5th Australasian Congress on Applied Mechanics, ACAM 2007
10-12 December 2007, Brisbane, Australia

Optimising slew torque on a mining dragline via a four degree of freedom dynamic model

Charles H. McInnes[1] and Paul A. Meehan[1]

[1] *CRC mining and the Department of Mechanical Engineering, The University of Queensland*

Abstract: A dynamic model of a dragline is developed in the form of a fundamental nonlinear rotating multibody system with energy dissipation. Its dynamic behaviour is investigated using measured field data. Model predictions of dynamic behaviour and stresses during operation are investigated and a comparison with measured data presented. Preliminary results from an investigation into reducing fatigue duty via improved slew torque control are also presented. The dynamics of the dragline bucket swing motion during house slewing (rotation) are of particular importance for both structural loading and efficient operation.

Keywords: dragline, dynamic model, fatigue, optimisation, productivity.

1 Introduction

A dragline is a large, powerful, rotating multibody system which operates in a similar manner to a crane. They are used as the primary means for removal of overburden in open-cut coal mining and are typically the bottleneck for productivity. Recent efforts to increase dragline payload have renewed interest in the tradeoff between short term productivity and fatigue damage inflicted on structural components of the dragline. With this in mind, a four degree of freedom dragline model has been developed.

The dragline model is described in section 2 and the four governing equations are presented and analysed. The process of tuning the model and a comparison of predicted and measured stress is given in section 3. The slew torque is then optimised manually to minimise duty while obtaining a similar bucket trajectory. The term duty is used in this paper to refer to estimated fatigue duty, with one unit of duty corresponding to the consumption of one ten-millionth of the estimated fatigue lifetime of the part on which the stress is measured [1]. This is based on the lifetime, measured as a number of cycles, of test specimens subjected to alternating stress. A number of algorithms are available for converting measured stress data to duty [2].

A simple, realistic torque profile is first assumed to give an indication of what could be achieved in practice. The model is then optimised allowing for more extreme variations in slew torque in order to demonstrate the contribution of out of plane angle to duty.

2 Dragline model

Figure 1 shows a diagram of the dragline model in the inertial reference frame XYZ. The origin O is at the intersection of the boom axis and the vertical slew rotation axis Y. The house and boom structure is modelled as a rigid body that rotates (slews) about the vertical axis with moment of inertia I_h. The bucket is modelled as a point of mass m suspended by the massless drag and hoist ropes d and h, which control the motion of the bucket within the bucket plane. The bucket position is defined by the four independent degrees of freedom: ϕ, θ, P and B. The slew angle ϕ is a measure of the rotation of the rigid house and boom assembly about the Y axis. The out of plane angle θ represents the angle between the vertical plane through the boom axis and the bucket plane. The location of the bucket within the bucket plane is measured along two orthogonal axes, B parallel to the boom axis and P perpendicular to this axis, which are alternatives to the typical measures, d and h.

For the purpose of modelling, only two damping terms and three forcing terms are considered. A viscous damping term c_t is applied to out of plane bucket motion. The three forcing terms are the slew torque M about the Y axis acting on the house and boom assembly and the resolved rope forces F_P and F_B acting between the bucket and the boom.

The coordinates B and P can be obtained from d, h, L, β and the boom length, where d and h are the lengths of the drag and hoist ropes, L is the horizontal distance from the Y axis to the intersection of the boom axis with the drag rope and β is the angle of inclination of the boom axis relative to the horizontal XZ plane.

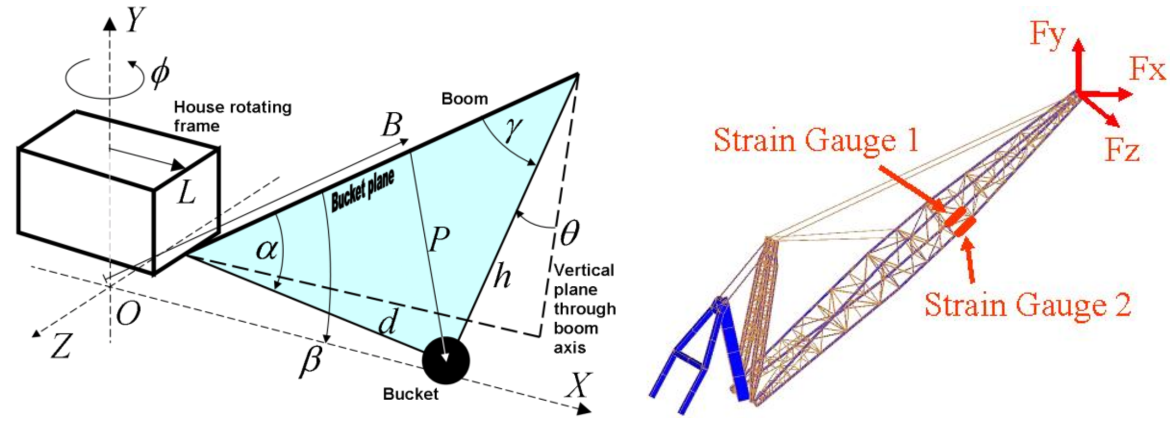

Figure 1. Dragline diagram, coordinate frames and strain gauge locations

The out of plane angle damping is given by:

$$D = \frac{1}{2} c_t \dot{\theta}^2 \tag{1}$$

There is similar damping on the slew motion, however this is introduced by modifying the applied torque M directly. There is no forcing term applied to the out of plane angle. The tension in the hoist and drag ropes is resolved, using trigonometry, into the forcing terms in the B and P directions, F_P and F_B. The dragline is assumed to operate on flat ground so that the slew axis coincides with the vertical Y axis. Thus the only potential energy term is the bucket height.

The equations of motion obtained from the Lagrangian analysis are given below. Each equation contains one forcing term (equal to zero for (3)). The four equations are coupled through several nonlinear terms resulting from the rotating multibody motion of the system.

$$M = \ddot{\phi}\underset{\sim}{I} + m\left(\begin{array}{l} \ddot{\theta}P\left(P\sin\beta + B\cos\theta\cos\beta\right) + \left(\ddot{P}B - \ddot{B}P\right)\sin\theta\cos\beta \\ +2\dot{\phi}\left(\dot{\theta}P\sin\theta\left(P\cos\theta - \underset{\sim}{D}\sin\beta\right) + \dot{P}\left(P\sin^2\theta + \underset{\sim}{D}\sin\beta\cos\theta\right) + \dot{B}\underset{\sim}{D}\cos\beta\right) \\ -\dot{\theta}^2 PB\sin\theta\cos\beta + 2\dot{\theta}\dot{P}\left(P\sin\beta + B\cos\theta\cos\beta\right) \end{array}\right), \tag{2}$$

$$m\left(\begin{array}{l} \ddot{\theta}P^2 + \ddot{\phi}P\left(P\sin\beta + B\cos\theta\cos\beta\right) + 2\dot{\theta}\dot{P}P + gP\cos\beta\sin\theta \\ -\dot{\phi}^2 P\sin\theta\left(P\cos\theta - \underset{\sim}{D}\sin\beta\right) + 2\dot{\phi}P\left(\dot{P}\sin\beta + \dot{B}\cos\theta\cos\beta\right) \end{array}\right) + c_t\dot{\theta} = 0, \tag{3}$$

$$m\left(\begin{array}{l} \ddot{P} + \ddot{\phi}B\sin\theta\cos\beta - \dot{\theta}^2 P + 2\dot{\phi}\left(\dot{B}\sin\theta\cos\beta - \dot{\theta}P\sin\beta\right) \\ -\dot{\phi}^2\left(P\sin^2\theta + \underset{\sim}{D}\cos\theta\sin\beta\right) - g\cos\beta\cos\theta \end{array}\right) = F_P, \tag{4}$$

$$m\left(\ddot{B} - \ddot{\phi}P\cos\beta\sin\theta - \dot{\phi}^2\underset{\sim}{D}\cos\beta - 2\dot{\phi}\cos\beta\left(\dot{P}\sin\theta + \dot{\theta}P\cos\theta\right) + g\sin\beta\right) = F_B, \tag{5}$$

where the underscore $\sim$ indicates a function of the degrees of freedom:

$$\underset{\sim}{D} = B\cos\beta + P\sin\beta\cos\theta\,, \tag{6}$$

represents the radial distance of the bucket from the slew axis, if the bucket position is projected onto the vertical plane through the boom axis and

$$\underset{\sim}{I} = I_h + m\left(\underset{\sim}{D}^2 + \left(P\sin\theta\right)^2\right) = I_h + m\underset{\sim}{r_Y}^2\,, \tag{7}$$

represents the instantaneous moment of inertia of the system about the Y axis, with $\underset{\sim}{r_Y}$ representing the radial distance of the bucket from the Y axis. Thus (2) can be considered the moment equation in the XZ plane, with the coefficient of m representing the moments arising from the Coriolis and other inertia forces with respect to O. The moment equation for the bucket through the angle θ is represented by (3), with the $\ddot{\theta}$, damping and gravity terms defining a damped inclined pendulum. The rest of the terms arise due to Coriolis and other inertia forces. The linear acceleration of the bucket along P and B are represented by (4) and (5). Each contains a forcing term, a linear acceleration term, a gravity term and several Coriolis and other inertia terms.

More detail on the model derivation and validation has been published previously [3].

The model predicted stress is based on the predicted hoist rope tension and geometry. The forces on the boomtip are resolved into vertical and horizontal forces within the vertical plane through the boom and an out of plane force. Referring to Figure 1, the sensitivities of predicted stresses σ_1 and σ_2 at the locations of strain gauges 1 and 2 respectively to boomtip forces, in units of Pa/N are:

$$\sigma_1 = -19.53Fx + 41.58Fy + 108.8Fz \qquad \sigma_2 = -19.53Fx + 41.58Fy - 108.8Fz\,. \tag{8}$$

These sensitivities were obtained from a finite element analysis [1]. Note that the components of the hoist rope tension within the vertical plane through the boom do not result in significant bending forces within the boom, due to the geometry of the boom's support ropes. However, out of plane forces do result in bending forces on the boom, resulting in significant sensitivity of stress (and hence duty) to the out of plane bucket motion.

Two values for bucket mass are used during each cycle, the measured gross bucket mass, which changes between cycles and the empty bucket mass. The mass is reduced via a linear ramp function towards the end of the swing cycle. Note that small amounts of dirt are lost from the mouth of the bucket through all or most of the swing phase.

3 Tuning and validation

The model described above was tuned and validated against measured data from three cycles of dragline operation. The measured data was obtained from a dragline in central Queensland in February 2005 [4]. The process of filling the bucket with overburden (fill phase) is not modelled because the bucket is not free to swing sideways during this phase and the hoist rope is not always taut. After filling the bucket, the house and boom structure is rotated to carry the suspended bucket off to one side (swing phase). The bucket load is then dumped onto the spoil pile and the empty bucket is swung back (return phase).

In order to tune the model, it is first run with the measured slew angle, measured rope lengths and initial out of plane angle as inputs. The necessary rope tensions and slew torque and the resulting boom stress were predicted by the model. Figure 2 shows the difference between the measured and the model predicted stress over the three cycles. The stress was measured on both sides of the boom via strain gauges, at a location determined via a finite element analysis to have the highest sensitivity of stress to bucket load [5]. The average stress and the difference between the two values are plotted. The phase times and measured stress are from an onboard duty monitor.

The average predicted stress, which is an indication of the modelled bucket weight, is consistently less than the measured stress at the beginning and end of the dump ramp function and greater than the measured stress during the middle of the dump. This indicates that a more ideal dump model would have an 'S' shaped profile.

In addition to the start and end of the dump, other tuning parameters were used to produce Figure 2. The damping term c_t was determined during a previous investigation with a simplified dragline model

[6]. The offset on the measured stress data was adjusted to match the model predicted data. The initial out of plane angle at the start of each swing phase and its first derivative were tuned to match the predicted stress difference to the measured difference over the first half of the swing phase.

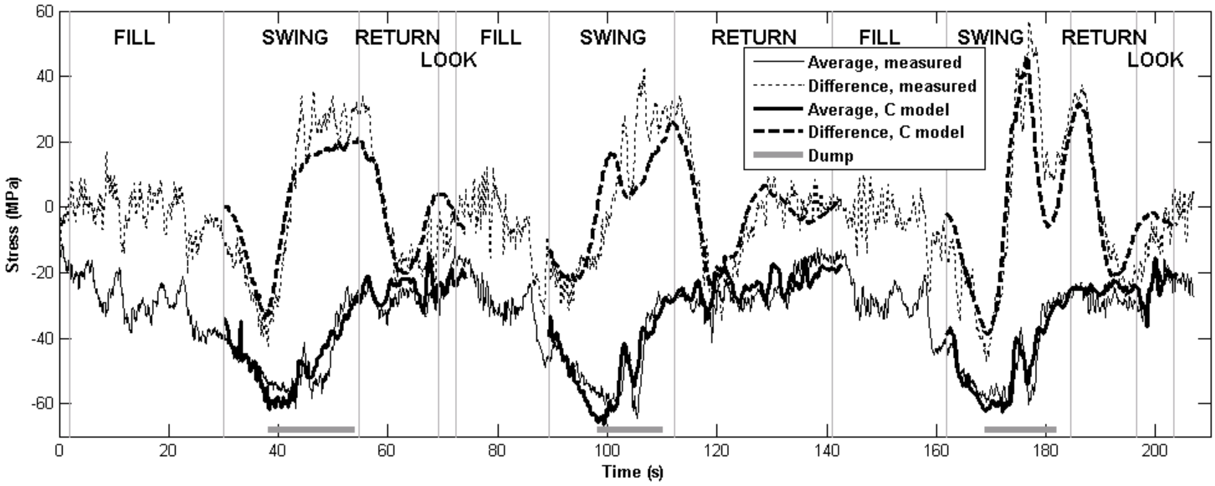

Figure 2 Measured and predicted stress

4 Optimisation

The model developed above will now be used to optimise slew torque over the swing, return and look phases of one cycle of operation. The purpose of this is to demonstrate the contribution of unnecessary out of plane angle to duty, as well as the significant potential to reduce duty and increase productivity that could be achieved using model predictive control. The original bucket location during the dump and at the end of the return phase will be used as a constraint. The slew torque will then be tuned manually to minimise duty. The slew torque profile will first be limited to ramp functions and the minimum possible number of torque reversals. The model is then retuned allowing for frequent step changes in torque. Note that the duty for a stress cycle exceeding the fatigue limit σ_e (40 MPa) is based on the overall stress range at each location:

$$duty = \left(\frac{\sigma_{range}}{\sigma_e} \right)^3 \tag{9}$$

The cycle chosen for analysis is the third cycle from Figure 2. It was chosen because it had the greatest variation in stress difference between opposite sides of the boom. This stress difference is caused by out of plane angle, which is highly sensitive to slew motion [7]. This cycle had an unusually high duty for the given bucket load of approximately 152 tonnes gross weight. The previous cycle had almost the same bucket load but significantly less duty (only 2.8 units compared to 7.7, based on measured stress at one of the gauges).

During tuning, the slew angle and rope lengths were inputs to the model, leaving the out of plane angle as the only degree of freedom modelled dynamically. Now the slew torque is input to the model instead of the slew angle, so that both the slew angle and the out of plane angle will be predicted by the model. The rope lengths remain inputs and are unchanged, so achieving the same angular bucket location ensures that the optimised case involves roughly the same productivity but with less duty incurred on the machine. The torque is adjusted to reduce predicted duty while keeping a similar bucket trajectory.

The angular location of the bucket relative to the vertical Y axis is given by:

$$\phi_{bucket} = \phi + \tan^{-1} \frac{P \sin\theta}{B\cos\beta + P\cos\theta \sin\beta} \tag{10}$$

4.1 Simple torque profile

The process of manually optimising slew torque first assumed a simple torque profile, shown in Figure 3a. A maximum torque of 35 MNm was assumed, which is less than the maximum model predicted torque based on measured data (labelled ‘original’). By adjusting the timing of the minimum and two maximum torques, predicted duty at the two strain gauge locations was reduced from 4.3 and 6.0 to 2.5 and 4.2 respectively.

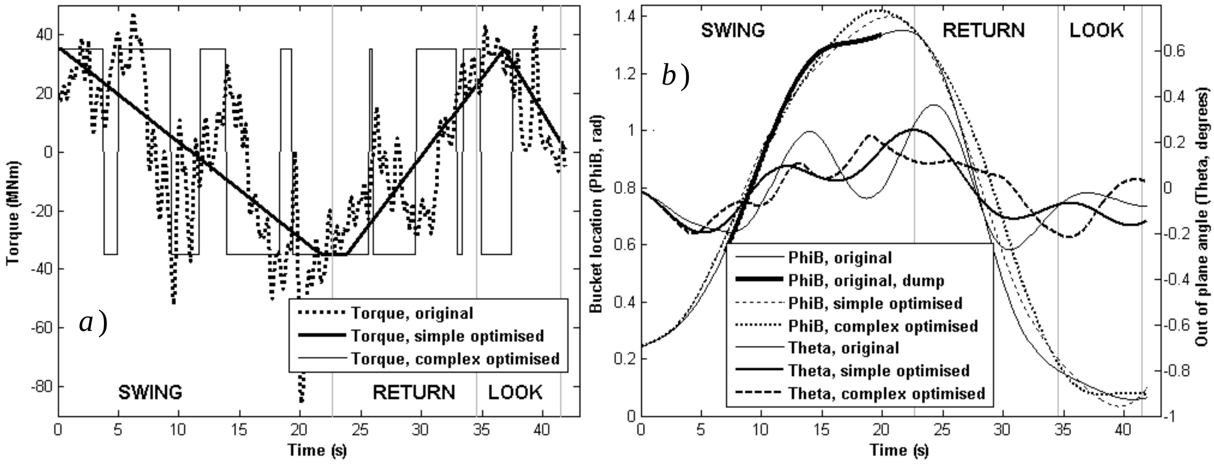

Figure 3a) Optimised torque profiles and b) bucket trajectories

The predicted stress profiles at each strain gauge are shown in Figure 4. The large out of plane angle caused both the maximum and minimum stress for the cycle under consideration to occur during the swing or return phase. The bucket locations and out of plane angles are shown in Figure 3b.

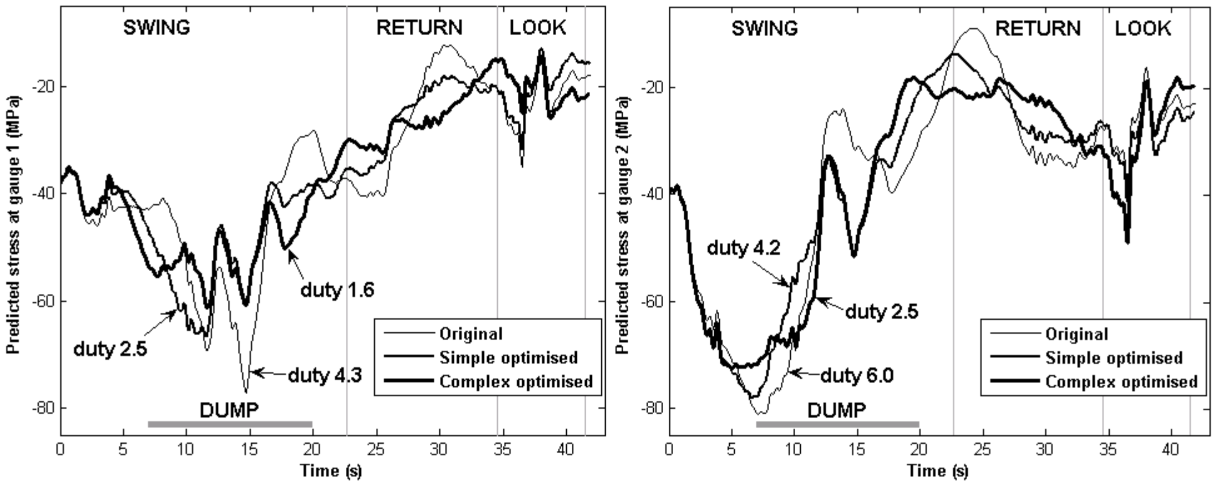

Figure 4 Change in predicted stress at gauge 1 and 2 resulting from optimisation

Note that the new bucket angular position leads the original position at the start of the swing phase. This could cause the bucket to strike the spoil pile. It is unknown whether the spoil pile location added an additional constraint to the bucket motion for this cycle. The operator temporarily reversed the slew torque at roughly ten seconds into the swing phase, which could have been intended to reduce out of plane angle or avoid the spoil pile. The change in slew torque initiated significant out of plane bucket motion during the rest of the swing phase and the beginning of the return phase. If the spoil pile were a constraint, the operator would have incurred less duty by waiting until the bucket had been hoisted further before beginning the slew motion, so that the reversal in slew torque was not necessary.

4.2 Complex torque profile

The case outlined above with the simple torque profile achieved roughly a one third reduction in duty by altering the slew torque and hence the out of plane angle, but keeping the in plane bucket motion similar. This indicates that a lot of the duty incurred during the cycle is caused by unnecessary out of plane motion. To further reinforce this point, the slew torque was again optimised allowing for a more complicated slew torque profile switching between a maximum and minimum of ±35 MNm. The ‘reversals’ in the slew torque (see Figure 3a) were timed to place an effective upper and lower limit on the stress at each gauge. This torque profile is unrealistic and would cause excessive dynamic vibrations and wear in other parts of the machine. However it does demonstrate the significant contribution to duty from out of plane angle. Under this new torque profile, the duty was further reduced from 2.5 and 4.2 down to 1.6 and 2.5.

Note that the range of the average predicted stress is 47 Mpa, which would alone cause 1.7 units of duty. This represents an effective minimum duty for the given bucket mass and in plane bucket motion. The stress range caused by hanging a full bucket from the boomtip under static conditions is 62 MPa, with an associated duty of 3.8. The stress range caused by hanging a full, then empty bucket from the boomtip is 37 MPa, which causes no duty as it is below the fatigue threshold of 40 MPa. The fatigue threshold corresponds to one unit of duty. All three values (0, 1.7, 3.8) represent a theoretical minimum duty, depending on the assumptions used. It appears that the bucket was not dropped on the ground during the fill phase, which is typical for a deep digging situation. Thus the theoretical minimum of zero units of duty is more appropriate if hoist rope tension could also be optimised, while 1.7 is more appropriate for the scenario above of optimising only the slew torque.

5 Conclusions and recommendations

A four degree of freedom dynamic dragline model has been presented. Reasonable agreement was achieved between predicted stress and measured stress over three cycles using a limited number of tuning parameters. A single cycle with high duty was chosen for further analysis involving manual tuning of the slew torque profile. As expected, significant reductions in duty were achieved.

The ‘complex’ optimised torque profile represents the limit of what could be achieved with a more complicated, but unrealistic torque profile under the given constraints.

The manual optimisation procedure limited the number of variables that could be optimised. The rope lengths were kept as inputs and the bucket trajectory was kept similar to ensure the resulting dragline motion was realistic. This ‘unnecessary’ constraint limited the scope to minimising duty. However, by using numerical optimisation methods and more appropriate constraints, it is anticipated that the model could be used to find torque and rope tension profiles that maximise long term productivity, taking cycle time, bucket mass and the downtime associated with duty into account.

6 Acknowledgements

This research would not have been possible without funding from the Cooperative Research Centre for Mining and The University of Queensland Department of Mechanical Engineering.

References

[1] Meehan PA, Guan Z, Gurgenci H. Dragline dutymeter report. Australian Coal Association Research Program report no. C9034, 2003.

[2] McInnes CH, Meehan PA. Equivalence of four-point and three-point rainflow cycle counting algorithms. International Journal of Fatigue, in print.

[3] McInnes CH, Meehan PA. A four degree of freedom dynamic dragline model for predicting duty and optimising bucket trajectory. in CRC Mining Technology Conference. 2007. Perth, WA, Australia: The Australasian Institute of Mining and Metallurgy.

[4] McInnes CH, Meehan PA, McDonald A. Advanced duty detection system. Australian Coal Association Research Program report no. C12038, 2005.

[5] Townson P, Murthy P, Guan Z, Gurgenci H. Optimisation of design load levels for dragline buckets. Australian Coal Association Research Program report no. C7003, 2001.

[6] Meehan PA, Austin KJ. Prediction of chaotic instabilities in a dragline bucket swing. International Journal of Non-Linear Mechanics 2006;41(2):304-12.

[7] McInnes CH, Meehan PA. Reducing dragline fatigue via improved operator feedback. in CRC Mining Technology Conference. 2005. Fremantle, WA, Australia: The Australasian Institute of Mining and Metallurgy.