Controlled damping of a 48-metre wide spray rig.

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Abstract

A method is investigated for damping the oscillation of a very wide agricultural spray rig. The present mechanical dampers couple the rolling disturbance of the vehicle into the tilt of the spray booms and an alternative technique is desirable. A method whereby the booms are driven in a manner analogous to a tightrope walker's balancing pole is shown to be effective.

Introduction

Any sway in a 48-metre wide spray rig leads to a risk that one of the booms will touch the ground or vegetation. The investigation was initiated by the owner of such a rig, who was dissatisfied with its performance. Attention was focussed on modifications that could be added to the existing rig to improve its stability. A previous attempt at control that relied on ultrasonic boom-tip height measurement had been seen to be ineffectual. The present technique is proposed as a solution and the steps to achieve a practical implementation are outlined. Experiments have been performed to measure the system's parameters to enable the simulation and strategy to be refined.

The central concept is that the booms will themselves provide the same means of balancing as a tightrope walker's pole. Hydraulic cylinders are already present for lifting the booms and by operating them separately a balancing action can be achieved. If one is raised while the other is lowered, the pivot-suspended central frame on which they are mounted is thrown to one side, causing a gravitational couple and hence a rotational acceleration. This allows any oscillation to be damped. Empirical simulation shows that just four sensors must be added, two of which might already be present. These are solid-state-gyroscopes that must be added to measure the angular velocity of each boom while angle-sensors measure the angle of each boom relative to the frame.

To describe the method in detail, it is necessary to establish the background.

Background.

The spray rig that is the subject of investigation was manufactured by 'Hayes Spraying'. An image and description can be found via the web reference (Hayes, 2010). The two spray arms are kept roughly horizontal by mounting them on a central frame that is pivoted about a point high on its centre. When deflected from the horizontal, the assembly swings and oscillates with a period of several seconds. Figure 1 shows a photograph.

Although the booms will settle to the horizontal when stationary, when in motion the swing will be excited by the jolting of the trailer until it builds up to a dangerous level. Some means must be found to damp the oscillation.

Figure 1. The spray rig.

In the existing system, a number of passive shock-absorber units have been connected between the swinging frame and the body of the trailer, shown in figure 2. Although these address the problem to some extent, they introduce a problem of their own.

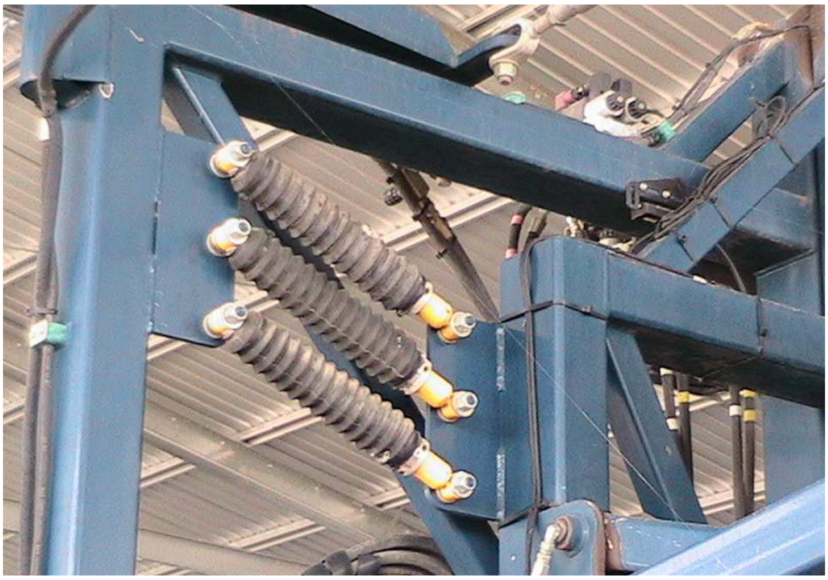

Figure 2. Shock absorber dampers.

When we consider the means whereby the jolting of the trailer disturbs the boom assembly, there are three clear sources. The first is by lateral disturbance of the pivot point as the trailer rocks or swerves. The large radius of gyration of the boom, relative to the height of the pivot above the centre of gravity, indicates that this will be relatively slight. Its exact sensitivity can be calculated from a measurement of the period of pendulum oscillation.

Second is the vertical disturbance of the pivot point. Apart from causing flexing of the boom arms, any effect on swing will be through parasitic amplification. It is also likely to be small.

Thirdly and most substantial is the rocking of the trailer about the axis of the pivot. At present this is coupled into the boom-swing through the shock absorbers provided for damping. It is already seen that this causes the problem at the root of the investigation. If the shock absorbers can be omitted, the source of disturbance can be greatly reduced and performance will be substantially improved.

Without the shock absorbers, some other means of damping must be found, leading to the concept of using the boom arms as a balancing pole.

Alternative techniques

By altering the point of suspension, the coupling of lateral pivot movement into sway can be modified. Video that is to be found at reference (Damman video 2010) appears to show a boom pivoted at its centre of gravity. As the trailer crosses 'Speed bumps' under alternate wheels it rocks violently, but there is no tilting of the rig. As the video nears its end, however, the rig appears to be starting a slow swing.

Bearing in mind the possible disturbance that can be caused by a damping actuator, it seems to be important that any damping force be applied in a compliant manner. Some researchers (Anthonis and Ramon 2003) have proposed the use of pneumatic actuators. This also seems to be the damping method used in the Dammann system, “The leveling with dual air cylinders has many advantages in hills, cornering and using slope correction.” (translation, Dammann 2003).

The method presented here is attractive in that it involves no additional actuators beyond those that are already in place.

The simple concept.

As shown in figure 3, each boom can be raised by a hydraulic cylinder. These can be driven in concert so that as one arm is raised, the other is lowered by an equal amount.

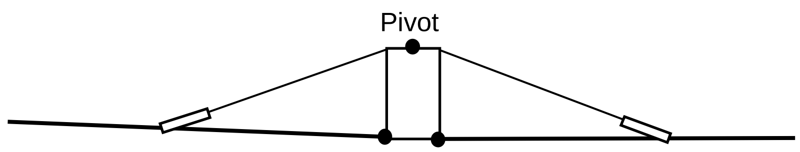

Figure 3. Boom geometry.

Until the boom starts to swing, this will not instantly result in any major change in the boom angle. Instead the assembly will be thrown to one side, as shown in figure 4. Here the pendulum action will cause a rotational acceleration that can be directed to damp any rotational velocity.

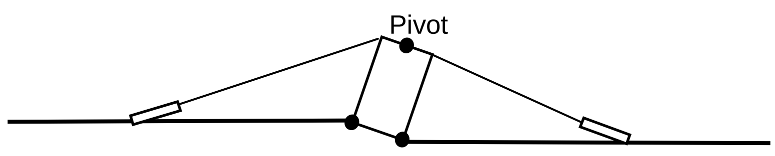

Figure 4. Lateral displacement of the centre of gravity.

A simple JavaScript simulation verified that such damping was possible, using a software framework described in (Billingsley 2009). A screen-shot of the response to an initial tilt is shown figure 5. The plot shows the frame angle, which is allowed to perform two cycles of oscillation before control is applied to the boom lift. The other trace shows the angle of the left boom, relative to the frame.

The next concern was the way in which the strategy could be implemented in practice.

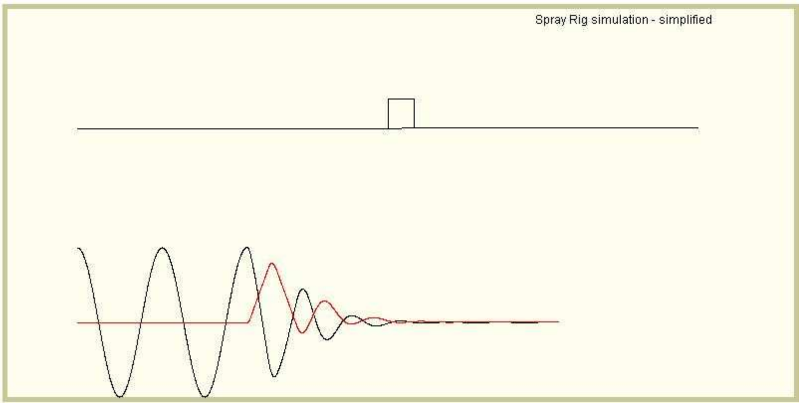

Figure 5. Simple simulation. Control is applied after two cycles of oscillation.

Practical details.

The hardware has more dynamics than the simple approach assumed. In order to limit the stresses induced by vertical disturbances, the cylinders that lift the boom arms are fitted with 'accumulators', chambers partly air-filled to give the same effect as a damped spring. Without these, there would be a much greater incidence of snapping in the chains that lift the booms and the stress in the boom at the lifting point could cause buckling.

Thus actuation of the valve does not cause an immediate change in the rate-of-change of boom angle, but is moderated by the spring-damper effect of the accumulator. Similarly it will not have the immediate complementary effect on the opposite boom assumed in the simple simulation. Instead the inertia of the central frame and the dynamics of both accumulators must be modelled.

Before any accurate simulation or control design could take place, readings had to be taken to determine the system parameters. The experiments are described after an analysis of the system's state variables.

Spray rig state variables and simulation.

From the suspension point onwards, there are three angles that must be considered. These are those of the left and right 'wings' and the central frame. They will all be measured anticlockwise from the horizontal and be represented by *tilt[0]* for the central frame and *tilt[1]* and *tilt[2]* for the left and right wings respectively. We can denote their velocities by *tiltrate[]*.

We will make a distinction between '*tilt*', relating to state variables measured from the horizontal, and '*ang*', angles between components that can be algebraic combinations of state variables.

The booms are each raised by a hydraulic cylinder. If at first we ignore the accumulators, the oil flow will define a 'demanded' boom angle *angdem[i]* relative to the frame. We will denote the magnitude of its rate-of-change as *urate,* where this will multiply the valve inputs *u[1]* and *u[2]* that will take values 0 or +/-1. The rate of change of *angdem[]* will thus be simply *urate*u[]*. (For now we can make an approximation that the demanded rate of lift is independent of the angle, rather than requiring a trigonometric solution.)

$$\frac{d}{dt} angdem[i] \; = \; urate*u[i]$$

The hydraulic actuators are fitted with 'accumulators', air cylinders that soften the response and limit the cable stresses. We can represent their coefficients in terms of the boom accelerations that will be caused when the boom angles differ from the 'demanded' values. The accelerating couple, implicitly multiplied by the moment of inertia of a boom arm, can be written as a combination of deviation from demanded tilt angle and demanded tilt rate.

$$tiltdem[i] \; = \; tilt[0] + angdem[i]$$

$$tiltratedem[i] \; = \; tiltrate[0] + urate*u[i]$$

$$couple[i] \; = \; k1*(tiltdem[i] - tilt[i]) + k2*(tiltratedem[i] - tiltrate[i])$$

where *k1* is the 'spring term' and *k2* provides damping, so that for *i*=1 and 2

$$\frac{d}{dt} tiltrate[i] \; = \; couple[i]$$

Important parameters of the system are the moments of inertia of the three sections and the pendulum acceleration when the centre of gravity is displaced. A further approximation that can be made in the early simulation is that when a boom is lifted, the rise in the centre of gravity of the assembly does not significantly change the pendulum time constant.

The pendulum effect will operate through the frame. The frame will be subject to the reaction to the couples that rotate the booms, plus a pendulum term proportional to its angular displacement. If its moment of inertia is *iframe* times the inertia of one boom, we have:

$$\frac{d}{dt}(tiltrate[0] = \; (-couple[1] - couple[2])/iframe - kpend*ang[0]$$

Experimental measurement of the parameters.

A number of simple experiments can enable the parameters to be measured. By adding solid-state gyros, the angular velocities of the left and right booms are logged at intervals of 0.1 seconds, which should be faster than any significant time constants. Linear accelerometers for measuring the boom tilts are not available at present, but are easy to procure if they are found to be necessary. The experimental procedure was as follows.

1. One or other boom-lift flow is switched on for several seconds. The difference between left and right boom rates will show a transient that steadies to a value of *urate*. From the response, *k1* and *k2* can also be estimated.
2. The assembly is tilted by pulling down on one 'wing'. It is then released, so that the pendulum time-constant and any remaining damping can be determined.

The value of *iframe* is hard to assess. It cannot be set to zero, because it is central to the simulation. However the error caused by a guessed value should not have a great effect.

A data-logging system was constructed, using an Asus ‘netbook’ computer that contained a solid-state ‘hard drive’. A ‘Labjack’ LJU3-HV interface was connected to the computer via a USB port and this allowed four analogue channels to be encoded, although only two were used.

The sensors used were CSR03 from Silicon Sensing, with a nominal sensitivity of 20 mV per degree per second.

Responses were logged by the owner of the spray rig on February 24th, 2010 and the computer was flown back to Toowoomba for analysis.

The first file, recorded at 8.15 a.m. and of 15 seconds duration, appeared to represent stationary sensors.

The second file, recorded at 8.34 a.m. represented some 15 minutes in duration and was a combination of several tests. These included applying steps of drive to the boom lift, swinging the frame assembly and driving the vehicle across the field.

Much has been learned from this data, both for performing more refined tests and for designing the final system.

It was seen that the logged values were quantised to increments of 5 mV, which would indicate ten-bit precision. The Labjack datasheet claimed two more bits, so perhaps the software can be refined.

However there was noise on the signals with a standard deviation of 8 mV that could act as ‘jitter’ to interpolate any smoothed signals. Since the analysis was based on angle, rather than angle rate, the integral of the signals made both noise and quantisation negligible. Of concern, however, was the signal datum, which was deduced as that value which would bring the integral back to zero.

Experimental results

First the sensors were calibrated by rotating each of them steadily through 90 degrees and back. The result is shown in figure 5.

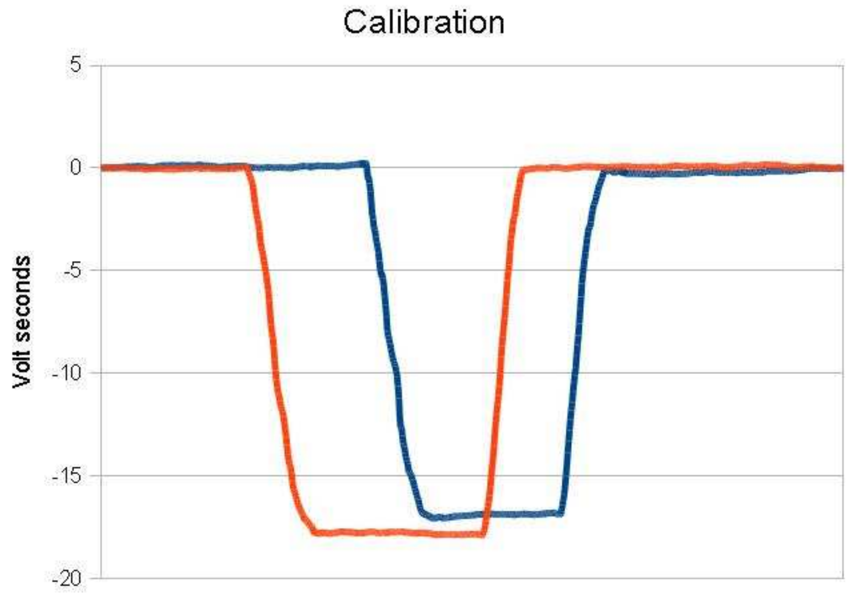

Figure 5. Calibration results.

From this it can be deduced that the gains are close to their nominal value, with the integrals for 90 degrees being 17 volt seconds and 17.8 volt seconds. When the lower gain is corrected to bring it into line, the signals represent 5 degrees per volt second.

For the step test, the difference of the gyro signals is taken so that swinging is eliminated. The results are shown in figures 6 and 7.

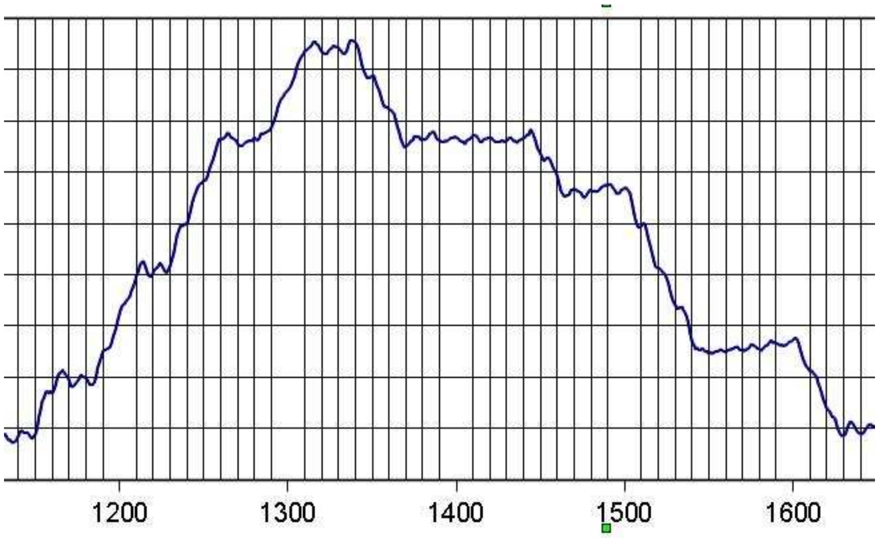

Figure 6. Changes of boom angle. Horizontal scale shows sample number, in increments of 0.1 seconds. Vertical gradations are at 0.5 volt second intervals or 2.5 degrees.

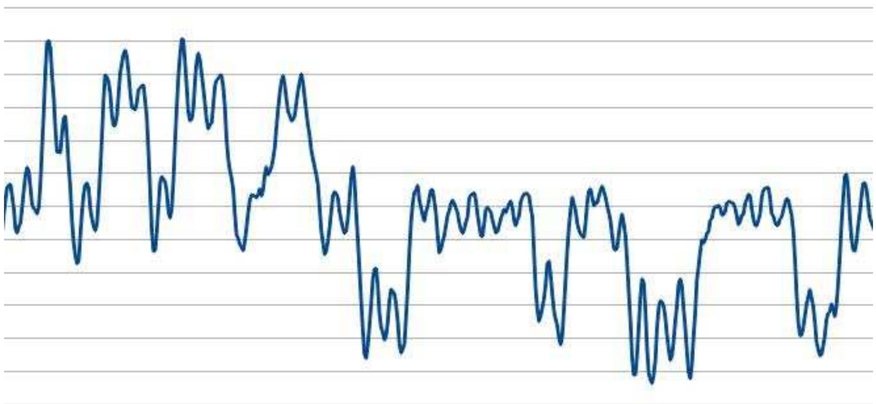

Figure 7. Difference of gyro signals, smoothed, gradations of 10mV or 0.5 degrees per second.

It was seen that there was an undamped 'bounce' with a period of a second. Because of its lack of damping it was hoped that this represented flexing of the booms rather than the accumulator response. By placing the sensors at a different location along the booms, this component of the signal would then possibly then be eliminated. From a video record taken on a previous visit, it appears more likely that the bounce is an undamped response of the lifting system. The parameters of the simulation must therefore be changed to explore ways in which the response can be damped, perhaps with the application of phase advance to any angle feedback.

The velocity signal was smoothed with a non-causal filter, equivalent to a continuous transfer function of $1/(1\text{-}0.1s^2)$. (Billingsley 2007) The conclusion is that *umax* = 1.5 degrees per second, that is .025 radians per second.

To assess the period of pendulum swing, the gyro signals were added together. The response of the integrated signal is shown in figure 8.

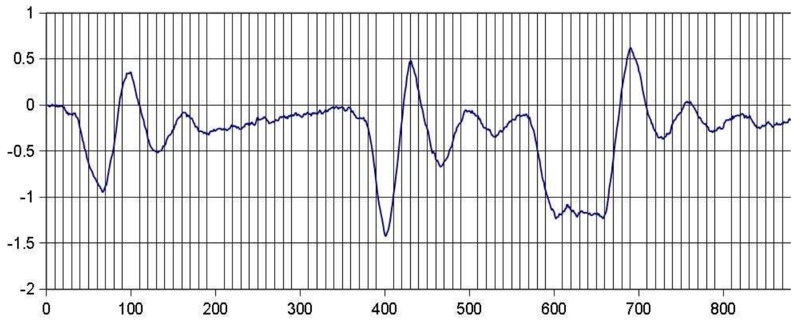

Figure 8. Pendulum response.

Clearly the shock-absorber dampers had not been removed, as would have been desirable. However the oscillation period is seen to be about 6.5 seconds.

The sensitivity of the system to lateral movement of the pivot can be calculated as follows. The period of oscillation is that of a pendulum of length 10 metres. The boom length is 24 metres from the centre. The vertical boom-tip excursions will therefore be 2.4 times the lateral movement of the pivot.

Simulation code

The corresponding simulation code is as follows. Because the accumulator time constants are relatively short, a short simulation time step must be taken.

```
for(i=1;i<=2;i++){
  angdem[i] += urate*u[i]*dt;
  tiltdem[i] = tilt[0]+angdem[i];
  tiltratedem[i] = tiltrate[0]+urate*u[i];
  couple[i] = k1*(tiltdem[i]-tilt[i])
  couple[i] += k2*(tiltratedem[i]-tiltrate[i]);
  tiltrate[i] += couple[i]*dt;
}
tiltrate[0] -= (couple[1]+couple[2]+kpend*tilt[0])*dt/iframe;

for(i=0;i<=2;i++){
  tilt[i] += tiltrate[i]*dt;
}
```

Now all that remains is to close the loop with a strategy for *u[1]* and *u[2]* and to draw the rig.

Outline strategy

There are several extra concerns of stability that have to be addressed. The boom-lifts will couple into the gyro signals of both the corresponding and the opposite booms via the dynamics of the accumulators, so that a number of loops with a short time-constant are involved. The gyro signals cannot be depended on to have the same sensitivity, so provision should be made for setting unequal gains.

The boom angles will have a downward limit, so constraints must be applied. With these and additional upward constraints, the control becomes highly nonlinear. Although mark-space modulation can be applied for fine control of the boom lift, the on-off valve limits will dominate any practical stability considerations. Simulation must be used to explore all the modes that can be excited, with responses tested from extreme disturbances.

Further nonlinearity is added by the accumulators. Although forces will be proportional to deflection for small values, there is a limit to the air volume and the force will be inversely proportional to the compressed volume.

Since the datum of the gyro signals is of such significance, much is to be gained in applying a high-pass filter to them. With a time constant of two seconds, this would not greatly increase the noise component. In addition it would give a further phase-advance to stabilise the feedback loop.

Consideration of the 'bounce' dynamics suggested that it might also be damped by application of the rate-gyro signal, high-pass filtered with a time constant not of two seconds but of half a second. Pragmatic adjustment of the simulation revealed that this could be successful, but also brought to light another mode of oscillation.

Since the moment of inertia of the frame is much less than that of the booms, when the shock-absorber dampers are removed there will be a 'shimmy mode' in which the booms only move slightly while the frame oscillates from side to side. This will have a period considerably shorter than the one-second bounce.

The following code can no doubt be tuned to give a faster response. A touch of angle feedback is added to bring the booms back to a neutral position. By adding the derivative of angle, the 'shimmy' could be damped more quickly, but does not seem to be of importance.

The result is shown in figure 7. The traces show the tip heights of the left and right booms and the sum of the angular momentum of the two booms. Initial conditions have been chosen that represent both 'sway' and 'bounce'.

```
//sensors and control

for(i=1;i<=2;i++){
  gyro[i]=tiltrate[i];
```

```
    angle[i]=tilt[i]-tilt[0];
    slowgyr[i]=slowgyr[i]+dgyr[i]*dt;
    dgyr[i]=(gyro[i]-slowgyr[i])*2; // half second TC
  }

  if(auto>0){

    u[1]=-100*(dgyr[1]) - 5*(angle[1]);
    u[2]=-100*(dgyr[2]) - 5*(angle[2]);
    if((angle[2]<0)&&(u[2]<0)){u[2]=0;}
    if((angle[2]>.1)&&(u[2]>0)){u[2]=0;}
    if((angle[1]<-0.1)&&(u[1]<0)){u[1]=0;}
    if((angle[1]>0)&&(u[1]>0)){u[1]=0;}

    if(angle[2]>.1){tilt[2]=tilt[0]+.1;}
    if(angle[1]<-0.1){tilt[1]=tilt[0]-.1;}
    if(angle[1]>0){tilt[1]=tilt[0];}


    for(i=1;i<=2;i++){
      if(u[i]>1){u[i]=1;}
      if(u[i]<-1){u[i]=-1;}
    }
  }
```

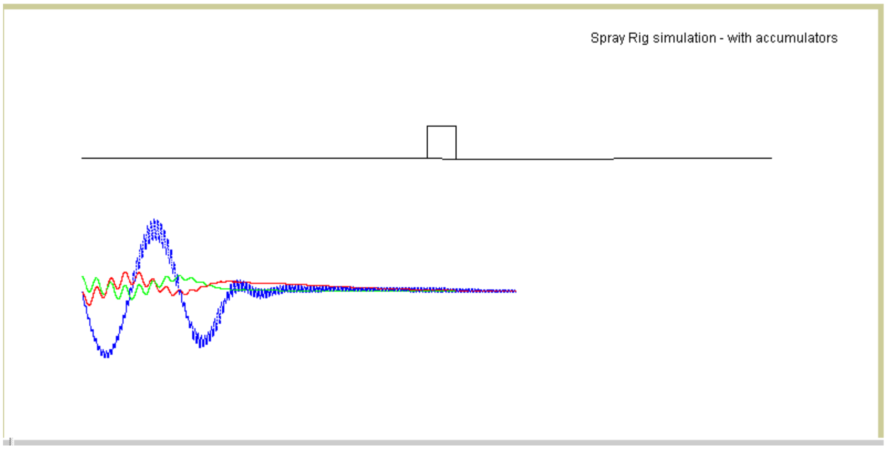

Figure 7. Screen grab of the simulation, representing 'sway', 'bounce' and 'shimmy'.

Conclusions

A simple control concept can require much greater complexity when it is to be applied to a real mechanical system. A 'drag and drop' simulation with inserted transfer function blocks will almost certainly give a misleading result. Nonlinear effects such a limits on the control actuators and required limits on the boom angles will mean that unexpected modes can be excited. The simulation must represent all imaginable interactions and responses must be examined with pragmatism in mind.

For this particular application, concerns still remain about the adequacy of the sensor gains and digitiser resolution. The benefits are clear of applying a high-pass filter to the gyro signals, both in the additional compensator dynamics and in removing the effect of the sensor datum.

It is hoped that the experiment can be performed soon of removing the shock absorbers and replacing them with on-line control of the boom-lift cylinders. This will depend on the good will of the owner of the rig. The adoption of the technique in production rigs will then depend on the extent to which the manufacturers can be convinced.

Acknowledgmens.

I am grateful to Gerrit Kurstjens for bringing such an interesting problem to my attention and for providing the facilities to carry out experiments.

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