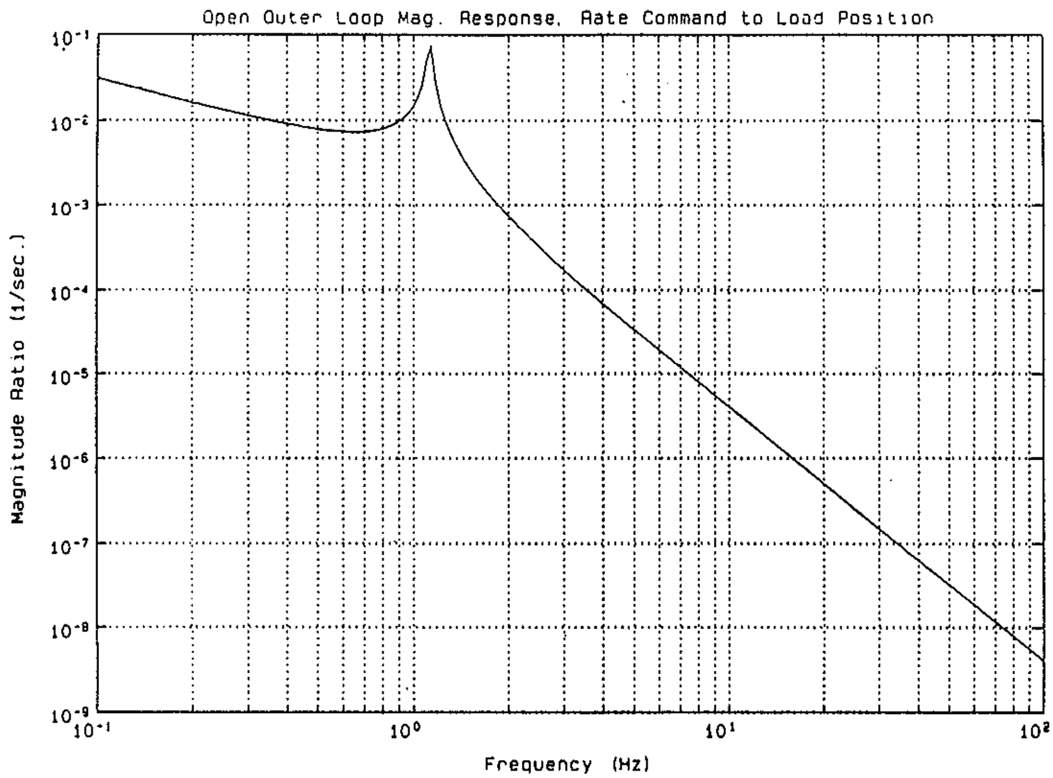


Figure 7.
Block diagram corresponding to Figure 6 showing both the inner tachometer loop and the outer position loop.



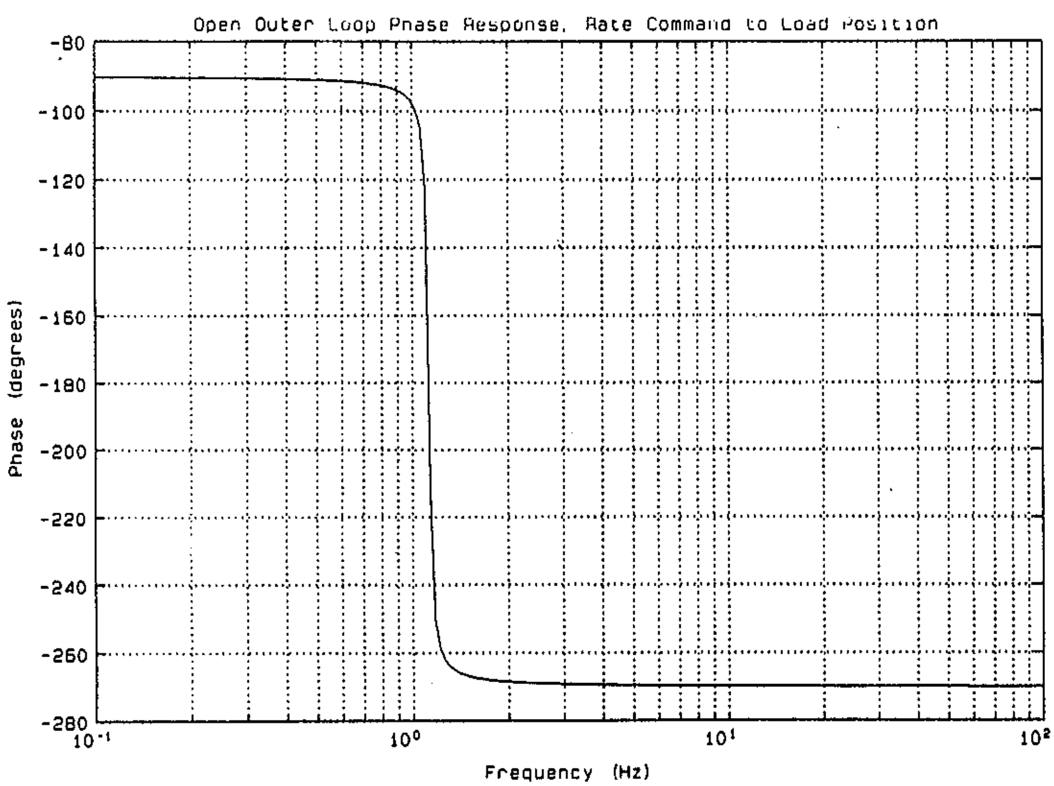


Figure 8.
Open outer loop/closed inner loop response of the model system. Note the resonance near 1 Hz. and the accompanying steep transition to lage negative phase angles. These features are obstacles to closed-loop bandwidths of much more than 0.1-0.2 Hz.

If this model system is taken to represent the simplified dynamics of a large telescope, a closed-loop bandwidth of 0.1 Hz is nothing to be proud of, and we would look for ways to improve the performance. Note that this bandwidth is governed (through the resonance peak) by the stiffness of the motor shaft. One approach, then, to raising the bandwidth is to simply make the shaft stiffer. This solution will work, but will be more or less attractive depending on the size of the load inertia. Generally, for small mechanical systems, stiffening the shaft is a feasible approach. For large systems, this approach can lead to components that are unrealistically oversized.

Fortunately, there is another solution.

STEP THREE.
SYSTEM DYNAMICS WITH ONLY AN OUTER LOOP: A MIRACLE OCCURS.

Figure 9 shows a sketch of the system with only rate feedback from the load, and Figure 10 is the corresponding block diagram. To investigate the dynamics of this system, we first consider the open-loop transfer function from torque at the motor rotor to rate at the load. This transfer function is derived in Appendix A, and the magnitude and phase responses are shown in Figure 11.

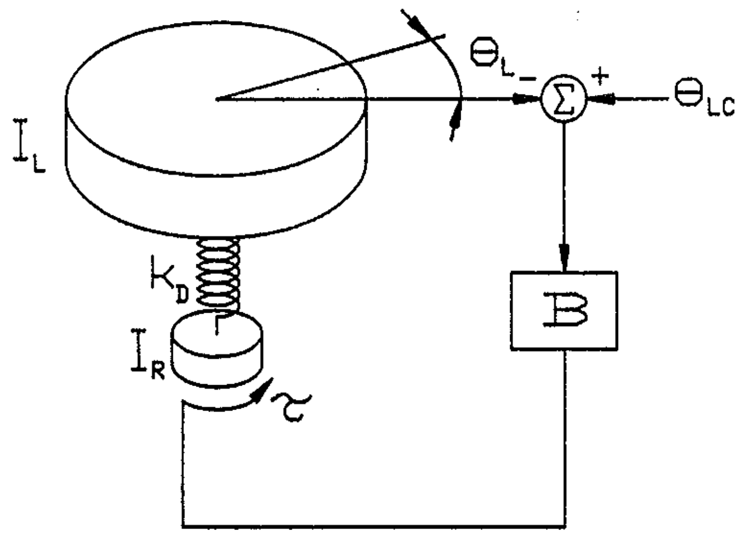


Figure 9.
Sketch of the system showing only a single closed loop around the load position.

As in the previous case with tachometer feedback, the frequency response shows a resonance peak with an associated transition to large negative phase angles. There is an important difference however that should be immediately obvious. This time, the resonance peak is at a much higher frequency, and the frequency is given by the free-boundary vibrational mode. This frequency is higher than any other frequency that the system can display. In that sense, this is the best you can do.

As before, this resonance is a bandwidth-limiting feature because of the steep transition to large negative phase angles. Also as before, we can estimate the achievable closed loop

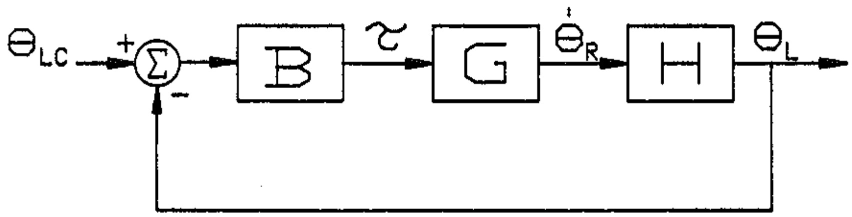


Figure 10.
Block diagram corresponding to Figure 9 showing the single loop around the load position.

bandwidth in the system by taking it to be a factor of ten below the resonance frequency. Referring to Figure 11, this indicates that a closed loop bandwidth of about 2 Hz should be possible.

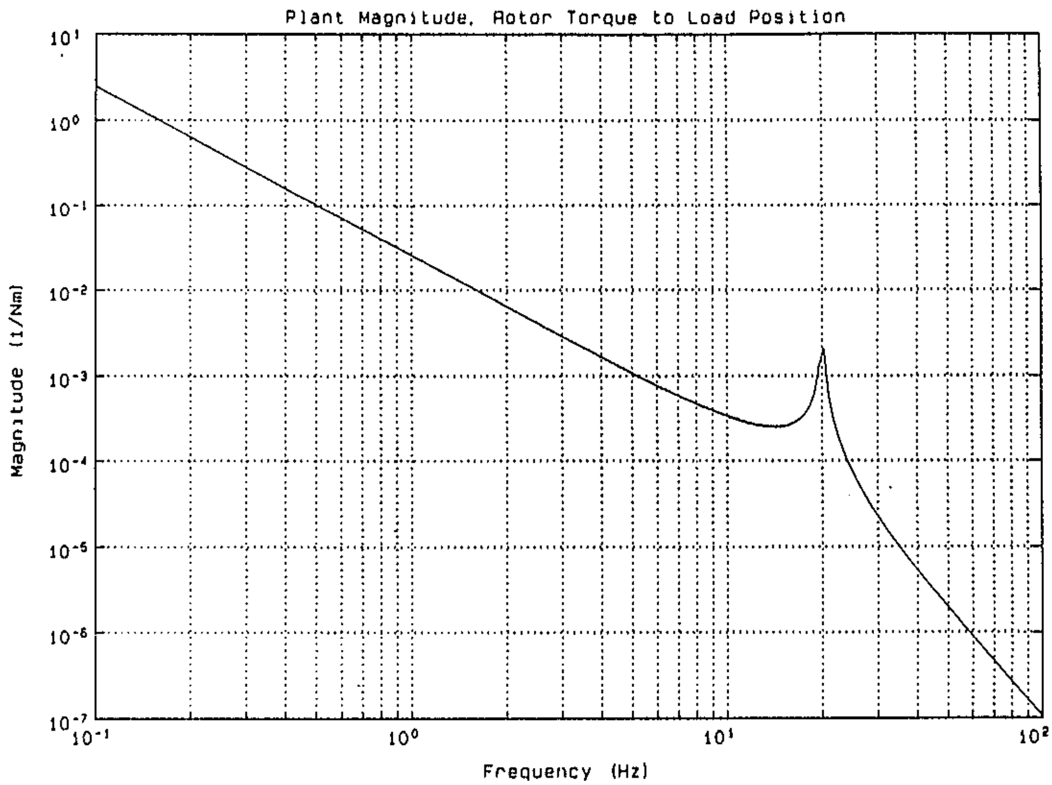
To verify this, Figure 12 shows the closed-loop magnitude and phase response for the system with a simple, practical compensator selected for the block B. Note that the actual bandwidth is about 3.2 Hz--a bit better than our estimate (but, then of course this is just an analysis, and I knew what the answer would be).

It should be noted that in this system, even better performance (in some regards, orders of magnitude better) is possible with a more sophisticated controller, but that is not the point of this report. The point here is that with a simple control system of the last example, we get performance is that could hardly be hoped for in the system considered in Step Two above. For the type of mechanical system considered in this report, this result indicates that tachometer feedback can be a source of serious trouble.

SUMMARY

Stabilizing an inner tachometer loop presents no special problem. Control of the outer position loop can be a difficult task however since the inner loop causes the locked rotor resonance to appear in the outer loop dynamics. This predicament can be avoided if the outer loop is stabilized directly using the motor torque as the input to the system and the load position as the output. In this case the tachometer is not needed.

The implications for telescope design are straightforward. If you can make the coupling between the drive motor and the telescope stiff enough, tachometer feedback as described above can give adequate performance and is easy to implement. "Stiff enough" means that the locked-rotor frequency is a factor of ten above the desired bandwidth. If the coupling is not stiff enough, abandon the tachometer and encode near the output of the drive system. In any case, wider bandwidth will be possible by encoding near the output.



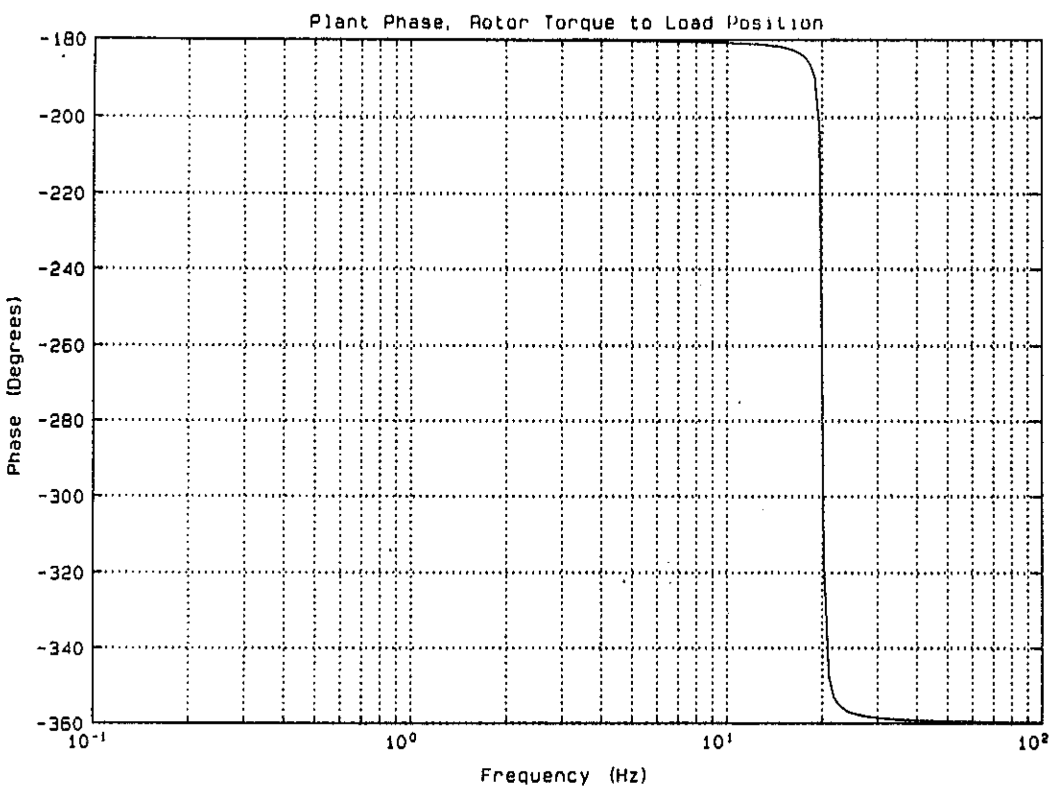


Figure 11.
Open loop magnitude and phase of the system of Figure 10. This system has only a single loop around the load position. Note that the resonance peak and phase transition occur at about 20 Hz, a relatively high frequency compared to the 1 Hz resonance in Figure 8.

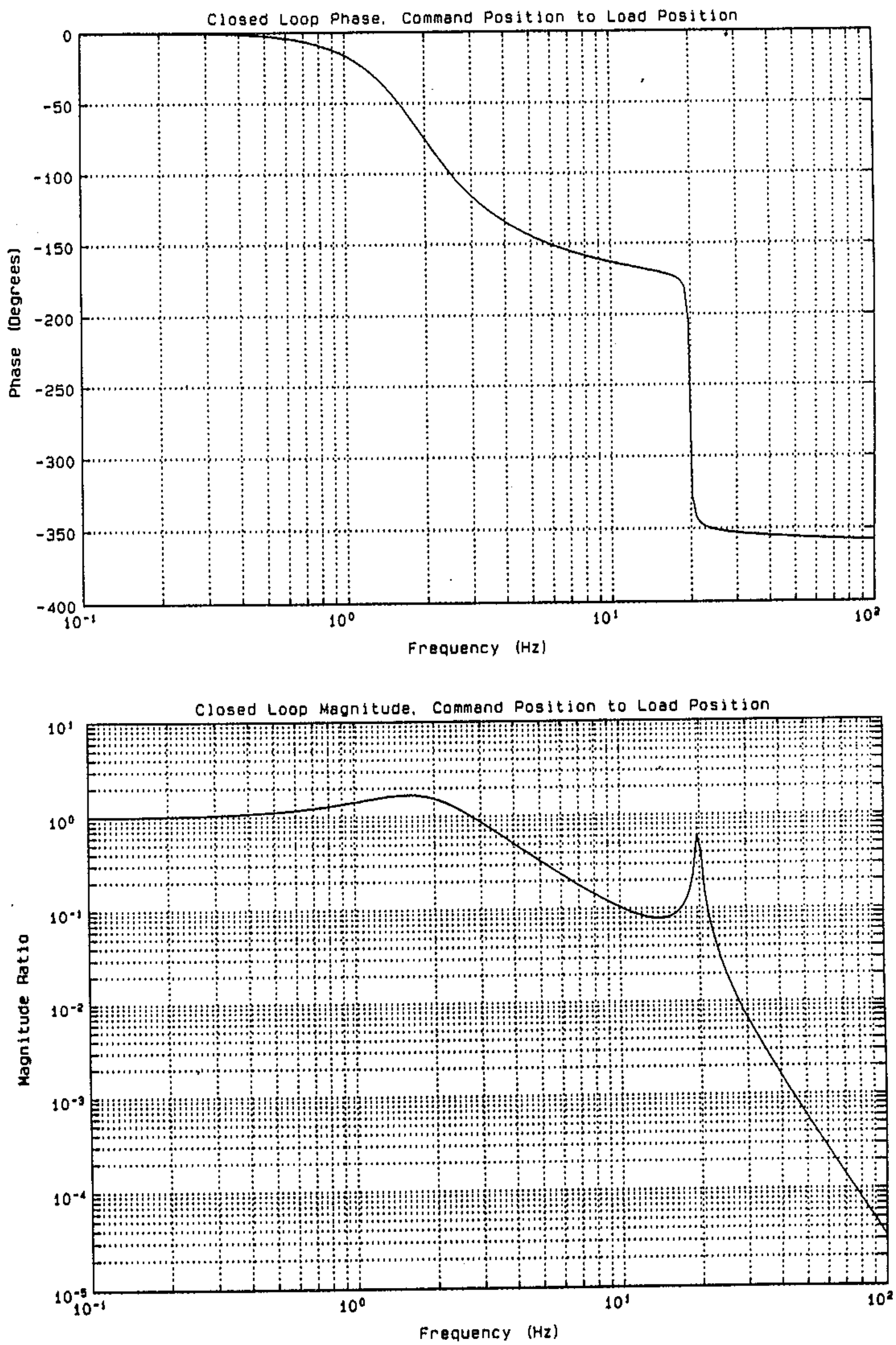


Figure 12
Closed-loop magnitude and phase of the single loop system. Note that the final bandwidth is about 3.2 Hz.

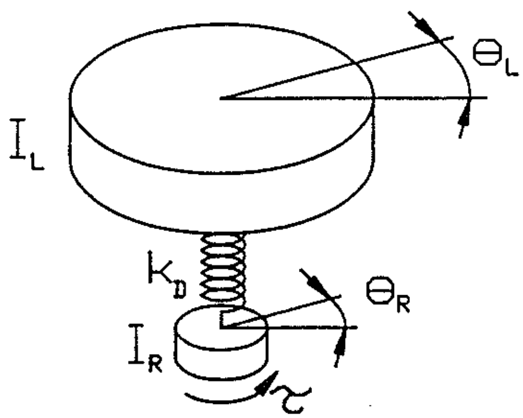
EPILOGUE

So why then does anyone use tachometer feedback at all? The answer is the simplicity with which such a system can be stabilized. The inner tachometer loop can be stabilized at very high bandwidths using only a proportional gain as the feedback compensator. In practice, you have only one pot to turn (or one constant to change, for the digital folks) when adjusting the system performance. Setting up this inner loop is nearly foolproof. The outer loop is just as easy to stabilize provided that the bandwidth is kept sufficiently far from the locked-rotor resonance. In smaller mechanical systems, this is usually an attractive solution. In larger systems, this solution may be unacceptable due to the excessively low bandwidth that is available. The determining factor is simply the frequency of the locked-rotor resonance; if this frequency is low, poor disturbance rejection and slow speed of response will be the results.

There is another thing that tachometers are good for, and it has to do with tailoring the output impedance of the actuator to suit the driven plant. The electrical analog is a well known topic in electrical engineering, but this strategy is somewhat unconventional in the field of feedback control of mechanical systems. Real gains can be made with this technique, but this should be the subject of another report.

APPENDIX A

DYNAMICS ANALYSIS



Equations of Motion:

$$\tau + k_D(\theta_L - \theta_R) - I_R \ddot{\theta}_R = 0 \qquad (1)$$

$$k_D(\theta_R - \theta_L) - I_L \ddot{\theta}_L = 0 \qquad (2)$$

with

$I_L = 316\ kg.m^2$
$I_R = 1\ kg.m^2$
$k_D = 15791\ Nm/rad$

Taking the Laplace transform of (2) and rearranging:

$$H = \frac{\theta_L}{\dot{\theta}_R} = \frac{k_D}{s(k_D + I_L s^2)} \qquad (3)$$

This transfer function shows a resonance at

$$s = \pm j\sqrt{\frac{k_D}{I_L}} = \pm j2\pi f_R$$

where f_R is the locked-rotor resonant frequency. Combining (3) and (1) in Laplace transform space yields

$$G = \frac{\dot{\theta}_R}{T} = \frac{k_D + I_L s^2}{s(I_R I_L s^2 + k_D(I_R + I_L))} \qquad (4)$$

This transfer function shows an antiresonance at

$$s=\pm j\sqrt{\frac{k_D}{I_L}}=\pm j2\pi f_A$$

where f_A is again the locked-rotor resonant frequency. The transfer function G also shows a resonance at

$$s=\pm j\sqrt{\frac{k_D(I_R+I_L)}{I_R I_L}}=\pm j2\pi f_R$$

where this time f_R is the free-boundary resonant frequency. Combining G and H into one transfer function, we find

$$GH=\frac{\theta_L}{T}=\frac{k_D}{s^2(I_R I_L s^2+k_D(I_R+I_L))} .$$

This transfer function shows a resonance at

$$s=\pm j\sqrt{\frac{k_D(I_R+I_L)}{I_R I_L}}=\pm j2\pi f_R$$

where again, f_R is the free-boundary resonant frequency.

APPENDIX B

PLOT GENERATING PROGRAMS

```
% Motenc.m
% This file runs in matlab.
% First bit calculates response for a two body system.  Torque is applied
% at Ir, and rate is measured at Ir.  Simulates the open loop response
% of a system with tach feedback on the motor shaft, a shaft compliance
% and a substantial load inertia.  Shows that crossover frequency of a rate
% loop is not necessarily limited by the shaft compliance.

%Calculate the model parameters.
Il=sqrt(10)*100; %Load inertia.
Ir=1; %Bottom inertia.
fr=20; %Frequency of bottom inertia on the shaft compliance.
kd=(2*pi*fr)^2*Ir; %Calculate shaft compliance.

%Calculate the poles and zeros.
obdtnum=[0 Il 0 kd]; %Open loop numerator.
obdtden=[Ir*Il 0 kd*(Ir+Il) 0 ]; %Open loop denominator.
zeros=roots(obdtnum);
poles=roots(obdtden);
zeros=zeros-(0.01*abs(zeros));
poles=poles-(0.01*abs(poles));
[A,B,C,D]=zp2ss(zeros,poles,1);
f=logspace(-1,2,250);

%Calculate the frequency response.
[mag0,phase0]=bode(A,B,C,D,1,2*pi*f);
loglog(f,mag0),grid
title('Open Loop Magnitude Response, Torque to Rotor Rate')
xlabel('Frequency (Hz)')
ylabel('Magnitude (Nms)')
pause
YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\motenc
end

semilogx(f,phase0),grid
title('Open Loop Phase Response, Torque to Rotor Rate')
xlabel('Frequency (Hz)')
ylabel('Phase (Degrees)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\motenc
end


%Calculate the closed inner loop response as a function of the loop gain.
gA=1/mag0(1);
obdocdnum=gA*obdtnum;
obdocdden=obdtden+gA*obdtnum;


otdocnum=gA*kd;
otdocden=[obdocdden 0];
zeros=roots(obdocdnum);
poles=roots(obdocdden);
zeros=zeros-(0.01*abs(zeros));
poles=poles-(0.01*abs(poles));
k=abs(deconv(conv(obdocdnum,poly(poles)),conv(obdocdden,poly(zeros))));
k=k(length(k));
[A,B,C,D]=zp2ss(zeros,poles,k);
[mag1,phase1]=bode(A,B,C,D,1,2*pi*f);
```

```
gA=10*gA;
obdocdnum=gA*obdtnum;
obdocdden=obdtden+gA*obdtnum;
otdocnum=gA*kd;
otdocden=[obdocdden 0];
zeros=roots(obdocdnum);
poles=roots(obdocdden);
zeros=zeros-(0.01*abs(zeros));
poles=poles-(0.01*abs(poles));
k=abs(deconv(conv(obdocdnum,poly(poles)),conv(obdocdden,poly(zeros))));
k=k(length(k));
[A,B,C,D]=zp2ss(zeros,poles,k);
[mag2,phase2]=bode(A,B,C,D,1,2*pi*f);


gA=10*gA;
obdocdnum=gA*obdtnum;
obdocdden=obdtden+gA*obdtnum;
otdocnum=gA*kd;
otdocden=[obdocdden 0];
zeros=roots(obdocdnum);
poles=roots(obdocdden);
zeros=zeros-(0.01*abs(zeros));
poles=poles-(0.01*abs(poles));
k=abs(deconv(conv(obdocdnum,poly(poles)),conv(obdocdden,poly(zeros))));
k=k(length(k));
[A,B,C,D]=zp2ss(zeros,poles,k);
[mag3,phase3]=bode(A,B,C,D,1,2*pi*f);

gA=10*gA;
obdocdnum=gA*obdtnum;
obdocdden=obdtden+gA*obdtnum;
otdocnum=gA*kd;
otdocden=[obdocdden 0];
zeros=roots(obdocdnum);
poles=roots(obdocdden);
zeros=zeros-(0.01*abs(zeros));
poles=poles-(0.01*abs(poles));
k=abs(deconv(conv(obdocdnum,poly(poles)),conv(obdocdden,poly(zeros))));
k=k(length(k));
[A,B,C,D]=zp2ss(zeros,poles,k);
[mag4,phase4]=bode(A,B,C,D,1,2*pi*f);

zeros=roots(otdocnum);
poles=roots(otdocden);
zeros=zeros-(0.01*abs(zeros));
poles=poles-(0.01*abs(poles));
[A,B,C,D]=zp2ss(zeros,poles,k);



fd=logspace(-1,2,200);
[magd,phased]=bode(A,B,C,D,1,2*pi*fd);




magn=[mag1 mag2 mag3 mag4];
phasen=[phase1 phase2 phase3 phase4];

loglog(f,magn,f(1),mag4(1)*2)
title('Closed Inner Loop Magnitude Response, Rate Command to Rotor Rate')
xlabel('Frequency (Hz)')
ylabel('Magnitude Ratio')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
        YN='N';
end

if YN~='N'
        meta d:\matlab\encpos\motenc
end
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```
semilogx(f,phasen)
title('Closed Inner Loop Phase Response, Rate Command to Rotor Rate')
xlabel('Frequency (Hz)')
ylabel('Phase (degrees)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
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end


magl= magd;
phasel= phased;

loglog(fd,magl), grid
title('Open Outer Loop Mag. Response, Rate Command to Load Position')
xlabel('Frequency (Hz)')
ylabel('Magnitude Ratio (1/sec.)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\motenc
end


semilogx(fd,phasel), grid


title('Open Outer Loop Phase Response, Rate Command to Load Position')
xlabel('Frequency (Hz)')
ylabel('Phase (degrees)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\motenc
end
```

```
% twobod.m
% This file runs in Matlab.
% Two body dynamics.  Torque in at one, position out at the other.
% Looks for model parameters.  If don't exist, runs motenc.m to
% establish them in the work space.

if ~(exist('kd') & exist('Il') & exist('Ir'))
        motenc2;
end

% Form the undamped plant numerator and denominator.
GHnum=[0 0 0 0 kd];
GHden=[Ir*Il 0 kd*(Ir+Il) 0 0];

% Set the zero, pole, and gain of the compensator.
% This set is a simple lead-lag.
cpole=-2*pi*4;
czero=-2*pi*1;
cgain=300;

% Zeros, poles, and gain for a compensator with a free integrator.
% Not used in the report.
% Permits zero DC error.
%cpole=[0 -2*pi*2];
%czero=[-1*pi*1 -1*pi*1];
%cgain=300/1.5;

% Damp the plant poles just a bit.
poles=roots(GHden);
poles=poles-(0.01*abs(poles));
GHden=poly(poles);

% Form the compensator numerator and denominator.
cnum=poly(czero);
cden=poly(cpole);

% Form the open loop numerator and denominator.
olnum=cgain*conv(cnum,GHnum);
olden=conv(cden,GHden);

% Form the closed loop numerator and denominator.
clnum=olnum;
clden=olnum+olden;

% Form the disturbance rejection numerator and denominator.
% Disturbance is a torque at the drive motor.
drden=clden;
drnum=conv(cden,GHnum);


% Plot the results.
f=logspace(-1,2,200);
[magp,phasep]=bode(GHnum,GHden,2*pi*f);
loglog(f,magp),grid
title('Plant Magnitude, Rotor Torque to Load Position')
xlabel('Frequency (Hz)')
ylabel('Magnitude (1/Nm)')
pause

YN=input('Save the plot? Y/N [N]:','s');


if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end
```

```
semilogx(f,phasep-360),grid
title('Plant Phase, Rotor Torque to Load Position')
xlabel('Frequency (Hz)')
ylabel('Phase (Degrees)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end


[magol,phaseol]=bode(olnum,olden,2*pi*f);
loglog(f,magol),grid
title('Open Loop Magnitude, Rotor Torque to Load Position')
xlabel('Frequency (Hz)')
ylabel('Magnitude (1/Nm)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end

semilogx(f,phaseol),grid
title('Open Loop Phase, Rotor Torque to Load Position')
xlabel('Frequency (Hz)')
ylabel('Phase (Degrees)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end


[magcl,phasecl]=bode(clnum,clden,2*pi*f);
loglog(f,magcl),grid
title('Closed Loop Magnitude, Command Position to Load Position')
xlabel('Frequency (Hz)')



ylabel('Magnitude Ratio')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end

semilogx(f,phasecl),grid
title('Closed Loop Phase, Command Position to Load Position')
xlabel('Frequency (Hz)')
ylabel('Phase (Degrees)')
pause
```

```
YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end


[magdr,phasedr]=bode(drnum,drden,2*pi*f);
loglog(f,magdr),grid
title('Disturbance Transmission Magnitude, Motor Torque to Load Pos.')
xlabel('Frequency (Hz)')
ylabel('Magnitude (1/Nm)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end

semilogx(f,phasedr),grid
title('Disturbance Transmission Phase, Motor Torque to Load Position')
xlabel('Frequency (Hz)')
ylabel('Phase (Degrees)')
pause

YN=input('Save the plot? Y/N [N]:','s');
if isempty(YN)
       YN='N';
end

if YN~='N'
       meta d:\matlab\encpos\twobod
end
```