This is a repository copy of *Load Control of a 300 hp Tunneling Machine*.

White Rose Research Online URL for this paper:
http://eprints.whiterose.ac.uk/76374/

---

**Monograph:**

---

**Reuse**
Unless indicated otherwise, fulltext items are protected by copyright with all rights reserved. The copyright exception in section 29 of the Copyright, Designs and Patents Act 1988 allows the making of a single copy solely for the purpose of non-commercial research or private study within the limits of fair dealing. The publisher or other rights-holder may allow further reproduction and re-use of this version - refer to the White Rose Research Online record for this item. Where records identify the publisher as the copyright holder, users can verify any specific terms of use on the publisher’s website.

**Takedown**
If you consider content in White Rose Research Online to be in breach of UK law, please notify us by emailing eprints@whiterose.ac.uk including the URL of the record and the reason for the withdrawal request.
LOAD-CONTROL OF A 300 H.P. TUNNELLING MACHINE

by

J.B. Edwards

Department of Control Engineering,
University of Sheffield,
Mappin Street, Sheffield S1 3JD.

Research Report No. 220

April 1983

N.C.B. Research Contract Ref. No. 226072
LOAD-CONTROL OF A 300 h.p. TUNNELLING MACHINE

by

J.B. Edwards

Department of Control Engineering,
University of Sheffield,
Mappin Street, Sheffield S1 3JD.

Research Report No. 220

April 1983

N.C.B. Research Contract Ref. No. 226072
SUMMARY

In response to a request from the National Coal Board, a mathematical model is derived for the rock-cutting operation of a 300 h.p. tunnelling machine currently in use at Cadley Hill Colliery. The existing over-riding load control system is shown to produce oscillatory load patterns when cutting harder rock that accord closely with preliminary observations underground.

The model is used to design and confirm the performance of an improved load control system using derivative power feedback. The necessary modifications to the existing control amplifier should be fairly straightforward.
1. INTRODUCTION

1.1 Machine Description and Operation

The rock tunnelling machine under investigation and presently operating at Gadley Hill Colliery is illustrated diagrammatically in Fig. 1. The machine is of the boom type (rather than the alternative full-face-cutter type) and excavates a tunnel of circular cross-section by the following sequence of operations:-

(a) With the boom in a central position, the rotating cutting head is sumped into the rock to its full depth (0.75m) by hydraulic pushing rams.

(b) Under the action of the boom-tilting jacks shown, the boom is moved outwards to a radius R equal to about 40% that of the finished tunnel.

(c) The whole boom assembly is now rotated at a peripheral speed, v, to complete an annular cut of mean radius, R.

(d) Radius R is now increased to a value close to that of the finished tunnel (5.5m) and a second annular cut made in the reverse direction so as to avoid progressive twisting of the power cable feeding the cutting-head motor.

(e) The sequence (a) to (e) is repeated following a second sumping operation before it becomes necessary to advance the machine as a whole. (The machine is not shown in its entirety in Fig. 1 but, behind the boom assembly, the machine carries an elaborate system for installing segmental concrete ring supports for the tunnel which are installed in parallel with the cutting operations).

1.2 Gearbox Failures

Over several months operation, a number of failures of the epicyclic reduction gear-box (1500: 6r.p.m.) via which the cutting head
is driven from a 300 h.p. squirrel cage induction motor (Fig. 1) have occurred requiring several enormously expensive replacements. This report is the first of a series outlining an investigation being carried out at Sheffield into the likely causes of these failures and possible means for their avoidance in future.

1.3 Existing Load Control

The existing load control system is of an overriding, or limiting type: a preset reference value $v_r$ for rotational speed $v$ is compared with a potentiometer measurement of the angle of the swash plate of the variable delivery pump feeding the boom-rotating drive motor and the error signal used to deflect the spool of an electrohydraulic servo-valve which directs oil, at system pressure, to the appropriate swash tilting cylinder. A speed $v_r$ very nearly equal to $v_r$, is thus attained very rapidly provided the load $P_1$ on the cutting motor, produced by $v_r$ and as measured by a power transducer, does not exceed a load reference setting $P_r$ (ideally = the 300 h.p. rating of the motor). Should $P_1 - P_r$ become positive (i.e. load exceed reference) as detected by a diode in the feedback of the comparator amplifier, then a preset proportion of this error is applied to back off the servo valve signal thus reducing pump delivery and hence speed $v$.

Detailed recordings are awaited, but the system is observed to oscillate in practice when $v_r$, the so-called rate reference, is set high and the machine is cutting rock rather than coal. The load oscillations observed on the motor monitoring the transducer are not excessive (say $\pm 20\% P_r$) but the swash angle and speed vary are observed to fluctuate between zero and values well in excess of the average. The observed frequency of oscillation is one cycle in about 3.0 seconds.

Complete stabilisation of the load control system is therefore an obvious objective that should be sought as a means of controlling
more precisely the average power consumption of the motor and hence the average load transmitted by the gear-box.

1.4 **Additional Causes of Load Fluctuation**

Although not apparent on the load meter, further components of load oscillation at much higher frequencies of the order of 10 Hertz are perceptible through the obvious vibration that occurs when cutting hard material. These are the result of chatter resulting from interaction of the cutting dynamics and the stiffness dynamics of the boom-structure, boom-drive and cutting-head-drive. A fuller study of these vibrations is necessary to ascertain their amplitude, and therefore their seriousness both on the machine as it now exists and on a modified control system whose behaviour could be degraded by the chatter. This will be the topic of subsequent analysis in a future report. The present report confines attention to improved control of the chatter-free machine.

A further cause of load variation is the heap of cut material that accumulates at the bottom of the cut face prior to its discharge via the scraper chain conveyor running beneath the boom and machine body (not shown in Fig. 1). As the boom ploughs through this heap at the bottom of the cut, particularly on its second sweep, a large additional drag is imposed by the recirculated material. No attempt is made in this first report to model this phenomenon and the following analysis is therefore confined to cutting operations made above this heap, i.e. through the top 270° of boom travel.

2. **MODELLING THE SYSTEM**

The following model is developed from ideas first published by the author in References (1) (2) and (3) (section 6). The numerical values used in the preliminary model are 'first estimates' and will require some adjustment in future as more complete data becomes available.
2.1 The Cutting Dynamics

These dynamics relate the boom speed, \( v \) to the power \( P_m \) supplied by the gear-box under investigation to the cutting head shaft. The 'bite' \( y \) made by each spiral line of cutting picks (Fig. 1) will be described by

\[
\frac{dy}{dt} = v(t) - v(t-T)
\]

where \( T \) is the time delay between the arrival of successive picks (in the same plane). Given a cutting head speed of 6 r.p.m. and that there are 6 spiral starts per revolution, \( T \) works out to be \( 60/(6 \times 6) = 1.67 \) s but a rounded value of 1.8 s is used here, partly for convenience of simulation and partly to allow for some nominal slip in the induction motor since the 6 r.p.m. corresponds to 1500 r.p.m. synchronous speed.

For simplicity it is here assumed that the cutting load \( P_m \) is proportional to bite \( y \),

\[
\text{i.e. } P_m = k_h y
\]

the constant of proportionality \( k_h \) depending on \( R \) and on pick sharpness and the hardness of the rock being cut. A nominal value for \( k_h \) has been determined as follows:

A brief inspection of the machine cutting operations indicated the uninterrupted boom revolution time to be about 18 min. when operating at a load \( P_m \) of 160 kw (\( \approx \) to \( P_e = 200 \) kw if the motor and gear-box efficiency is taken to be 80\% overall). Taking the mean circumference \( 2\pi R \) of the annular cut to be 30 m thus yields a nominal value for \( v = 30/(18 \times 60) = 1/36 \) m/s and hence an average bite \( y = v T = 1.8/36 = 0.05 \) m (i.e. 5.0 cm). Thus an acceptable nominal value for \( k_h \) would appear to be \( 160/0.05 = 3200 \) kw/m.

2.2 Induction Motor Inertia Lag

The dynamic relationship between input power \( P_i \) to the induction motor and output power \( P_m \) may be approximately modelled by a first-order lag thus
\[ P_e = k_1 P_m / (1 + T_m D) \]  

(3)

where \( k_1^{-1} \) is the overall motor and gear-box efficiency so that \( k_1 \) is here set at 0.8 \(-1\) = 1.25 and \( T_m \) is the time constant of the motor's rotor, cutting head and gear-box: the former being dominant because of the enormous reduction ratio. \( D = d/dt \) and \( T_m \) is given by

\[ T_m = - J / (\dot{\Omega} / \Omega) \]  

(4)

where \( \dot{\Omega} / \Omega \) is the slope of the motor's torque-speed characteristic, being measured in radians p.u. time.

Now assuming that, as is normal, peak torque, \( T_m' = 2.5 \) full load torque, here = 2.5 . 200 = 500 synchronous kw and that peak torque occurs at about 10% slip, i.e. when slip, \( \Delta \Omega \equiv 150 \) r.p.m. then

\[ \frac{\dot{\Omega}}{\Delta \Omega} = - \frac{T_m'}{\Delta \Omega} = - \frac{(500. 10^2 \cdot 60)}{2\pi \cdot 1500} \left( \frac{60}{150. 2\pi} \right) \]

= 203 N.m/s

Now the motor's moment of inertia is given as 4.89 kg m\(^2\) and allowing a further 100% for other rotating parts makes \( J = 9.78 \) kg m\(^2\). Thus from equation (4) \( T_m \) works out to be 9.78/2.03 = 0.048s.

2.3 Electrohydraulic Servo System

The swash-plate pump and servo-valve can be regarded as being basically an integrator (since the rate of swash travel, and hence the rate-of-increase of \( v \), will be proportional to valve-opening). The feedback from the swash potentiometer, inbuilt into the control system around this integrator, renders the system model again as a first-order lag. (The valve itself, having a bandwidth if 50 + 100 Hertz, can be regarded as being virtually instant-acting in this application). The system can therefore be modelled thus:

\[ v = k_2 P_e / (1 + T_h D) \]  

(5)
where $P_e$ is the power error signal (here measured in kw) entering the servo amplifier (driving the servo valve) and $k_2$ is the overall gain (sensitivity) of the electronic and electrohydraulic amplifier system. The lag $T_h$ has yet to be determined either from a recorded step response or from valve and servo cylinder capacitance data. Initial inspection at Cadley Hill suggested a rapid response in this part of the system and a value of $T_h = 0.25s$ has been assumed initially. The value of $k_h$ is, of course, readily adjustable.

2.4 Overall System Model and Existing System Performance Prediction

Combining the individual component models of the hydraulics, the cutting head, its drive motor and the controller yields the block diagram representation of Fig. 2. Various alternative controllers are shown including the existing overriding controller. Also shown in Fig. 2 is an electronic smoothing filter for the removal of a.c. ripple from the power transducer output. It is modelled by

$$P' = P_i/(1 + T_f D)$$

(6)

where $P'_i$ is the filtered value of $P_i$ and $T_f$ is the filter time constant. To effectively remove the electrical ripple (50 Hertz) $T_f$ need only have a value of about, $5/2\pi50 = 0.016s$, such a value being virtually negligible in the present context of overall system behaviour. The value of $T_f$ actually used at present has yet to be determined and for the moment the above minimum value is assumed. The diagram provides the basis for a digital simulation of the system, the results of which are described fully in Section 4.

For the moment we consider only Fig. 3 which shows the predicted performance of the existing system with the rate reference $v_r$ set to produce a speed $v$ just less than that needed to bring in the overriding load control ($P_r$ here being set at 200 kw). The desired steady speed
is clearly achieved following switch-on but halfway through the test
a 50% step increase in rock hardness (i.e. in $k_h$) is imposed so actuating
the load control. Clearly sustained oscillations of cycle-time $\approx 4s$ occur.
This frequency is only very slightly slower than that observed, and, in
this connection, it should be noted that $T_m$ was here set at an initially
estimated value of 0.5s not the updated figure of 0.048s calculated in
Section 2.2. Reducing $T_m$, which turns out not to have a dominant effect,
will obviously increase the predicted oscillation frequency so yielding
good correlation between simulation results and those observed on site.

It will also be noted from Fig. 3 that whereas $P_i$ oscillates with
moderate amplitude ($\pm 38\% P_r$) in the harder material, $v$ fluctuates much
more widely, i.e. between zero and $k_h$ the value ($= P_r/k_h T$) at which
load control is activated. Both these amplitudes accord roughly with
initial observation. As proper recordings of $P_i$, swash angle and boom
rotation become available it should be possible to tune up the model of
Fig. 2 to produce predictions of improved accuracy. Present predictions
seem sufficiently accurate however, to permit the existing model to be
used as a basis for the initial design of controller improvements.

3. DESIGN OF LOAD CONTROL COMPENSATOR

3.1 Inverse Nyquist Locus

Assuming the use of a linear continuous controller of the form

$$v_x = k_1 (P_r - P'_i) \tag{7}$$

then the overall open loop transfer function of the uncompensated system
is (from Fig. 1):

$$G(s) = \frac{P'_i}{P_r - P'_i} = \frac{k_1 k_2 k_h (1 - e^{-TS})}{(1 + T_f s)(1 + T_h s)(1 + T_m s)} \tag{8}$$

and, setting Laplace variable $s = j\omega$, where $\omega$ is the angular frequency
of any imposed error signal $P_r - P'_i$, yields an inverse Nyquist
locus* of the form shown in Fig. 4. It is interesting to note from
consideration of the locus (shown as a broken line) of the ideal case
of $T_f = T_h = T_m = 0$ that the critical frequency $\omega_c$ at which unstable
oscillations will occur (if gain $k_1 k_2 k_h$ is set sufficiently high)
must lie in the range

$$\pi/T < \omega_c < 2\pi/T$$

(9)
i.e. the critical oscillation period must lie in the range $T$ to $2T$
(i.e. 1.8 to 3.6 seconds - similar to those observed on the actual
machine).

3.2 Setting the Controller Gain

To achieve stability without significant slugging of the system
it is necessary to reduce the system gain only until the critical 'minus-
one point' on the locus of Fig. 4 is close to the point at which the real
axis is crossed (i.e. at $\omega = \omega_c$). The locus is then to be deflected
away from this critical condition by means of dynamic compensation
networks designed in 3.3.

For the parameters assumed ($T = 1.8s$, $T_m = 0.5s$, $T_h = 0.25s$ and $T_f$ negligible) it is found that $\omega_c = 2.09$ rad s$^{-1}$ and at this frequency

$$|G(j\omega_c)^{-1} k_1 k_2 k_h| = \left| \frac{(1+T_f j\omega_c)(1+T_h j\omega_c)}{m_c h_c} \right| \omega_c$$

$$\left| 1 - e^{-T_c j\omega_c} \right|$$

(10)

$$= \frac{1.55}{1.89} \frac{1.10}{1.25} \frac{2.09}{3200} = 1.89$$

so that, to make $G(j\omega_c) = 1.0$, gain $k_1$ must be set to

$$k_1 = 1.89/k_2 k_h = 1.89/(1.25 \cdot 3200) = 0.47 \cdot 10^{-3} \text{ m/s/kW}$$

*i.e. a polar plot of $(P_1 - P_1')/P_1'$ in amplitude and phase as frequency $\omega$
is varied*
In the simulations a value of \( k_i = 0.3 \times 10^{-3} \, \text{m/s/kW} \) was used to allow a margin of safety.

3.3 Dynamic Compensator Design

To deflect the inverse Nyquist locus away from the critical minus one point to give a resonant closed-loop gain \( * = \) the static closed-loop gain, an added vector shift of approximately unit magnitude and \( 135^\circ \) phase shift is needed at \( \omega = \omega_c \) as Fig. 4 illustrates. To achieve this, it is proposed to employ an additional feedback from the power-transducer via a differentiating network of transfer-function

\[
P_d' = P_i' = k_d T_d \frac{s}{1 + T_d s}
\]  

(11)

where \( P_d \) is a so called 'derivative power' feedback and an additional forward path lag of transfer-function

\[
P_e' = \frac{P_e}{1 + T_L s}
\]  

(12)

where

\[
P_e = P_r - P_i
\]  

(13)

the control law now taking the form:

\[
v_r = k_i (P_e' - P_d')
\]  

(14)

rather than simply that of equation (7) i.e.

\[
v_r = k_i P_e
\]  

(15)

The added vector \( V(s) \) will be given by

\[
V(s) = \frac{k_d T_d s (1 + T_L s)}{(1 + T_d s)}
\]  

(16)

so that \( T_d \) is chosen \( \ll 1.0 \) to allow the differentiator to provide nearly \( 90^\circ \) of phase advance whilst the additional lead \( (\approx 45^\circ) \) is provided by the \((1 + T_L j \omega_c)\) term. We therefore set

\[
T_d \omega_c = 0.2 \, \text{, say}
\]

so that \( T_d = 0.2/2.09 \, \text{(say)} = 0.18 \)

*The closed-loop gain i.e. between \( P_r \) and \( P_i' \) at any frequency \( \omega \) is the reciprocal of the vector drawn from \( P_r \) critical -1 point to the locus at frequency \( \omega \).
Now for the correct slope of added vector, from equation (16)
\[- \tan^{-1} T_d \omega_c + 90^\circ + \tan^{-1} T_L \omega_c = 135^\circ\]
\[\therefore \tan^{-1} T_L \omega_c = 45^\circ + \tan^{-1} 0.209 = 56.8^\circ\]
\[\therefore T_L \omega_c = 1.53\]
so that \[T_L = 1.53/2.09 = 0.73\] s, \[\text{say 0.75s}\]

Hence length of added vector, \[\omega = \omega_c\]
\[= \frac{k_d \cdot 0.209}{1 + j \cdot 1.53} \cdot \frac{1 + j \cdot 2.69}{1 + j \cdot 0.209} = k_d = 1.0\]
\[\therefore k_d = 2.69.\]

These parameter values were therefore employed in initial simulation studies the results of which are given in Section 4.

4. SIMULATION RESULTS

Simulation result for the existing overriding controller are shown in Fig. 3 and were discussed in Section 2.4. Some instability is clearly obvious.

Fig. 5 shows the simulated performance of a proportional controller (i.e. equations 7 or 15) with the controller gain \[k_c\] set to \[0.3 \times 10^{-3}\] i.e. a predicted stable value for a hardness coefficient of 3200 kw/m bite. Clearly the system is stable with this softer material but when the hardness is increased by 50% to 4800 kw/m bite (after 25 seconds) instability just occurs; in accordance with the predictions of Section 3.2. The reference \[P_r\] was set at 200 kw for Fig. 5 but clearly the average value of \[P_i\] obtained falls below this value. This is to be expected since analysis of the system readily reveals that, in steady state:
\[ \frac{P_i}{P_r} = \frac{k_k k_T}{1 + k_k k_T} \]  

(17)

so that, theoretically, the average values of \( P_i \) expected are 137 kw and 153 kw in the softer and harder rock respectively: both figures according with the simulation results of Fig. 5.

Fig. 6 shows the effect of stabilising the uncompensated proportional control system of equation 7 by reduction of controller sensitivity from \( 0.3 \times 10^{-3} \) to \( 0.15 \times 10^{-3} \) m/s/kw. Stability is now achieved under both hardness conditions as expected but with \( P_r \) again set at 200 kw. Thus, although stability can be achieved by gain reduction above, this is at the expense of control accuracy: a 50% change in rock hardness causing a 20% change in load.

The trade-off between stability and static accuracy is much improved by using the dynamic compensator of Section 3 as Fig. 7 shows. The controller gain \( k_1 \) was here returned to \( 0.3 \times 10^{-3} \) m/s/kw and the compensator parameters used were \( k_d = 2.69, T_d = 0.1 \text{ s} \) and \( T_L = 0.75 \text{ s} \) (i.e. identical values to those derived in Section 3). \( P_r \) was again set at 200kw. Stability is clearly now achieved with the higher system sensitivity so that now the 50% hardness increase causes only a 12% increase in \( P_i \) on average. (It is obvious of course that pick wear may be regarded as an equivalent increase in rock hardness and its effects will be similar in practice). The system response remains good and its static accuracy improved still further if \( k_1 \) is raised to \( 0.45 \times 10^{-3} \) m/s/kw.

Fig. 8 is identical to Fig. 6 (computed for the uncompensated proportional controller) but with \( P_r \) raised from 200 kw to 263 kw to make the steady-state value of \( P_i = 137 \text{ kw} \) before the hardness increase i.e. as with the compensated system. As expected, the value of \( P_i \) now increases
to 163 kw after the 50% hardness change, compared to the 153 kw of Fig. 7.

Finally Fig. 9 is included showing the performance of the uncompensated system with the sensitivity raised to 0.5 \(10^{-3}\) m/s/kw yielding unstable behaviour with the higher and lower hardness values. Load oscillations are of unacceptable amplitude and rotational speed \(v\) varies in a hopping manner similar to its behaviour with the existing controller (Fig. 3). The results given would therefore appear to make an excellent case for the compensated proportional controller (Fig. 7).

5. CONCLUSIONS AND PROPOSALS FOR FURTHER WORK

A preliminary mathematical model has been derived for the rock-cutting operation of the 300 h.p. tunnelling machine and initial estimates for its parameter values have been derived. The performance of the model, under the control of a model of the existing load controller, when simulated on the digital computer, closely resembles that observed on the actual machine underground producing low-frequency load oscillations of about 4s cycle-time and ± 38% amplitude. Underground recordings of power consumption and swash-plate pump angle are needed for fine tuning of the model.

Frequency response analysis using the derived machine model shows that responsive control without servo oscillation should be obtainable by:

(a) conversion to continuous control action
(b) addition of derivative control action to the load feedback signal
and (c) insertion of a small lag in the power error path.

The findings are confirmed by simulation which also proves the modified system to be widely tolerant of machine and controller parameter variation.

Further investigation must be conducted to examine the causes and effects of the higher frequency chatter (16 Hertz or so) that is
discernable through machine vibration underground. It is unlikely that this can be compensated significantly by control action but its source would be worth knowing to aid future machine redesign. The chatter could upset the performance of the proposed controller but we feel sufficiently confident to proceed with the necessary circuit modifications to the existing amplifier on the basis of the results so far obtained in parallel with the chatter study.

An analogue machine and servo model is under development into which the modified amplifier will be connected to test its behaviour before installation on the machine for 'before' and 'after' trials. Ideally, an intermediate test of the amplifier on a valve-loaded hydraulic power pack would be desirable also to ensure no unforeseen complications associated with the electrohydraulic servo.

If an improved overriding control is ultimately preferred to the proposed continuous controller, the required conversion should be possible without undue complication.

6. REFERENCES

(1) Edwards J.B.: 'Modelling the dynamics of coal and mineral cutting'


Fig. 2 Block Diagram of Rock-Cutting System and Alternative Controllers
Fig. 3. Simulation results for existing controller.
Locus for $T = T_1 = T_2 = 0$
Note locus $\rightarrow \infty$ as $\omega \rightarrow 2\pi/T$

Compensated System Locus

Added vector $\{1/135^\circ\}$ needed for compensation

Locus of $G^{-1}(j\omega)$ for $k_1 = 0.3 \times 10^{-3} \text{ m/s/kW}$

$\omega_c$ - Critical point

$\omega = \pi/T$

$135^\circ$

Fig. 4 Inverse Nyquist Locus
Fig. 6. Proportional Controller, $k_i = 0.15 \times 10^{-3} \text{m/s/kw}$. No compensation.

$tr = 0.05$
$k_2 = 0.0005$
$TC = 0.06$

$P_i = 200 \text{ kw}$
Fig. 7 Continuous controller with compensator

$p_t = 200\text{kw}$
Fig. 8. Continuous controller. $A_1 = 0.3 \times 10^{-3}$ m/s/kw. No compensation, increased ref.

$F_{\theta}^S$
Fig. 9. Continuous controller, \( k = 0.5 \times 10^{-3} \) m/kg, without compensation.