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Edwards, J.B. and Sadreddini, S.M. (1984) Vibration due to the Resilience of a Tunnelling Machine Boom Structure and its Effect on Load-Control. Research Report. Acse Report 265 . Dept of Automatic Control and System Engineering. University of Sheffield

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Vibration due to the resilience of a tunnelling machine boom structure and its effect on load-control

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Research Report No. 265

September 1984

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1. Introduction

Previous analyses (1), (2) and (3) carried out for the N.C.B. on Research Contract No. 226072 in connection with the boom-type tunnelling machinesoperating at Cadley Hill Colliery and Middleton Mine were all based on the assumption of a completely stiff boom structure between the horizontal turret and the cutting-head. This assumption was necessary to make rapid progress in the redesign, modification and underground testing of the load control system in an attempt to alleviate the pressing problem of regular and serious damage to the expensive 1500 : 6 r.p.m. reduction gear-box transmitting the power for head rotation (300 h.p. approximately). Using a dynamic model involving the delay, T, between consecutive lines of cutting picks and simple time-constants T_h and T_m for the electrohydraulic boom-drive and for the 300 h.p. induction motor respectively, simulation and servo analysis were carried out to produce a redesigned load controller involving additional electronic lags, derivative and integral action.

2. Underground Trial Results

Whereas exhaustive simulation trials were conducted involving the original and redesigned controllers hooked up to real-time versions of the tunnelling machine simulation, the opportunities for underground testing (a) to validate the simulation and (b) to test the new controller were somewhat restricted. The results that were obtained underground were nevertheless most encouraging and Figs. 1 and 2 compare respectively the performances of the original (Bretby) system with the modified (Sheffield) system. The machine power is clearly controlled far more tightly in the case of the Sheffield system (Fig. 2), eliminating the low-frequency surging evident in Fig. 1. The tighter control is achieved principally through the use of a higher proportional gain, made possible by the added derivative action and the increased controller activity is evidenced by the trace of servo-valve current.

The photographs of the cutting patterns produced by the head using the Bretby and Sheffield systems (Figs. 3 and 4 respectively) also confirm the greatly increased stability of operation produced by the modifications.

From this evidence, the neglect of structural vibration in the dynamic model would appear to have been justified and one source of overload on the main gear box thus compensated. There remains, however, considerable high-frequency chatter on the power traces prior to filtration as seen in Figs. 5 (Bretby) and 6 (Sheffield) and the modified controller has done little to eliminate this second potential source of overload. Indeed this was acknowledged at the outset. The high frequency load oscillations arise from vibrations of various types and in this report, vibration due to resilience of the boom structure is investigated. For this purpose therefore the resilience was actually measured underground on the EIMCO machine at Middleton Mine. The measurement was carried out by first anchoring the machine solidly against the tunnel walls (as during normal cutting operations), sumping in, ranging the boom out to full tunnel radius and, with the drum drive switched off, applying increasing pressure to the boom rotation motors from the swash-plate pump. During this latter phase, a graph was taken of turret rotation angle versus pressure as the boom structure was stressed up to full working pressure (above 2000 p.s.i.) and, from the slope of the graph, the effective stiffness, λ_b, of the structure could be determined. Details are given in Appendix 1.

As will be shown, the inclusion of boom resilience in the mathematical model does predict high-frequency chatter similar to that exhibited in Figs. 5 and 6, by both theoretical analysis and computer simulation. Reassuringly, the effects of these simulated vibrations do not appear to degrade the improvements in overall (i.e. average) load control effected by the modifications implemented. The continued presence of this high frequency chatter could however remain a source of trouble and its extent requires further study.

3. Inclusion of boom structure stiffness in the mathematical model

Once resilience in the boom structure is admitted a distinction must be made between v_2, the linear speed of the boom (around the tunnel periphery) and v_1 (= $\Omega_h R$) where Ω_h is the angular speed of the turret and R is the radius of boom rotation. These two linear speeds will only attain equality in a steady-state situation which will not be achieved if vibrations persist. v_1 will then only equal the average value of v_2. The force F_{ab} applied to the head by the boom will thus be governed by

$$dF_{ab}/dt = \lambda_b (v_1 - v_2) \qquad \ldots(1)$$

whilst the cutting force, F_{cb}, resisting boom rotation may be taken as being proportional to the pick bite y so that

$$F_{cb} = k_c y \qquad \ldots(2)$$

where $$dy(t)/dt = v_2(t) - v_2(t-T) \qquad \ldots(3)$$

provided $$y(t) \geq 0 \qquad \ldots(4)$$

T being the delay between consecutive lines of picks and k_c an effective rock-hardness constant. The difference between the applied and cutting forces will of course go to accelerate the boom and overcome any damping forces so that

$$M_b dv_2/dt + fv_2 = F_{ab} - F_{cb} \qquad \ldots(5)$$

where M_b is the effective mass of head and boom and f a damping factor. As in previous analyses, the mechanical power, P_m, delivered to the cutting head is also assumed proportional to bite y so that

$$P_m = k_h y \qquad \ldots(6)$$

where k_h like k_c also depends on rock-hardness, pick-sharpness, and depth of sump but also on drum drive speed. As before we assume simple lags between P_m and electrical power consumption P_i and between v_1 and speed v_{1d} demanded by the controller so that

$$P_i + T_m dP_i/dt = k_m P_m \qquad \ldots(7)$$

and $$v_1 + T_h dv_1/dt = v_{1d} \qquad \ldots(8)$$

where T_m and T_h are time constants and k_m is related to the average efficiency of the induction motor drive.

Fig. 7 shows the equations assembled into a block diagram form and, provided condition (4) is satisfied, the transfer-function between v_1 and v_2 is readily deduced from this diagram to be

$$\frac{v_2(s)}{v_1(s)} = \frac{1}{1 + s(f/\lambda_b) + s^2(M_b/\lambda_b) + (k_c/\lambda_b)\{1-\exp(-sT)\}} \quad \ldots(9)$$

where s is the Laplace variable ($\equiv d/dt$).

4. Mechanical stability prediction

From equation (9), the inverse transfer function between turret and boom speeds v_1 and v_2 may be considered to be a summation of two terms thus

$$v_1(s)/v_2(s) = G_m^{-1}(s) + G_c(s) \quad \ldots(10)$$

where $$G_m(s) = 1/\{1 + s(f/\lambda_b) + s^2(M_b/\lambda_b)\} \quad \ldots(11)$$

and may be regarded as the structural transfer function of the boom whereas

$$G_c(s) = (k_c/\lambda_b)\{1 - \exp(-sT)\} \quad \ldots(12)$$

and may be regarded as a transfer function of the cutting action. Now $G_m(s)$ is a familiar second-order lag having an undamped natural frequency ω_{nb} given by

$$\omega_{nb} = \sqrt{\lambda_b/M_b} \quad \ldots(13)$$

and using the value for λ_b of 2636.10^4 N/m measured at Middleton Mine together with M_b = 2500 kg (1.5 tonne for the cutting head plus a further 1.0 tonne allowed for the boom) yields

$$\omega_{nb} = 105 \text{ radians/s} \qquad (\equiv 16.3 \text{ Hertz})$$

The inverse Nyquist locus of $G_m^{-1}(j\omega)$ starts at 1+j0 and crosses the positive imaginary axis at $\omega = \omega_{nb}$ at $0+2j\zeta$ where damping ratio ζ is given by

$$\zeta = f/\sqrt{\lambda_b M_b} \qquad \ldots(14)$$

and given a reasonable degree of inherent damping (corresponding to, say, $\zeta = 0.5$), the radial clearance of the origin by the locus $G_m^{-1}(j\omega)$ is never less than unity suggesting no serious problem from the boom dynamics alone. The problem arises when the circles due to $G_c(j\omega)$ are superimposed. Using a value for k_c consistent with previous assumptions for the force/bite relationship and assuming equal radial and circumferential forces, i.e. $k_c = 1213\ 10^4$ N/m we obtain circle diameters of

$$2k_c/\lambda_b = 2.1213/2636 = 0.92$$

and although the circles lie outside the locus of $G_m^{-1}(j\omega)$ at low frequency, they move inside as ω passes through ω_{nb}. With such a large diameter (compared to unity) the net locus passes extremely close to the origin. With only a small increase in k_c the origin is embraced by the locus predicting mechanical instability.

Fig. 8 shows the loci for k_c = the nominal value of 1213.10^4 N/m and 1820.10^4 N/m respectively and Fig. 9 shows the associated computed step responses of the mechanical subsystem. Clearly a build-up of oscillations is occurring in the slightly harder rock, as predicted.

5. The effect of mechanical vibration on the load-control system

The foregoing analysis has shown that mechanical instability is readily induced by cutting rock only slightly harder than the nominal value estimated from average power and speed observations or by using picks only slightly less sharp than ideal. This is because the boom structure, although designed for high rigidity, has an effective stiffness, λ_b, only slightly larger than the nominal cutting stiffness, k_cx2. The recordings of Figs. 5 and 6 taken at Middleton confirm the presence of this mechanical chatter.

Rather than relying just on the short duration underground trials it was thought important to check that the controller improvements predicted using the "stiff-boom" model carried over to the flexible-boom situation by additional simulation studies.

The linear model of Fig. 7 breaks down in the mechanically unstable case because eventually oscillations become so large that the head bounces out of the cut so contravening the positive bite condition (4). If indeed the R.H.S. of equation (3) becomes negative then due account must be taken of the fact that the pick is now in fresh air and is no-longer producing the next buttock wall. The wall and the pick tip are no longer in identical positions and the model must be modified thus:

Let y_1 be the present leading pick position so that

$$dy_1/dt = v_2 \qquad \text{...(15)}$$

and let z_1 and z_2 be the present and previous wall positions respectively.

Now bite y is given by

$$y = z_1 - z_2 \qquad \text{...(16)}$$

and $$z_1 = y_1 \quad \text{if} \quad y_1 \geq z_2 \qquad \text{...(17)}$$

otherwise $$z_1 = z_2 \quad \text{if} \quad y_1 < z_2 \qquad \text{...(18)}$$

Finally $$z_2(t) = z_1(t-T) \qquad \text{...(19)}$$

where T is the delay between consecutive pick lines.

The model is represented in block diagram form in Fig. 10 and is clearly identical to the form given in Fig. 7 (apart from the interchange of the integrator and delay network) when the bite y is positive.

Augmenting the simulation in this way causes no change in the behaviour of stable systems, nor in that of unstable systems until the zero bite condition is reached. The high frequency oscillations then settle down to constant amplitude.

Figs. 11 and 12 compare the predicted behaviour of the Bretby and Sheffield load control systems in the presence of these vibrations and it is clear from these that the improvements to the average load pattern are sustained in adopting the Sheffield system and do not upset the operation of the compensating networks. (C.f. Figs. 5 and 6 in Research Report No. 220). Fig.13 shows the predicted machine behaviour without any load control whatsoever (i.e. at fixed turret speed) under the same conditions, confirming that the high-frequency oscillations are mechanical in origin and not control-generated.

6. Conclusions

By treating the cutting head and boom as a spring-mass system whose stiffness was measured on the EIMCO machine at Middleton Mine, a simulation has been produced that generates a vibrating load pattern of similar frequency content to those recorded on site.

Both underground trials and subsequent simulation (with this enhanced model) indicate that the presence of these high frequency vibrations in the cutting load does not adversely affect the improvements predicted for the load control system redesigned at Sheffield. Neither the original nor the improved load controllers can eliminate this mechanical vibration however: the improvements achieved affect only the low-frequency behaviour of the system and, in particular, the elimination of servo-hunting that was evident previously.

The high-frequency chatter remains a cause for concern because it alone could do considerable damage to the cutting head drive transmission, bearing in mind that the load oscillations (both recorded and simulated) are much smoothed by the inertia of the induction motor. Mechanical torque traces would show high frequency oscillations of amplitude some five times greater than those of Figs. 5, 6, 11 and 12.

The precise extent of the transmission torque vibrations will be intimately affected by both the precise cutting action and the induction motor dynamics and it is proposed that these factors should be investigated. The need for this detailed study is perhaps even more urgent in the case of the proposed double arm machine where adequate composite arm stiffness may prove difficult to achieve. The study should proceed in parallel with that of the double arm mechanism itself.

7. References

(1) Edwards, J.B., "Load control of a 300 h.p. tunnelling machine', University of Sheffield, Department of Control Engineering Research Report No.220, April 1983, 22pp.

(2) Edwards, J.B. and Sadreddini, S.M., "Controller modification and testing for underground trials", ibid, Research Report No.230, June 1983, 23pp.

(3) Edwards, J.B. and Sadreddini, S.M., "The addition of integral control action to prevent manually created overloads", ibid, Research Report No.237, Aug. 1983, 13pp.

Appendix 1

Calculation of Boom Structure Stiffness from Pressure and Turret Deflection Measurements

Given data:

Capacity of hydraulic motors (total) = 4x125 = 500 in^3

($4x188 = 752\ in^3$ for Cadley Hill machine)

Maximum pump delivery = 12 g p m

$\equiv 12x12^3/(6.24x60) = 55.4\ in^3/s$

$\therefore$ Maximum speed of motor rotation = 55.4/500 = 0.111 rev/s

Turret/motor gear ratio = 1/50.6 (1/51.07 at Cadley Hill)

$\therefore$ @ boom radius R = 2.75m (measured to drum centre),

maximum translational speed of head $v_{1max} = \frac{2.75 \times 2\pi \times 0.111}{50.6}$

$= 3.79 \times 10^{-2}$ m/s

{Check with observed maximum speed:

Theoretical time per turret revolution = 50.6/0.111 = 455.9 s

Observed time for 30^o rotation = 43s

$\equiv 43 \times 360/30 = 516.0$ s/rev

$\therefore$ reasonable agreement }

Force/pressure relationship

If τ = torque, Ω = angular speed, P = pressure, q = flow then $\tau\Omega = Pq$

Now $\Omega/q = (0.111\ 2\pi/50.6)x(55.4x2.54^3x10^{-6})^{-1}$

$= 15.2\ rad/m^3$

$\therefore \tau/P = 15.2^{-1}$ Nm per (N/m^3) i.e. m^3

$\therefore \tau = 15.2^{-1}$ P Nm (if P in N/m^2)

P measured in p.s.i. however

1 kg = 2.2 lb i.e. lb = 0.45 kg

$1\ m^3 = 10^4/(2.54)^2 = 1550\ in^2$

$\therefore$ 1 p.s.i. $\equiv 0.45x1550 = 697.5\ kg/m^2$

$= 697.5x9.81 = 6842\ N/m^2$

$\therefore \ \tau = 15.2^{-1}\ 6842\ P$ Nm (if P in psi)

$\therefore \ \tau(\text{Nm}) = 450\ P$ (p.s.i.)

$\therefore \ F_{ab}$ (linear applied force) $= \tau/R$ ($R = 2.75$ m)

$\therefore \ F_{ab}(\text{N}) = (450/2.75)P$ (p.s.i.)

i.e. $F_{ab}(\text{N}) = 163.7\ P$ (p.s.i.)

Boom-structure stiffness

λ_b = force p.u. deflection of boom at cutting head)

Measurements: Applied pressure of 2400 p.s.i. caused 5.5 mm of turret deflection, where 177 mm $\equiv 10^{\circ}$ deflection

$\therefore$ Angular deflection $= 5.5.10/177 = 0.31^{\circ}$

$\therefore$ Linear deflection between head and boom pivot

$= (0.31 \times \pi \times 2.75)/180 = 1.49.10^{-2}$ m

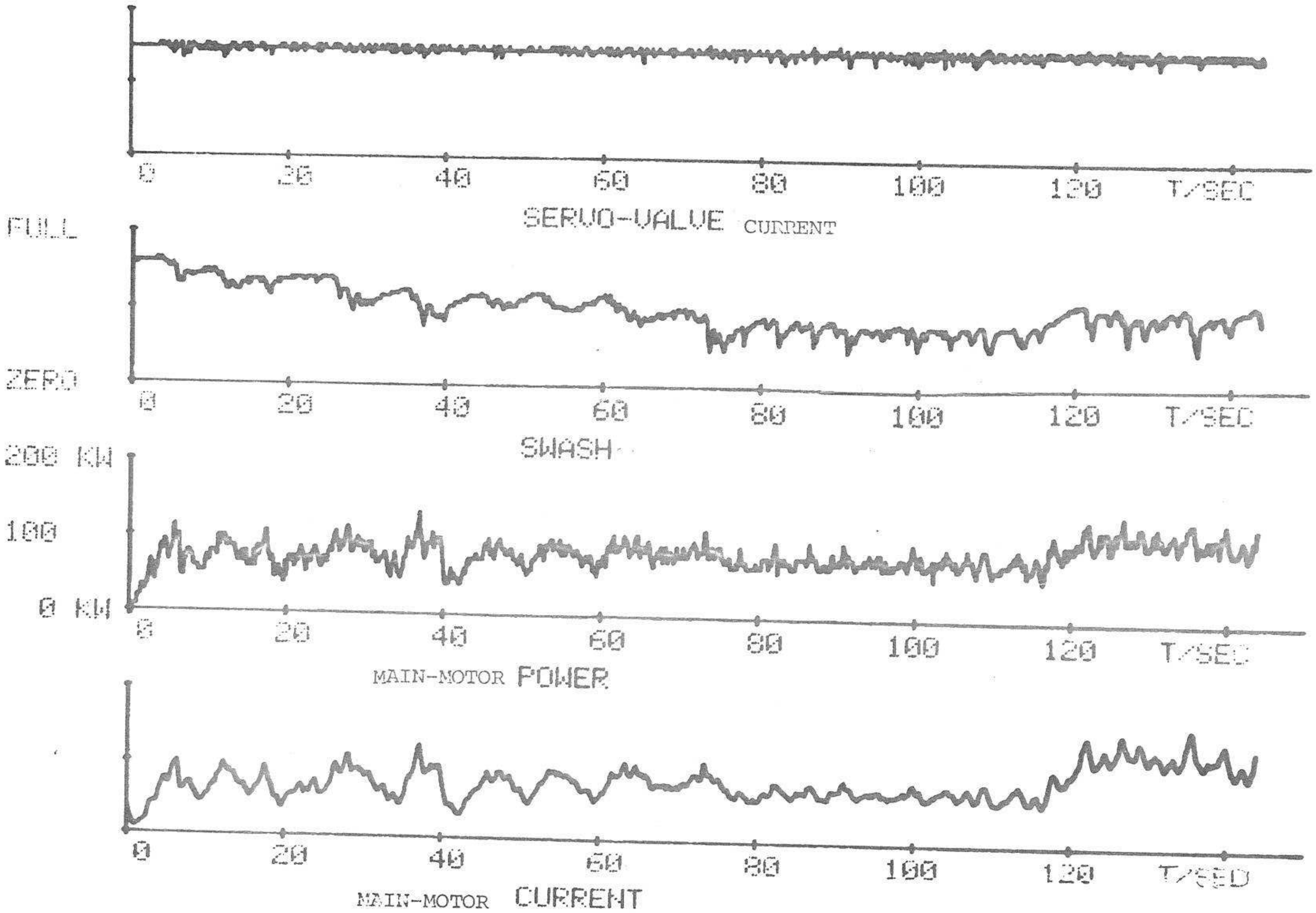
$\therefore \ \lambda_b = F_{ab}/\text{deflection} = 2400 \times 163.7/(1.49 \times 10^{-2})$

i.e. $\lambda_b = 2636.10^{4}$ N/m

i.e. Figure used in all simulations in the present report.

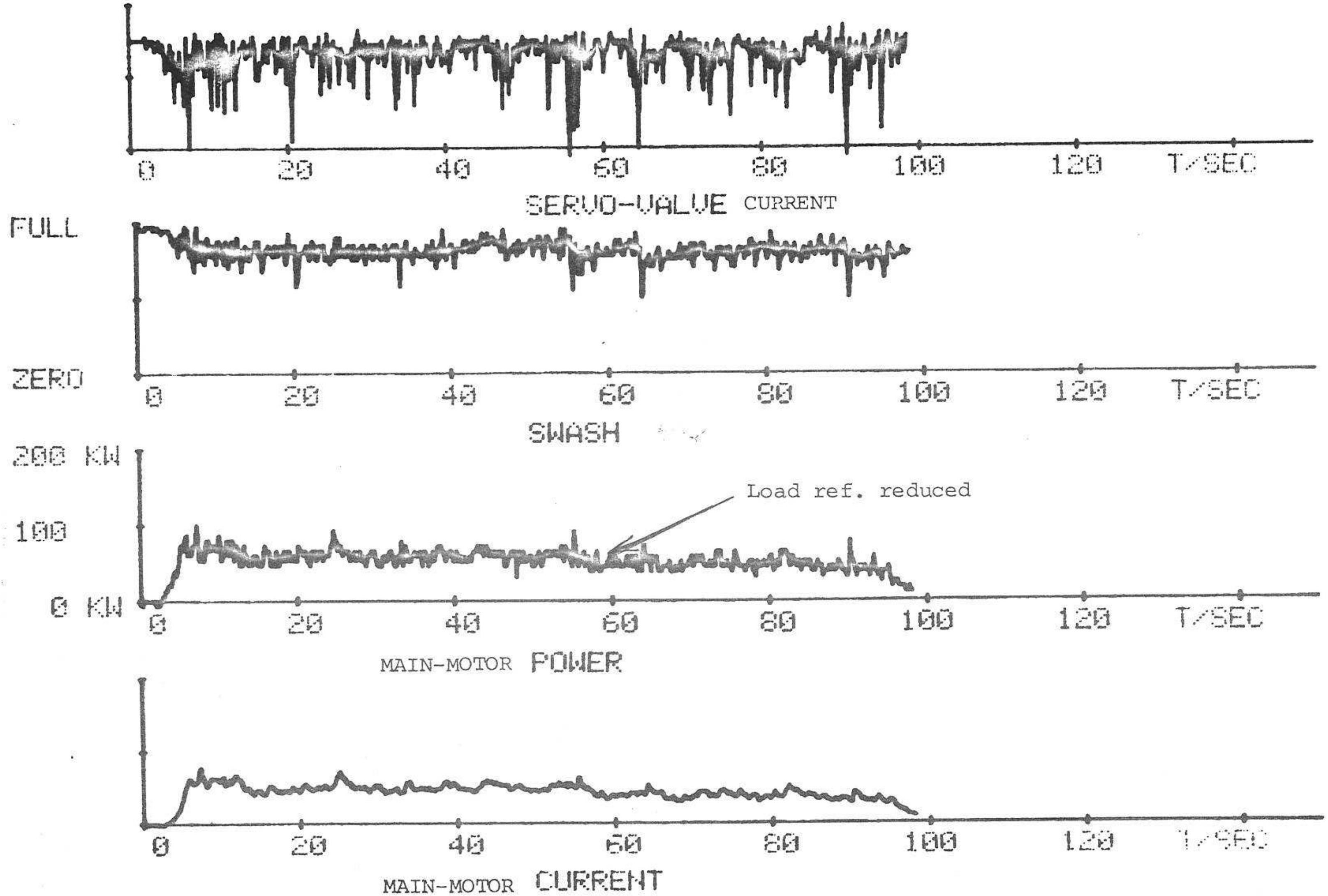
N.B. Compression of hydraulic fluid and hose expansion could reduce the effective value of λ_b still further. This required further investigation.

Fig. 1. Field Trial Results: Middleton Mine



PERFORMANCE OF UNMODIFIED SYSTEM (BRETBY)

Fig. 2. Field Trial Results: Middleton Mine



PERFORMANCE OF MODIFIED SYSTEM (SHEFFIELD)

<u>Fig. 3. Cutting pattern produced with original (Bretby) controller</u>

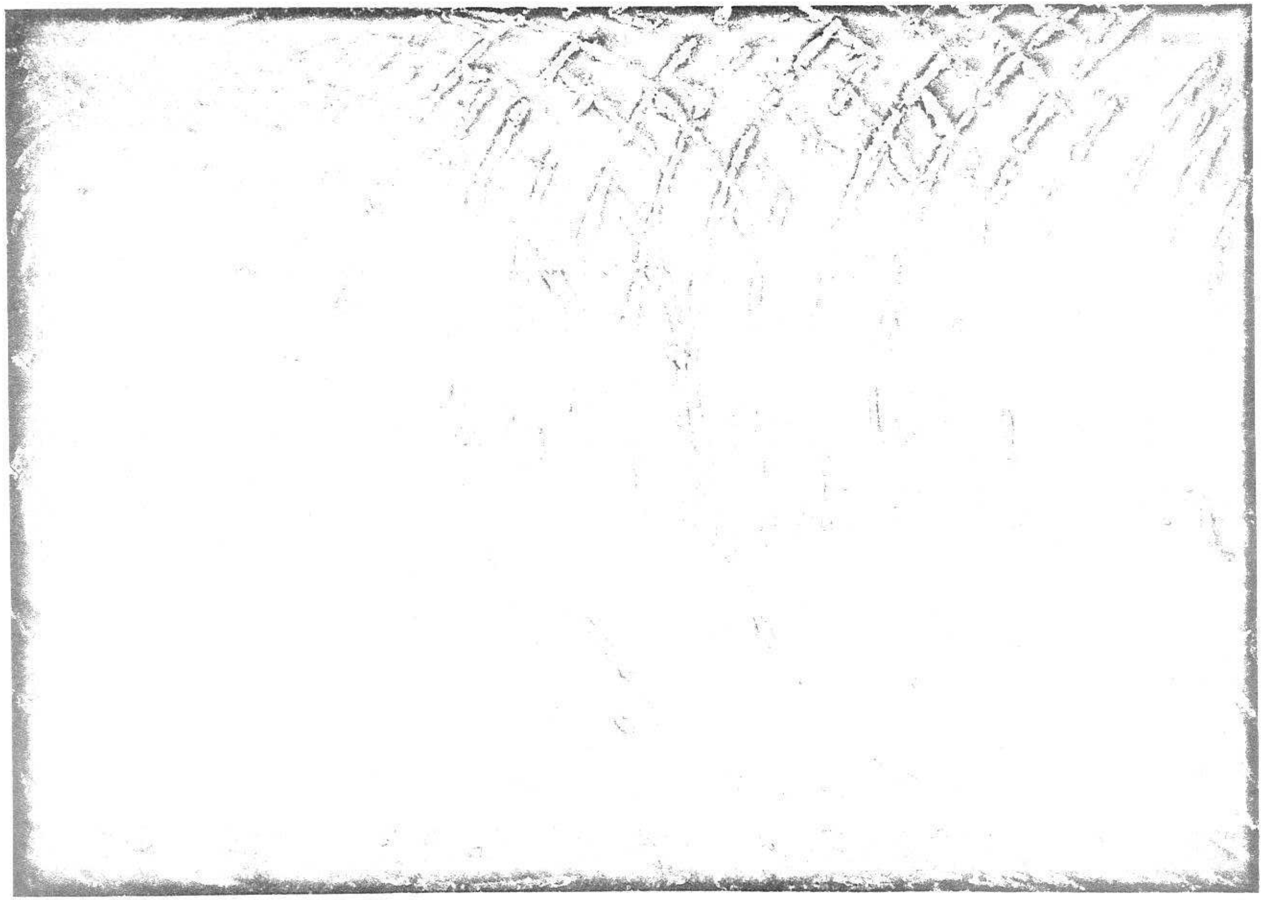


Fig. 4. Cutting pattern produced with modified (Sheffield) controller

Fig. 5. Unfiltered power traces with original (Bretby) controller

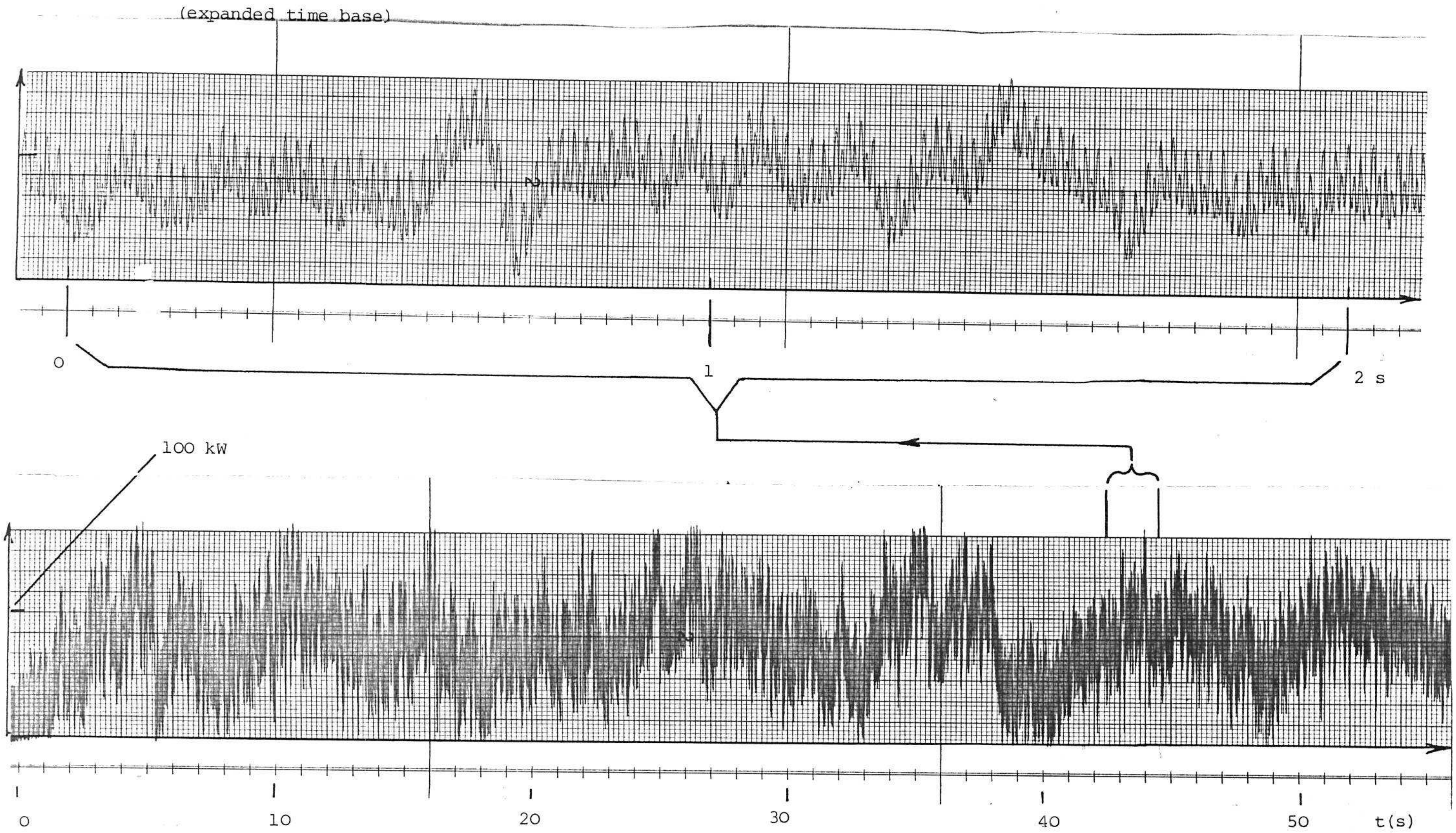
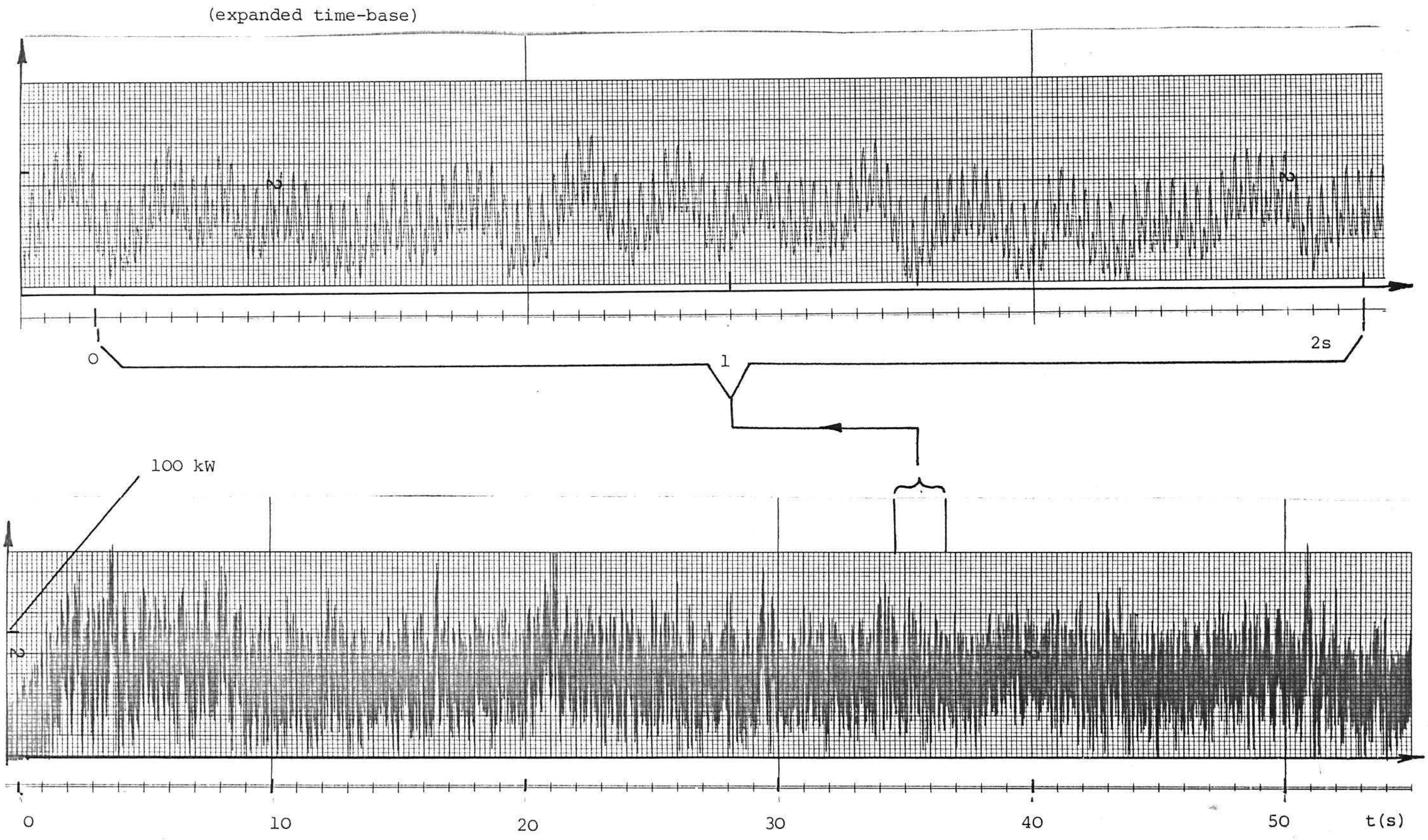


Fig. 6. Unfiltered power traces with modified (Sheffield) controller



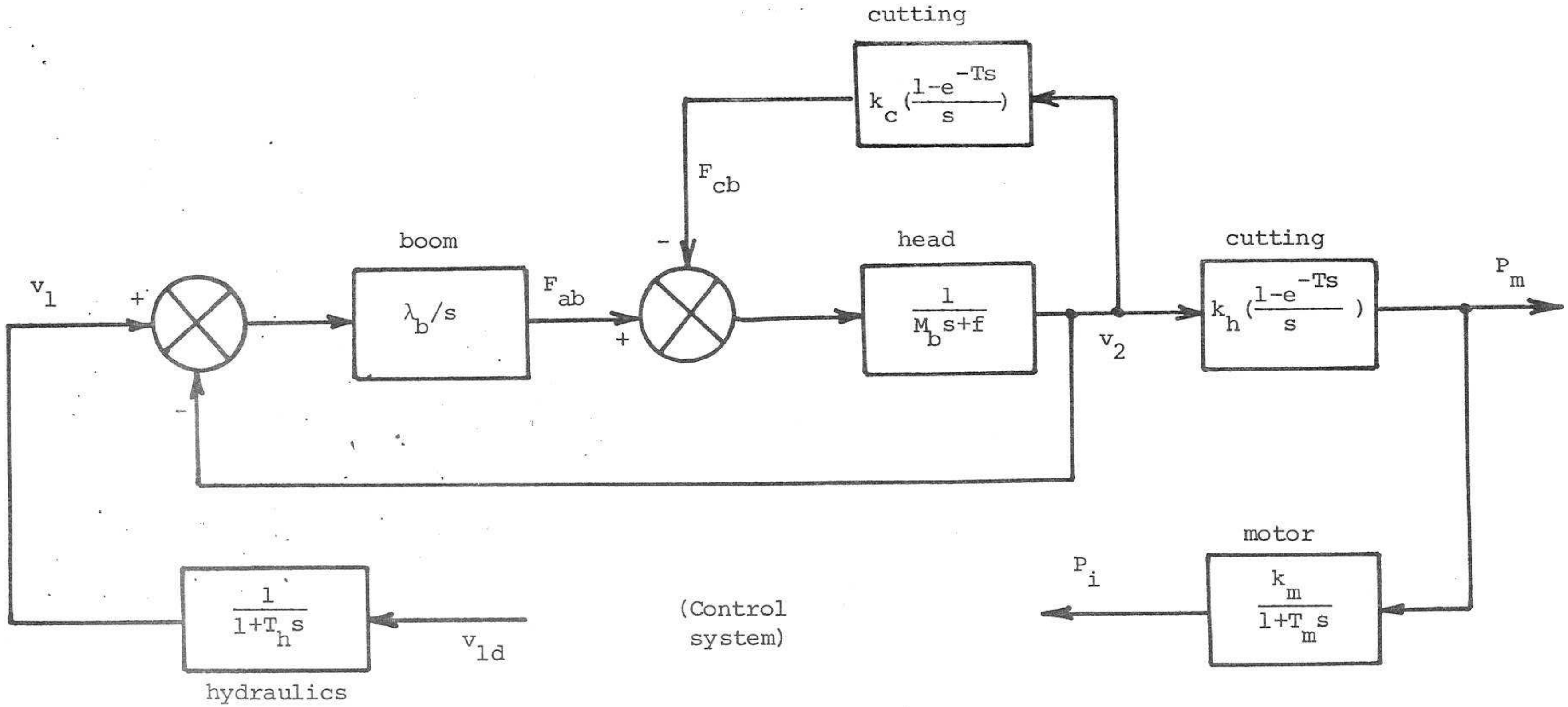
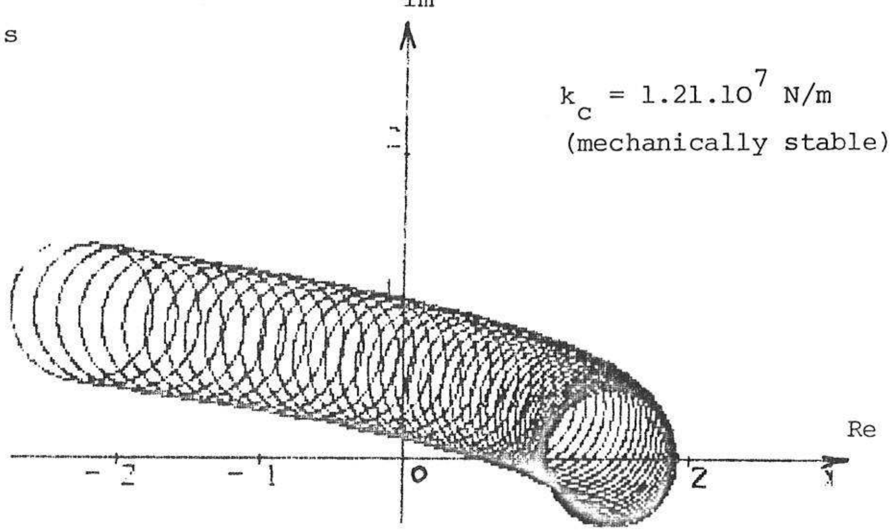


Fig. 7. Block diagram showing addition of Boom Dynamics

Fig. 8. Inverse Nyquist loci (v_1/v_2) with boom dynamics

$\lambda_b = 2.64.10^7$ N/m
$M_b = 2500$ kg
$f = 1.28\ 10^5$ N/m/s
$T = 1.42$ s



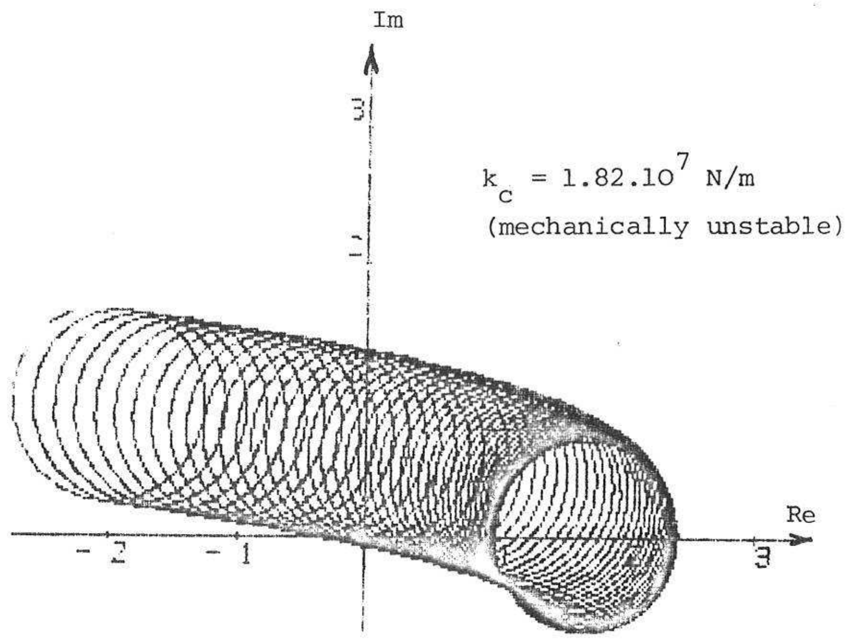
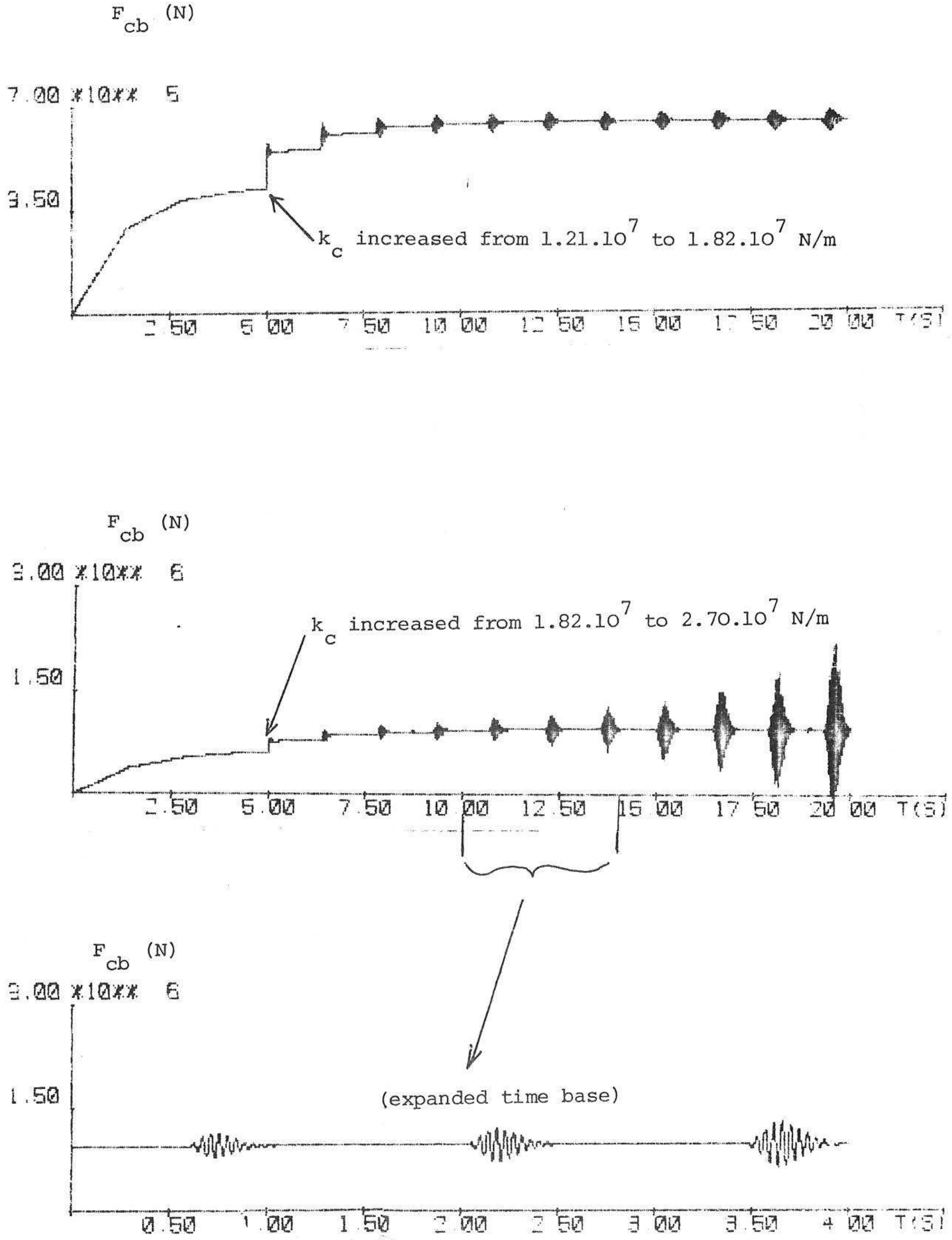


Fig. 9. Transient responses of cutting force at constant turret speed



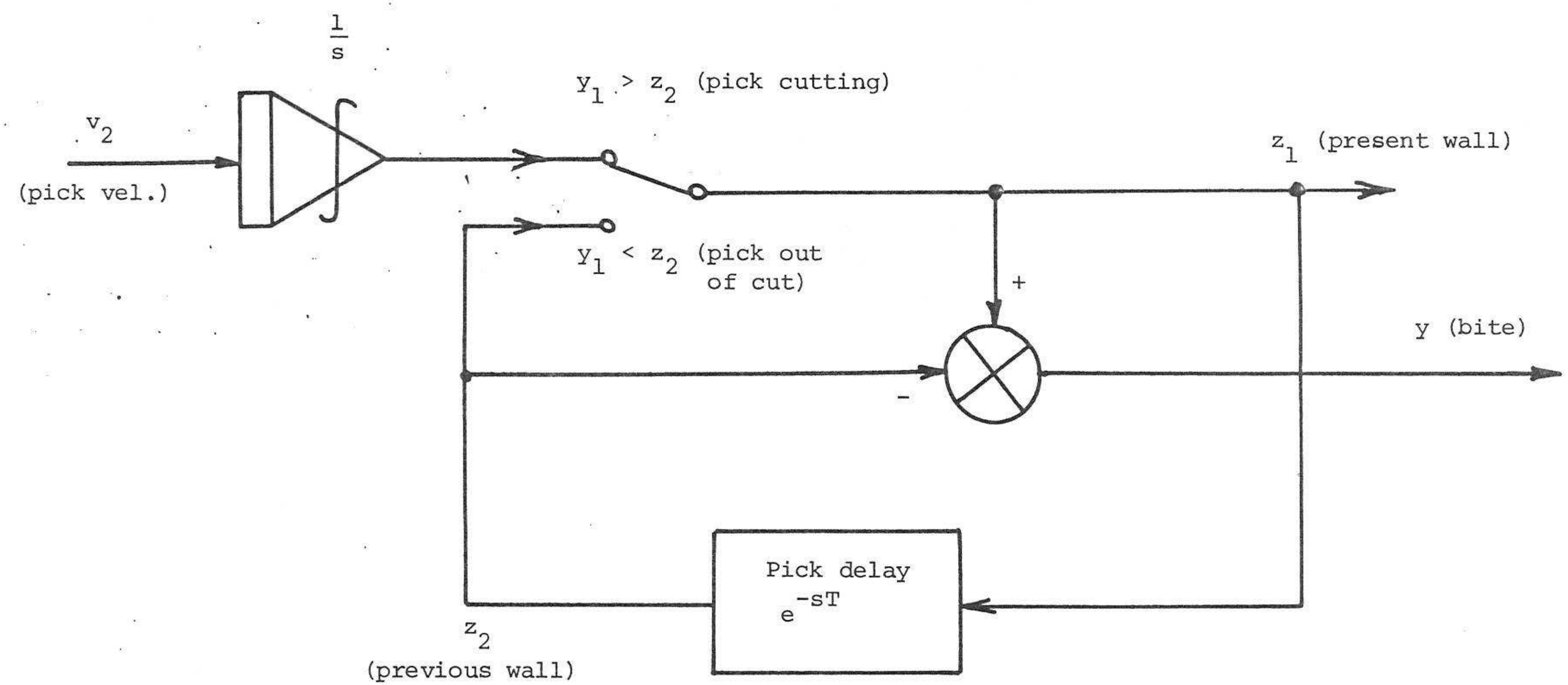


Fig. 10. Implementing the "zero-bite" constraint

Fig. 11.A. Simulated performance with boom dynamics included (original-Bretby system) showing low frequency surging

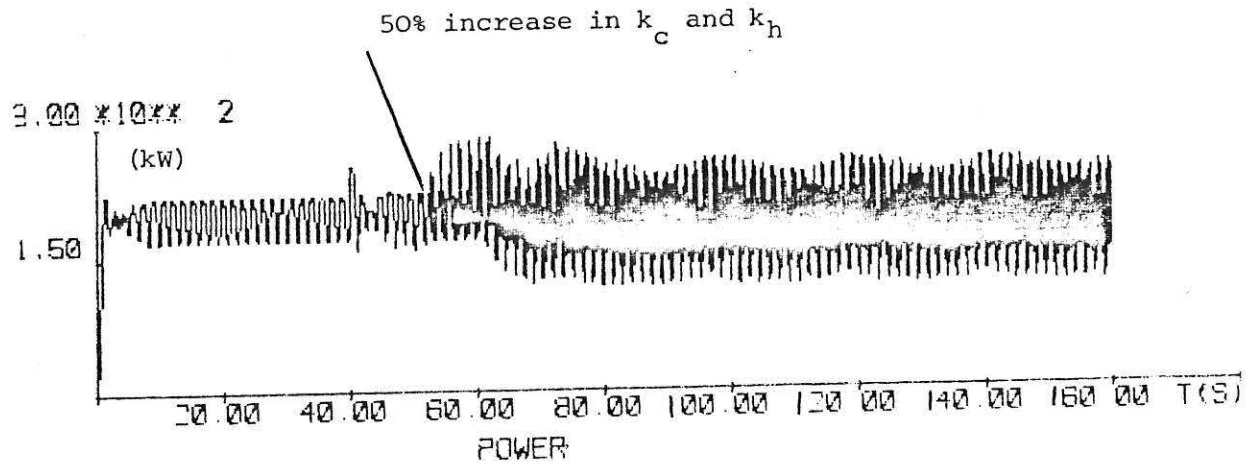
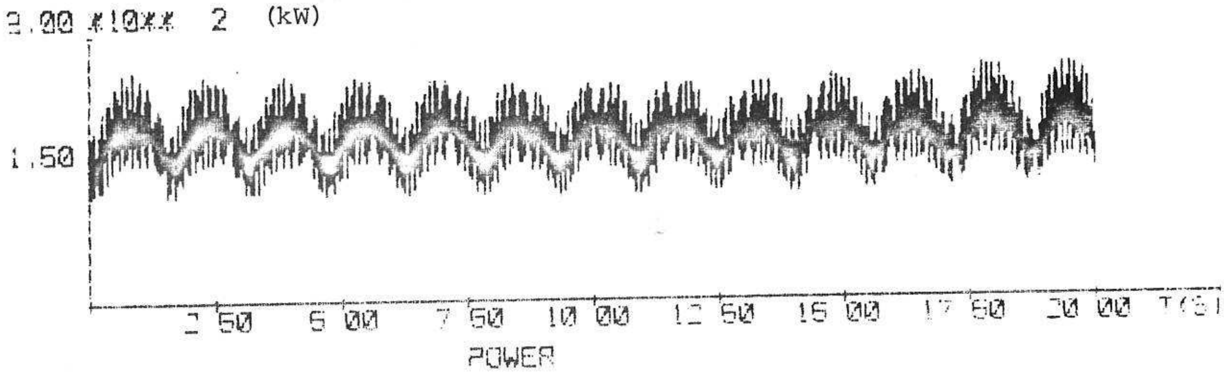
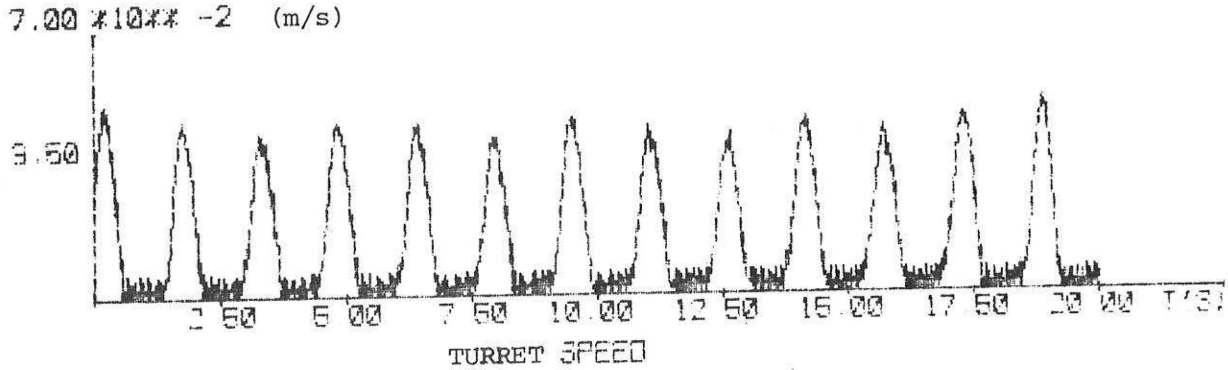
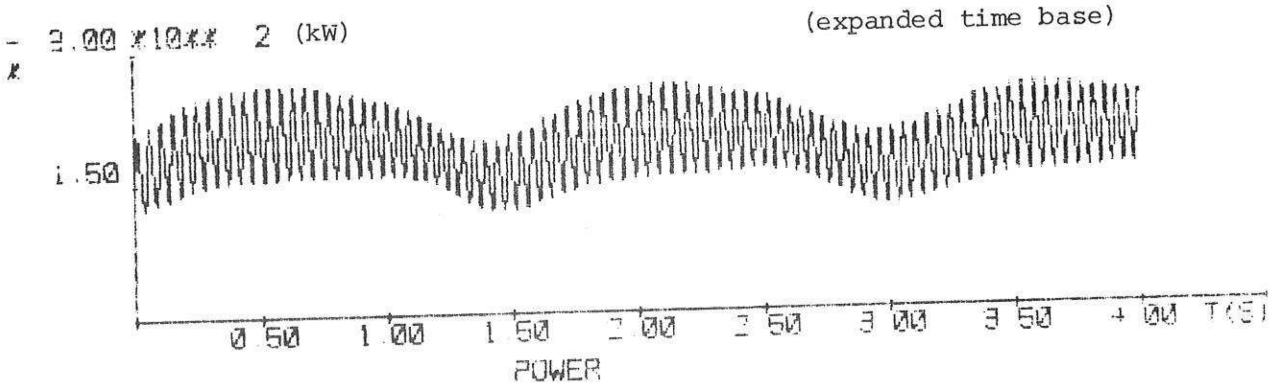


Fig. 11.B. Simulated performance with boom dynamics included (original Bretby system)





(expanded time base)



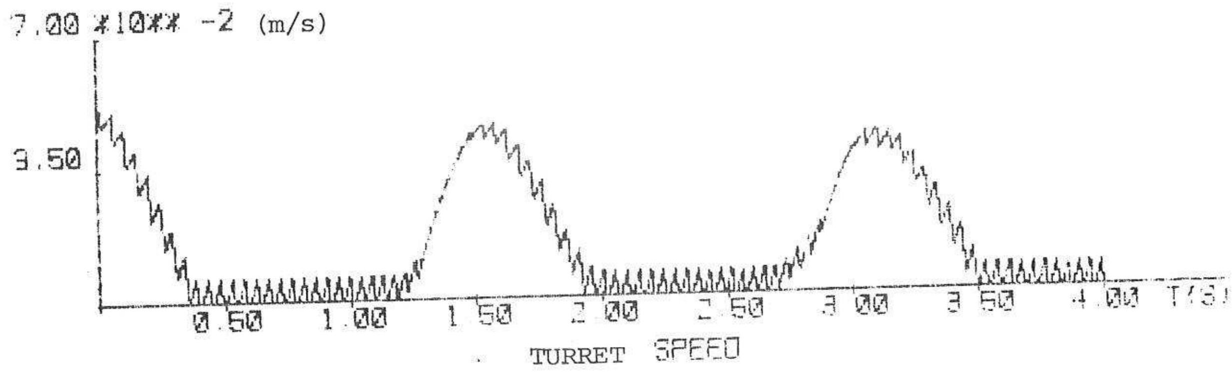


Fig. 12. Simulated performance with boom dynamics included (modified Sheffield system)

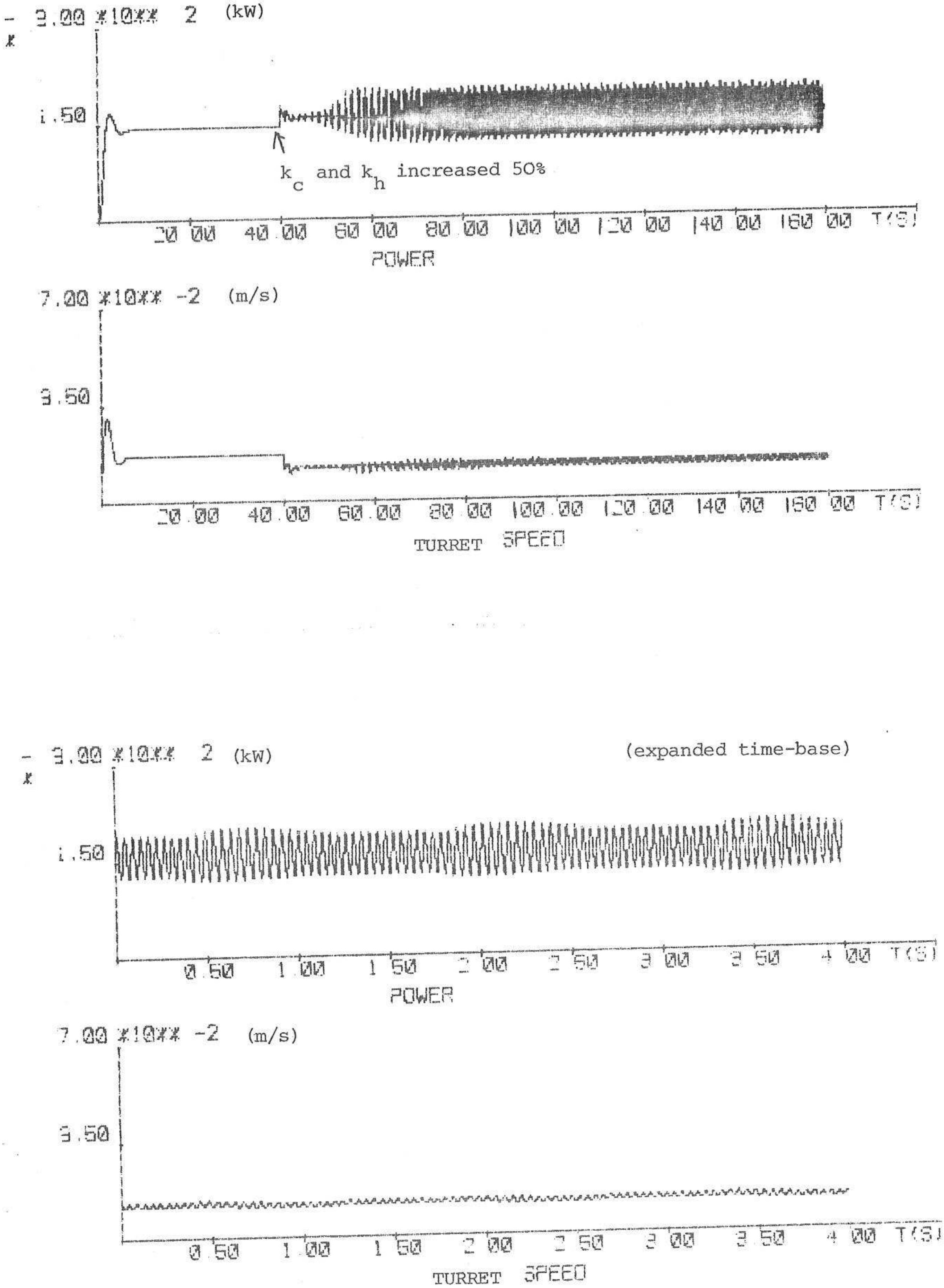


Fig. 13. Simulated performance with boom dynamics included
(Constant turret speed = 0.025 m/s)