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CONTROL OF LOAD AND PROFILE IN THE MACHINE-CUTTING

OF D-SHAPED TUNNELS : AN INITIAL INVESTIGATION

by

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1.   Introduction

        Previous reports in this series (Research Reports 220, 230 and 237)
have been concerned only with the load-control of tunnelling machines
producing tunnels of near-circular cross sectional profile. The investigations
there reported were aimed primarily at improving the load control on such
a machine currently operating at the N.C.B's Cadley Hill Colliery.

1.1 Middleton Trial

        Pending the redesign and recertification of the load controller for
that machine on the basis of the analysis and simulation results in
R.R. 220,230 and 237, it was decided to transfer attention temporarily to
the machine recently installed at Middleton limestone mine for cutting
trials in order that a prototype modified load controller might be tested
on an operating machine (rather than a simulator) unrestrained by Intrinsic
Safety restrictions. A successful one-day trial with the controller driving
the machine's boom-rotation servo was carried out on 14 October. The outer
(load-control) loop was,however, closed via the analogue simulator of the
machine's cutting dynamics (R.R. 230) pending future cutting trials. The
insertion of the real electro-hydraulic servo produced results very similar
to those already obtained with the controller coupled to the complete
machine simulation (cutting action and hydraulics) reported in R.R. 230 and
237.

        Minor practical problems involved recalibration of the simulator to
take the somewhat smaller rate-feedback signal than had earlier been assumed
(2-Volt swing instead of 5-Volts for full swash movement) and consequent
compensation of the rate-gain setting and servo-gain setting. An isolated
laboratory power supply to the controller also had to be used for the trial
due to incompatibility of the zero-volt lines of the controller and VIDAC
simulator when the former was operated from its designed power supply.

This incompatibility is apparently due to some accidental earth connection existing in the tunnelling machine's circuitry but the problem will disappear when actual cutting trials are undertaken and the VIDAC no longer needed.

Typical results obtained are shown in Figs. 1 and 2 which show the effect of simulated ± 50% hardness changes. It will be noted that the steady state cutting load is little affected by the hardness increase and the swash plate angle is reduced in consequence as expected. On reducing to 50% nominal rock-hardness, the load does fall despite the increase in swash angle because, in this softer material, the system has automatically changed over from load to rate control: all as expected. Figs. 3 and 4 show the effect of increasing and decreasing the driver's manual rate-reference in steps of 5% at nominal rock hardness. Initially, the swash angle follows the manual demand nearly proportionally until the preset load-limit is reached, whereafter only transient effects are produced and the speed and load are nearly 'hard-limited': again showing the effectiveness of the integral action introduced in R.R. 237.

1.2   Problems with D-shaped tunnel profiles

An opportunity for a cutting trial at Middleton Mine is now awaited and, provided our simulation of the rock cutting dynamics has been reasonably accurate, we have some confidence in the future success of the modified controller for cutting circular profiles! Now whereas the M.R.D.E. machine at Cadley Hill (to which our design efforts have so far been directed) does cut a circular tunnel, the Eimco machine at Middleton being a caterpillar-track-mounted machine (illustrated diagramatically in Fig. 5) is intended to produce a flat floor and therefore to cut a D-shaped cross sectional profile. This, of course, implies that both boom rotation angle, θ, and boom elevation angle, α, must be varied simultaneously during parts of the cutting cycle and cutting load will therefore depend on both rotational speed Ω (radians/sec) and elevation speed ω (radians/sec). Identical twin

controllers have been provided, one for manipulation of the swash-plates of each of the two separate pumps provided for powering these operations. The problem therefore arises as to how to incorporate the elevation servo into the load-control system and it is to this problem that the remainder of this report is addressed. The investigation reported is necessarily of a preliminary nature and confirmatory simulations and trials will be necessary for validating the ideas propounded and the tentative solutions proposed.

2. The Interaction of Load and Profile Control

The existing profile controller (profiler) operates on the boom elevation actuators not, at present, via the elevation swash angle itself but via a flow-metering valve interposed between the swash-plate pump and the elevating cylinders. The present controller comprises a cam, driven by the rotation shaft, and a follower which drives a potentiometer providing an electrical demanded elevation reference signal from which a signal from an actual elevation potentiometer is subtracted. The amplified error is used at present to actuate the flow metering valve through which the elevation cylinders are powered.

The scheme is not the only one conceivable. An all electric analogue or digital system could replace the cam reference arrangement giving scope for easy alteration of tunnel profile, and a scheme operating on both elevation and rotation swashes may have advantages.

Whilst cutting other than circular trajectories (concentric with the rotation drive centre) the cutting load will be dependent on the rates of travel of both the elevation and rotation drives, as well as on hardness, depth of sumping, pick sharpness etc. The machine may therefore be viewed as a multivariable system having two inputs : the demanded angular speeds Ω_d and ω_d of rotation and elevation respectively, and two outputs, namely instantaneous radius r and cutting power consumption P_i. The system therefore takes the form shown below:
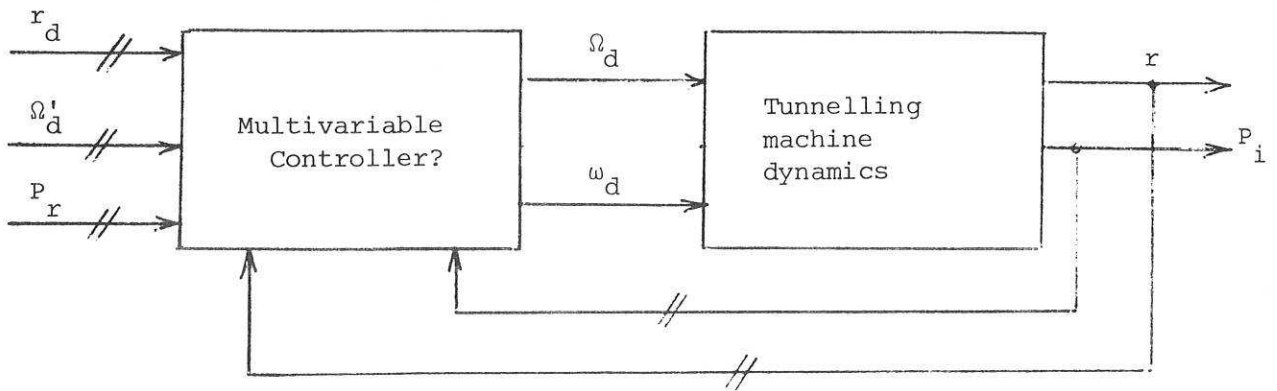
and the problem arises of how best to construct the <u>multivariable controller</u> to act on speed demands Ω_d and ω_d to make r follow the varying demanded radius r_d as closely as possible whilst P_i is regulated continuously at, or near, cutter motor rating P_r (if manual speed demand Ω_d' is set sufficiently high).

In this report and for the present we shall assume that, as on the existing machine, the profiler part of the control system affects only the demanded elevation rate ω_d (not Ω_d). For load control, again only part of the composite control system, two types will be considered as follows:

Type (a) : in which P_i affects demanded rotation rate Ω_d only

and   Type (b) : in which P_i affects both Ω_d <u>and</u> ω_d.

With the type (a) load controller, the composite control system is of the so called 'diagonal' type, each output setting only one of the two available inputs. Type (b) yields a nondiagonal multivariable composite system since one of the outputs at least drives both of the two inputs. Future work is needed, however, to determine the best intercoupling of r and P_i with Ω_d and ω_d.

3. <u>Type (a) (diagonal) controller design</u>

In previous reports it was assumed that mechanical power P_m output from the cutter motor was given by

$$P_m = k_h \, v \, (1 - e^{-Ts})/s \tag{1}$$

where k_h is a so called hardness factor (dependent also on depth of cut and pick sharpness), v is the linear velocity of the cutting head and T is the time interval between successive picks per plane. For circular tunnel profiles of constant radius R, v is clearly ΩR where Ω is the actual angular rotational speed of the boom but, in a varying radius situation, e.g. when cutting a straight vertical wall or horizontal floor, then v is given more generally by

$$v = \left((\frac{dr}{dt})^2 + (r \, \frac{d\theta}{dt})^2 \right)^{0.5} \tag{2}$$

where r is the instantaneous radius of the drum centre trajectory and θ is the angle of rotation: all as shown in Fig. 6.

Now because:

$$r = B \sin \alpha \simeq B \, \alpha \, , \, |\alpha| < 0.5 \text{ radians} \tag{3}$$

where B = boom length and α = angle of elevation in radians we obtain:

$$v = \left((B \, \frac{d\alpha}{dt})^2 + (r \, \frac{d\theta}{dt})^2 \right)^{0.5} \tag{4}$$

and since this angular elevation and rotational speeds are given by

$$\frac{d\alpha}{dt} = \omega \tag{5}$$

and

$$\frac{d\theta}{dt} = \Omega \tag{6}$$

we get

$$P_m = k_h \, B \, \{\omega^2 + \alpha^2 \, \Omega^2\}^{0.5} (1 - e^{-Ts})/s \tag{7}$$

Now for circular tunnels of constant radius R, controller gain settings, k_1, k_d and k_i (for the proportional, derivative and integral terms) have been deduced for stable and responsive control, regarding linear velocity

demand v_d, not Ω_d, as the system input. Because of the varying radius

situation now pertaining, Ω_d, must be regarded as the input and the gain

settings between power error P_e and Ω_d must now be taken as k_1/R, k_d/R and

k_i/R and radius R regarded as an external machine-gain parameter thus:
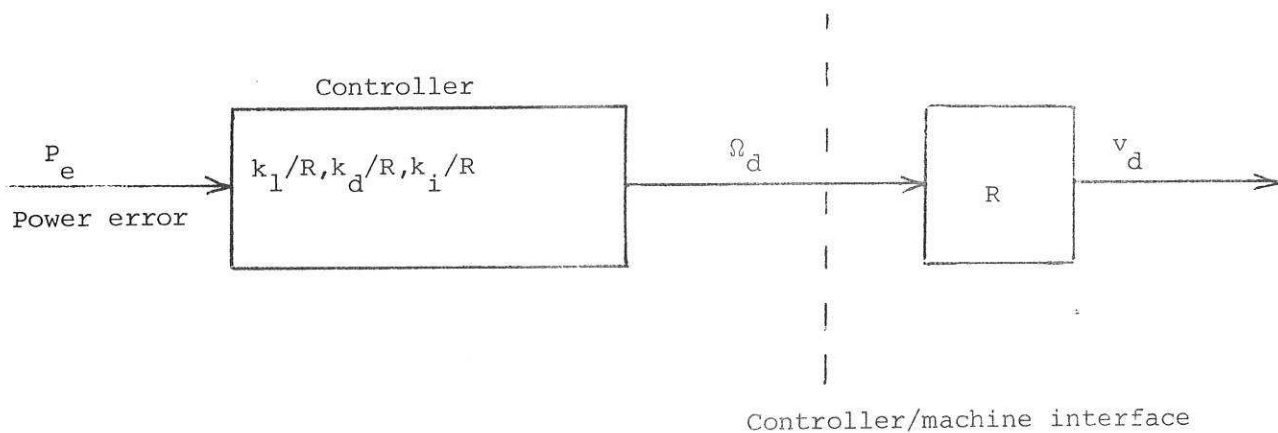
If R now changes to r in order to vary the cut tunnel profile, the net

gain between P_e and v_d should remain unchanged for the dynamic performance

to remain unchanged since the P_m/v relationship of equation (1) is unaffected

by radius. To preserve this constant relationship therefore, the true

controller gains between P_e and Ω_d must be rendered inversely proportional

to r thus:
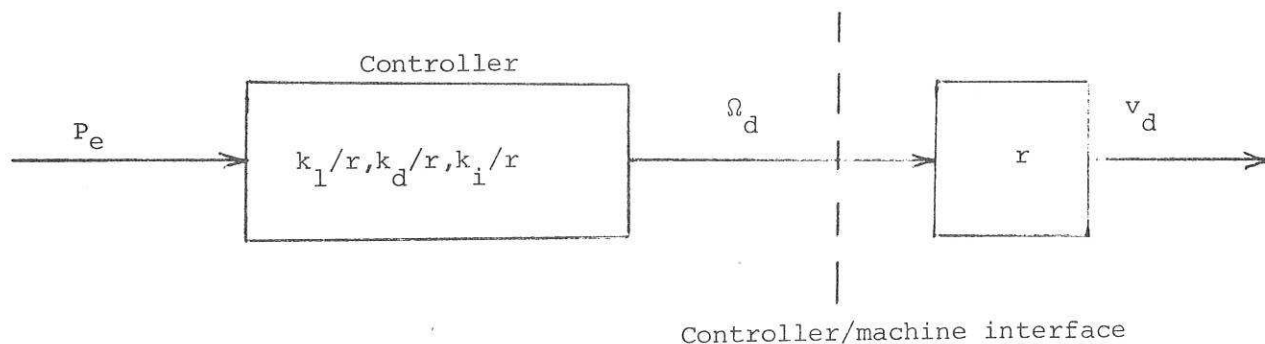
The type (a) controller therefore proposed is identical to that already

developed and described in previous reports but with its gains altered

adaptively in response to the r-measurement received from the profiler.

The profiler is left to operate on ω_d either via the existing valve

with the elevation swash angle, i.e. pump delivery, set constant, or via the swash itself with the in-line valve removed.

The overall dynamic system is illustrated in Fig. 7, and is currently undergoing simulation using a 380z microcomputer to act as the profiler whilst the VIDAC analogue machine is used to implement the load control system, apart from the non-linear operations $(\omega^2 + \alpha^2 \Omega^2)^{0.5}$ which will also be performed digitally due to analogue hardware limitations. Fig. 8 shows the program being used for profile generation. This could later form the basis of a digital system to be implemented on the machine itself if so desired. Fig. 9 shows the hybrid simulation scheme being used. Fuller details of the systems and their simulated performance will be given in a future report.

4. Type (b) (multivariable) controller design

An obvious problem arises with the type (a) controller in the special case where $L_d = 0$ and the desired flat floor lies below the machine rotation axis by only the cutting head radius. Here, demanded radius $r_d = 0$ at $\theta = 270^{\circ}$ and the load controller gains would become infinite. Although only a hypothetical situation, it does stress the fact that mechanical power P_m does depend on elevation speed ω as well as rotational speed Ω, and that ideally both $\underline{\Omega_d}$ and $\underline{\omega_d}$ should be actuated by the load controller. It is therefore necessary to consider separately the two components of P_m resulting from elevation and rotation:

Suppose the component of P_m produced by tangential velocity $r\,d\theta/dt$ is P_1 and that due to radial velocity dr/dt in P_2. Now from equation (1)

$$P_m = v\, Q(s) \tag{8}$$

where cutting dynamics $Q(s)$ are given by

$$Q(s) = k_h (1 - e^{-sT})/s \tag{9}$$

Now from Fig. 10 which shows the bite pattern of the cutting head whilst making a straight vertical cut it is clear that tangential velocity $r\,d\theta/dt$

is displaced by $\theta-180^\circ$ from the line of maximum bite so that

$$P_1 = r(d\theta/dt) \cos (\theta-180)Q(s) \tag{10}$$

and by similar reasoning

$$P_2 = (dr/dt) \sin (\theta-180)Q(s) \tag{11}$$

Furthermore, from the velocity diagram in Fig. 6 it is clear that

$$rd\theta/dt = v \cos(\theta-180) \tag{12}$$

whilst $\qquad dr/dt = v \sin (\theta-180) \tag{13}$

so that combining (8)(10) (11) (12) and (13) we get

$$P_1 = vQ(s) \cos^2(\theta-180) = P_m \cos^2(\theta-180) \tag{14}$$

and $\qquad P_2 = vQ(s) \sin^2(\theta-180) = P_m \sin^2(\theta-180) \tag{15}$

so that from the identity $\cos^2\beta + \sin^2\beta = 1$, as would be expected:

$$P_m = P_1 + P_2 \tag{16}$$

thus confirming the foregoing modelling.

Now since P_1 is produced by the tangential velocity it is clear that, ideally, this component of power should set rotational speed Ω_d in the control system, whilst P_2 sets ω_d. More precisely, from (10)

$$\Omega = d\theta/dt = P_1/\{r \cos (\theta-180) Q(s) \} \tag{17}$$

and from (11):

$$\omega = d\alpha/dt = (dr/dt)/B = P_2/\{B \sin (\theta-180)Q(s) \} \tag{18}$$

and since previously <u>for circular profile cutting</u>:

$$\Omega = P_m/r \ Q(s) \tag{19}$$

and the control law successfully demonstrated was

$$\Omega_d = (\dot{P}_e/r)K(s) \tag{20}$$

where P_e = power error and K(s) denotes the 3-term controller transfer function (designed in R.R. 220,230 and 237), then the control law that should ideally be adopted <u>for vertical downward cutting</u> should be

$$\Omega_d = P_e \, K(s)/\{r \cos(\theta - 180)\} \tag{21}$$

and
$$\omega_d = P_e \, K(s)/\{B \sin(\theta - 180)\} \tag{22}$$

It therefore follows from (14) and (15), which can be rewritten

$$P_1/\cos(\theta - 180) = P_m \cos(\theta - 180) \tag{23}$$

and
$$P_2/\sin(\theta - 180) = P_m \sin(\theta - 180) \tag{24}$$

and knowing

$$P_e = P_r - P_i \tag{25}$$

where $P_i = P_m k_m /(1 + T_m s)$ (26)

(where k_m and T_m are the induction motor efficiency^{-1} and time constant)

that the control laws may be expressed in terms of measurements P_i and

reference P_r thus (for vertical <u>downward</u> cutting)

$$\Omega_d = (P_r - P_i) \cos(\theta - 180) \, K(s)/r \;, \; 180 < \theta < 180 + \theta_m \tag{27}$$

and

$$\omega_d = (P_r - P_i) \sin(\theta - 180) \, K(s)/B \;, \; 180 < \theta < 180 + \theta_m \tag{28}$$

where, from Fig. 6, θ_m is known from the demanded profiles dimensions L_d

and R_d thus :

$$\theta_m = \tan^{-1} L_d/R_d \tag{29}$$

By similar reasoning, for horizontal cutting, the control law should

take the form

$$\Omega_d = (P_r - P_i)\sin(\theta - 180) \, K(s)/r \;, \; 180 + \theta_m < \theta < 360 - \theta_m \tag{30}$$

and

$$\omega_d = -(P_r - P_i) \cos(\theta - 180) K(s)/B \;, \; 180 + \theta_m < \theta < 360 - \theta_m \tag{31}$$

whilst for vertical <u>upward</u> cutting

$$\Omega_d = -(P_r - P_i)\cos(\theta - 180) K(s)/r \;, \; 180 + \theta_m < \theta < 360 \tag{32}$$

and

$$\omega_d = -(P_r - P_i)\sin(\theta - 180) K(s)/B \;, \; 180 + \theta_m < \theta < 360 \tag{33}$$

Consideration of control laws (30) and (31) shows that the aforementioned zero-r problem is now resolved since, in this situation, $\theta_m = 0$ and the $\sin(\xi-180)$ term in equation (30) remains zero throughout the horizontal traverse, (so counterbalancing the zero r-term in the denominator) and control is left to equation (31) acting on the elevation only.

4.1 Implementation

The control laws (27) thro' (33) will be tested by simulation but will require cam-switched sine/cosine resolvers to be installed on the Middleton machine between the power transducer and the two existing controllers (modified according to RR 220, 230 and 237). Ideally, radius modulation of the rotation drive gains would also be needed to satisfy (27), (30) and (32). A VIDAC computer might be used to provide the necessary sine cosine extraction and the radius modulation in a preliminary underground trial at Middleton. The exercise goes well beyond the existing contract, of course, this being confined to circular cutting control, and an additional contract or extension would be necessary.

5. Conclusions

Trials carried out at Middleton on the Eimco machine have shown that the modified prototype of the Cadley Hill machine controller will operate successfully on the real electro-hydraulic servo system and that the simulation also continues to give consistent predictions with the addition to the loop of this real mechanical hardware. Cutting trials are now eagerly awaited

An initial investigation of the problems and load control implications of cutting D-shaped tunnels has been presented showing the intimate interconnection of the profile and cutting dynamics in the system. A simulation demonstrating this intercoupling has been outlined and is being set up at Sheffield on a hybrid computer system.

Two types of load control (a) and (b) have been outlined and are being appraised on this simulator. Type (a) acts on the rotation drive only whilst type (b) acts on both rotation and elevation drives simultaneously. Profile control has been assumed to act only on elevation as with the present cam-controller at Middleton. The problem is a non-linear multivariable one, however, and is deserving of special consdieration for an ideal solution. In the meantime, the type (a) and type (b) controllers could be tried out in prototype form at Middleton Mine (after the simulation work at Sheffield) using a VIDAC computer temporarily for radius-gain-modulation needed by types (a) and (b) and for the sine/cosine extraction needed on type (b). A contract extension would be necessary for this additional work.

P_i (kw)

200

100

0

t

3

4

20 s

v

Full swash

Zero swash
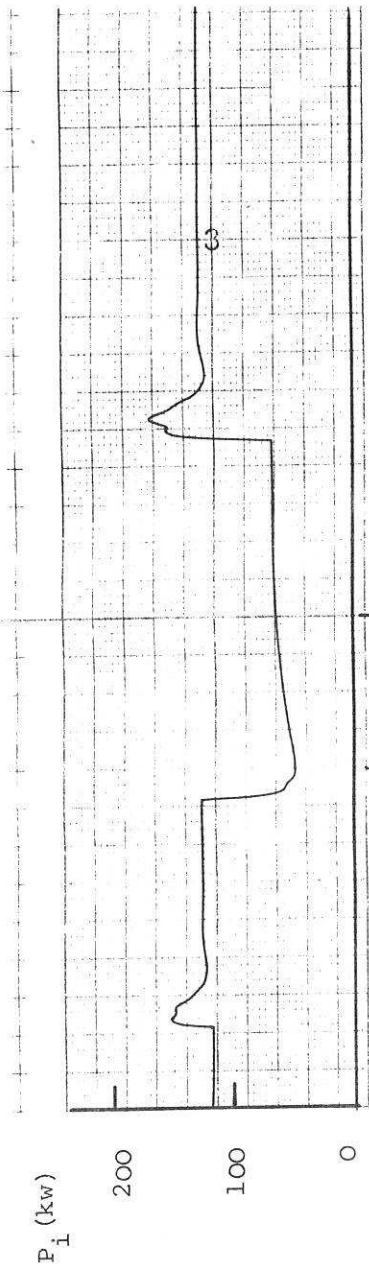
Settings:

Rate 45%
Gain 15%
Preset Load 37%
Rate ref. 40%
Derivative 100%
Current Feedback 100%
Load 25%
Null 50%

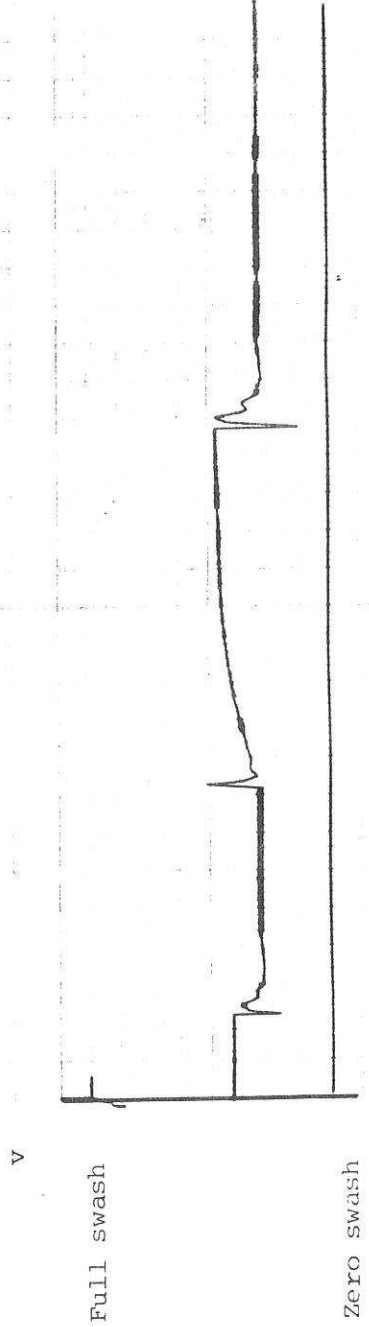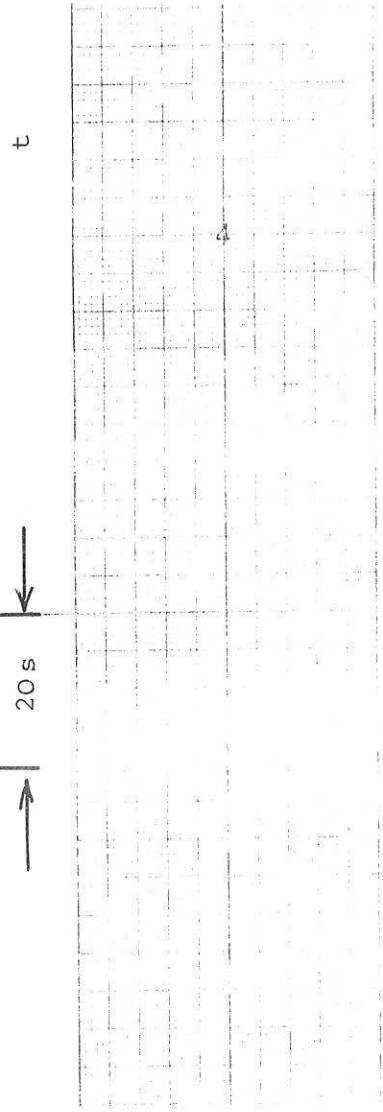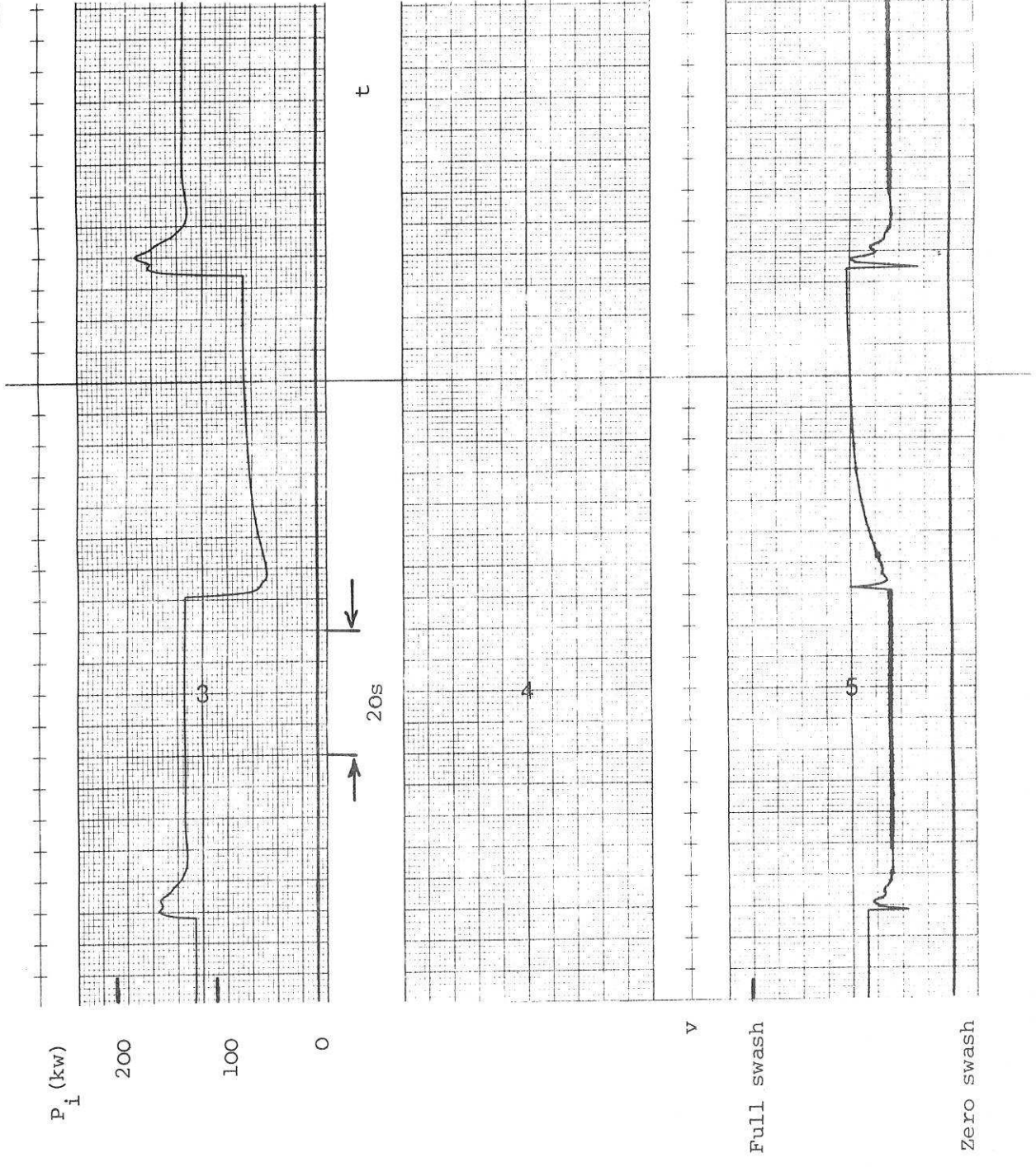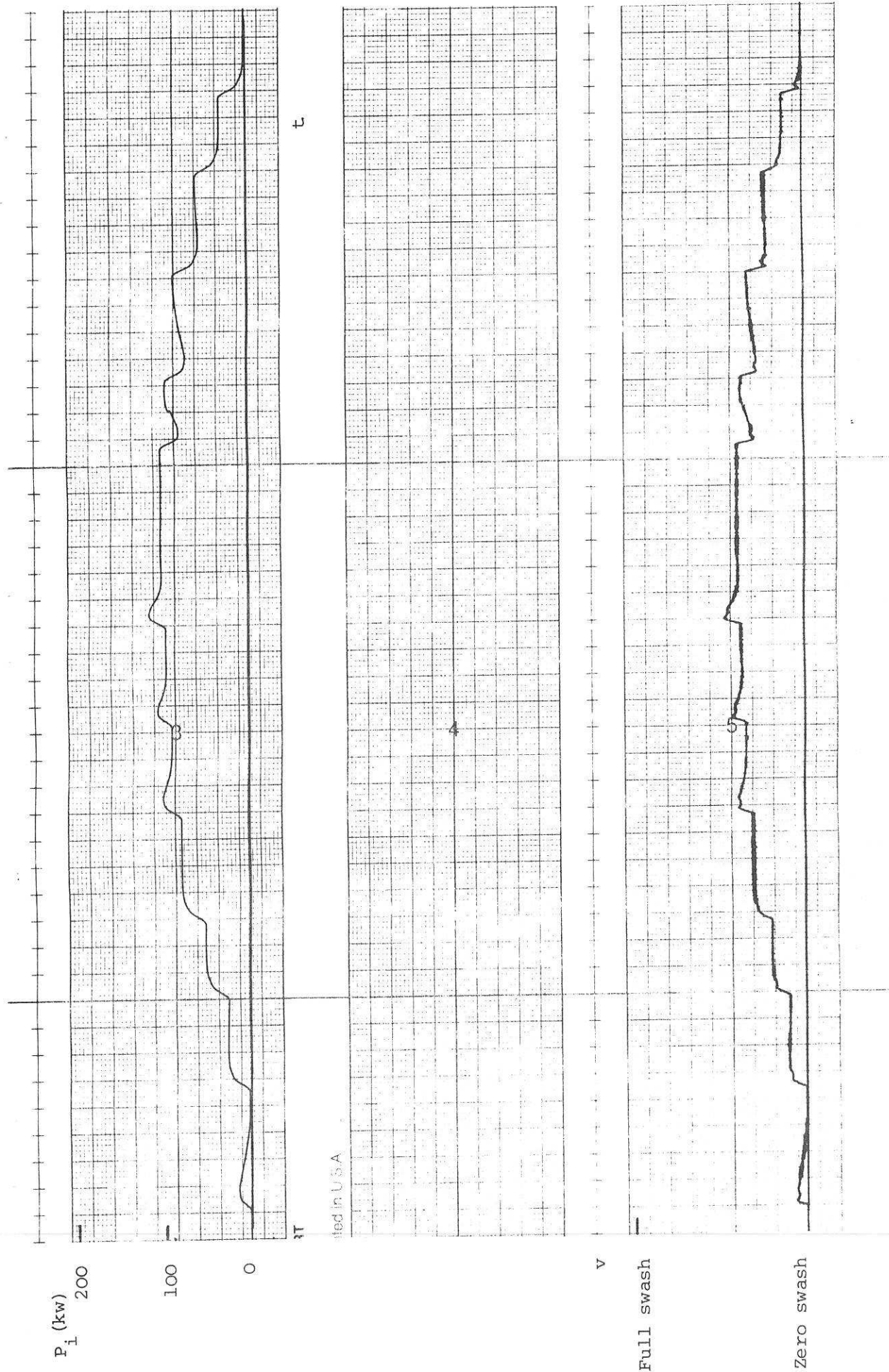Fig. 1 Response of Eimco machine to simulated ± 50% step changes in rock hardness

Fig. 2 Response of Eimco machine to simulated ± 50% step changes in rock hardness

Settings:

Rate 60%
Gain 10%
Preset load 37%
Rate Ref. 40%
Derivative 100%
Current Feedback 100%
Load 25%
Null 50%

P_i (kw)

200

100

0

t

v

Full swash

Zero swash

20s

3

4

5

P$_i$ (kw)

200

100

0

RT

Full swash

Zero swash

v

t

Settings

Rate 45% Derivative 100%

Gain 15% Current Feedback 100%

Preset load 37% Load 25%

Null 50%

Fig. 3 Response of Eimco machine to 5% changes

in manual rate-reference

P_i (kw)

200

100

0

t

v

Full swash

Zero swash
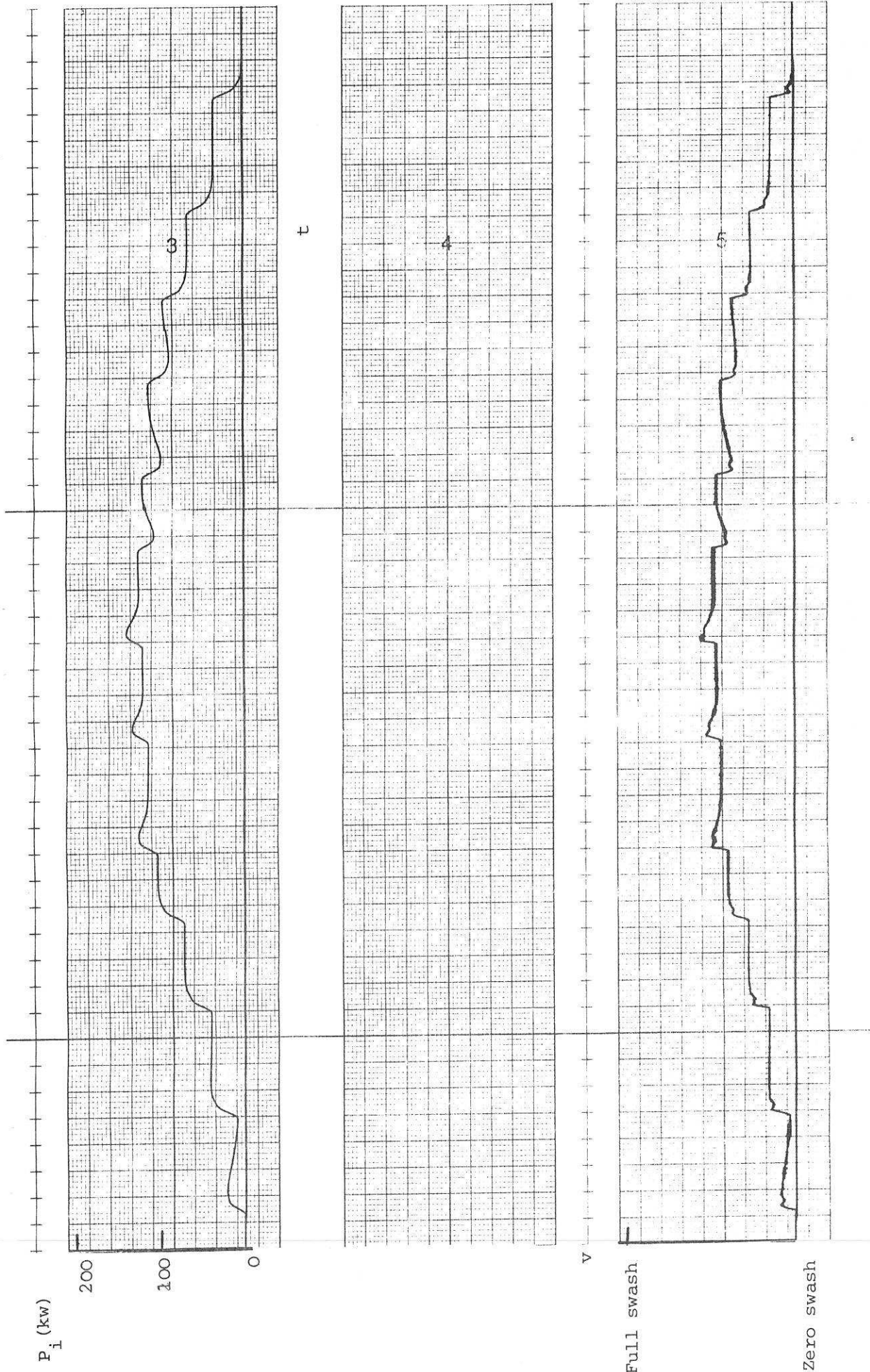
Settings:
Derivative 100%
Rate 45%   Current feedback 100%
Gain 15%   Load 25%
Preset load 50%   Null 50%

Fig. 4   Response of Eimco machine to 5% changes in manual rate reference
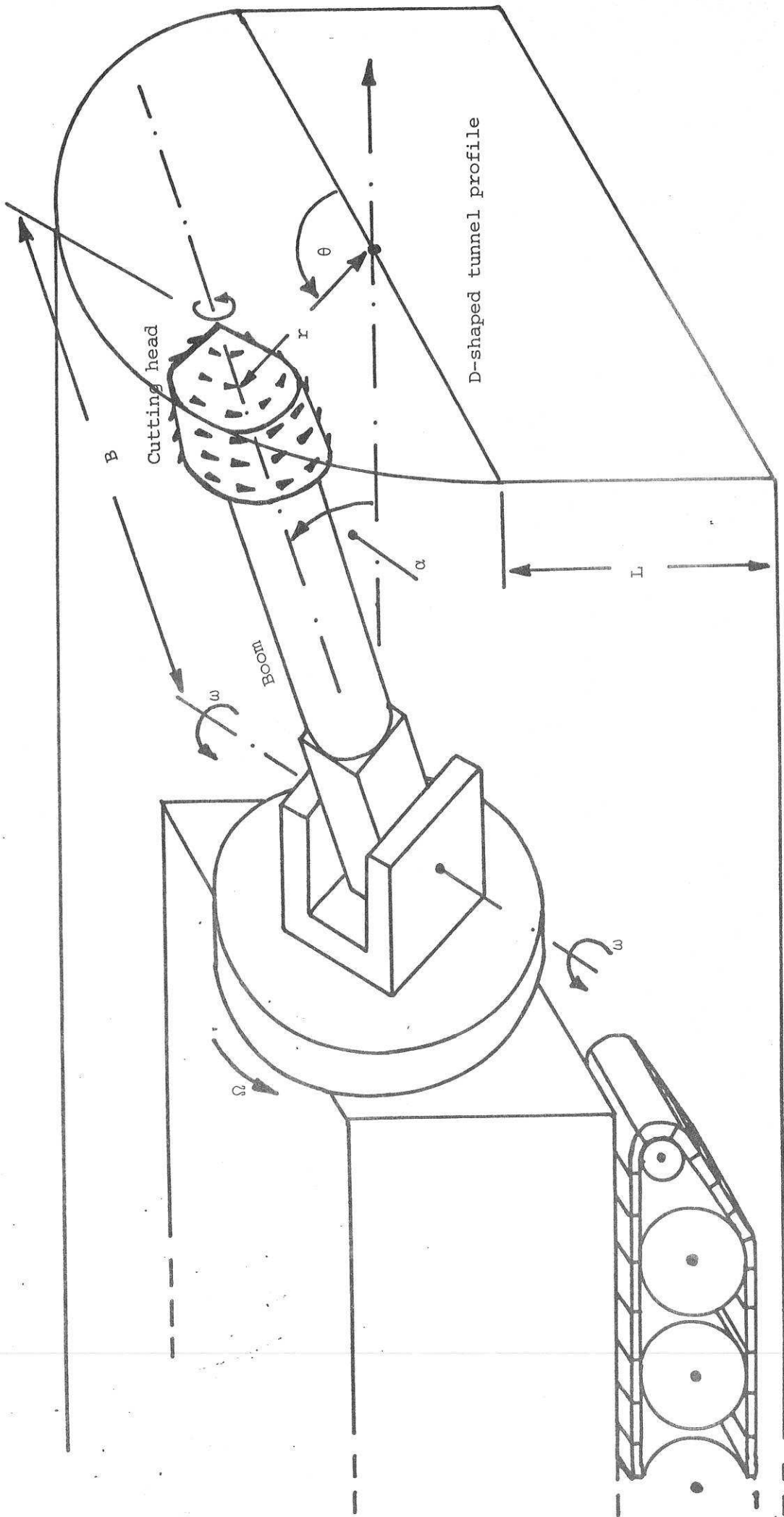
Fig. 5 Tracked tunnelling machine cutting D-profile

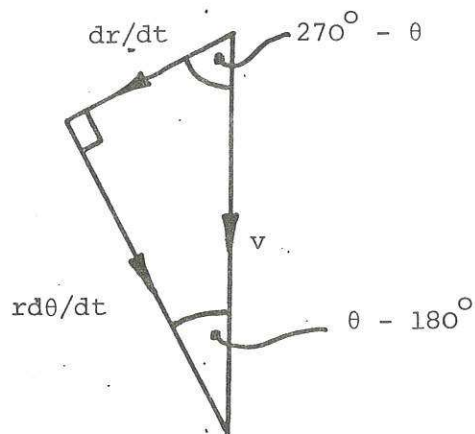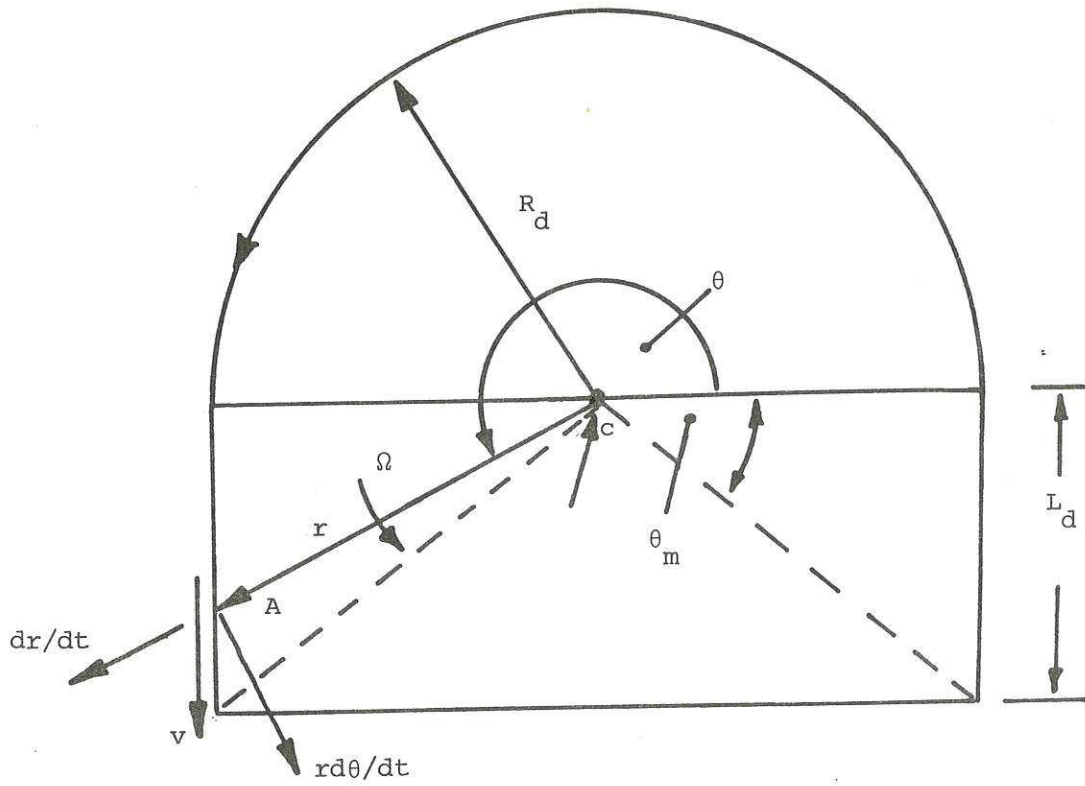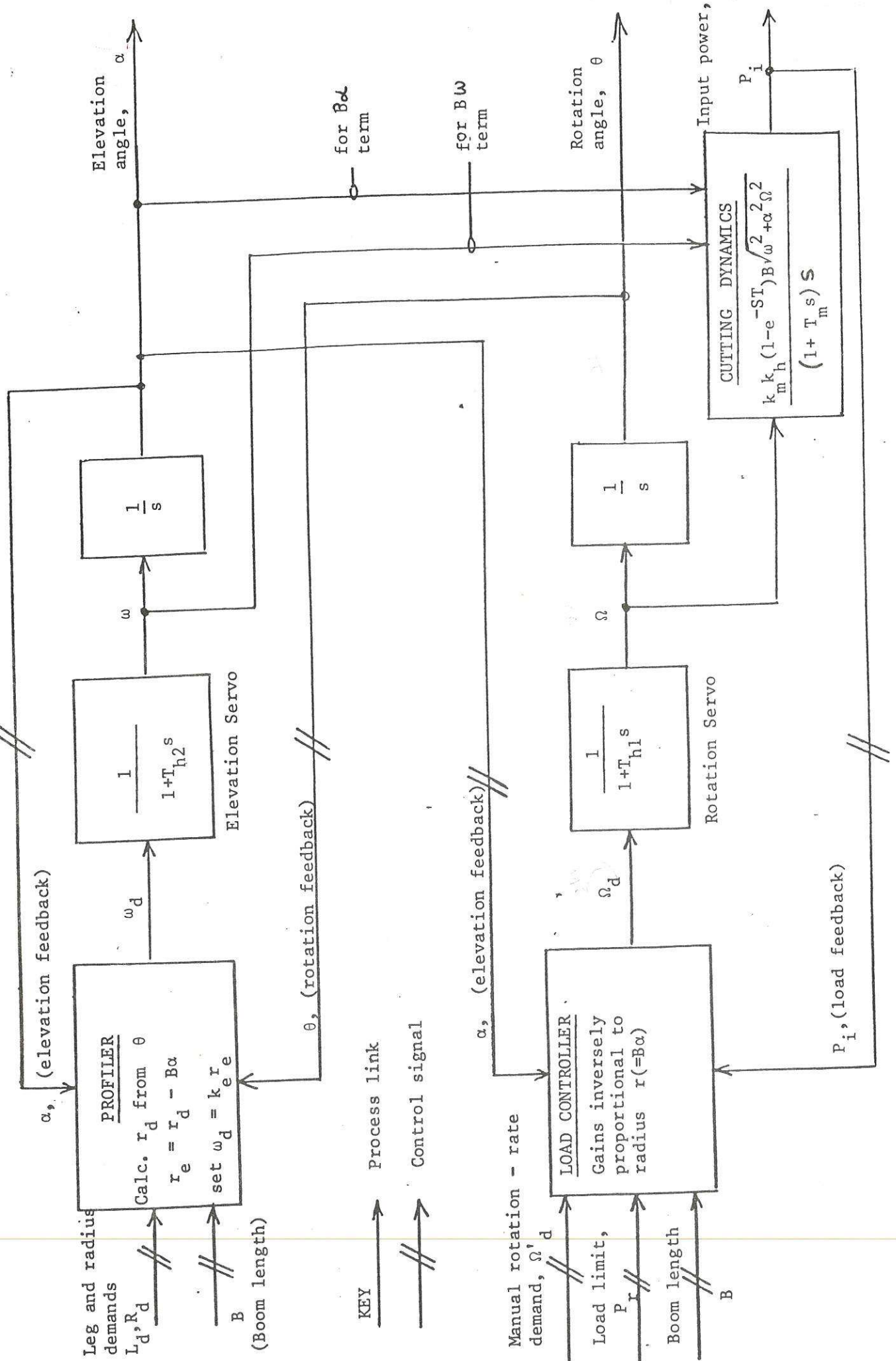Fig. 6 Ideal tunnel profile and velocity diagram

Fig. 7. Interaction of Tunnel Profile and Load-Control Systems

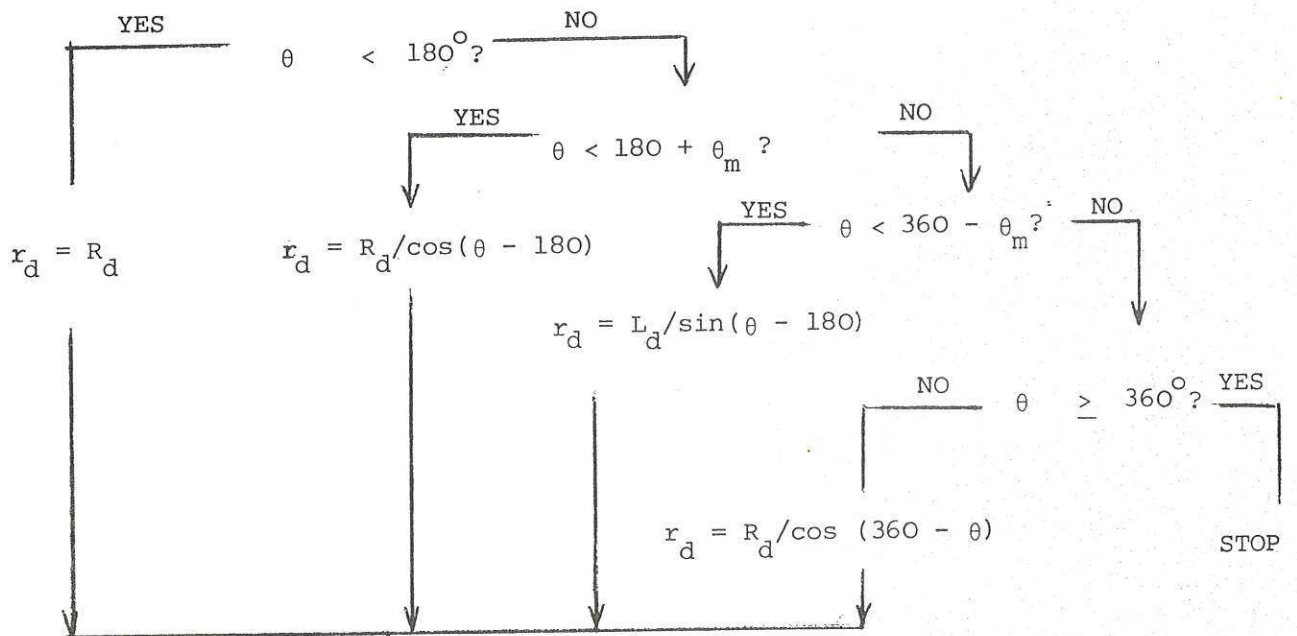Read B, $\theta(o)$, $\alpha(o)$, k_e, T_{h2}, ΔT

$\phi = \ = \exp(-\Delta T/T_{h2})$, $\Delta = 1 - \phi$

Read L_d, R_d (from operator)

$\theta_m = \tan^{-1} L_d/R_d$

Read Ω from VIDAC cutting-load simulator

$\theta = \theta + \Omega\Delta T$

YES $\theta \ < \ 180^{\circ}$?   NO

YES $\theta < 180 + \theta_m$?   NO

YES $\theta < 360 - \theta_m$?   NO

$r_d = R_d$

$r_d = R_d/\cos(\theta - 180)$

$r_d = L_d/\sin(\theta - 180)$

NO $\theta \ \geq \ 360^{\circ}$?   YES

$r_d = R_d/\cos(360 - \theta)$

STOP

$r_e = r_d - B\alpha$

$\omega_d = k_e \, r_e$

$\omega = \phi\omega + \Delta\omega_d$

$\alpha = \alpha + \omega\Delta T$

$r = B\alpha$

$v = B\sqrt{\omega^2 + \alpha^2 \Omega^2}$

Output r_d, r, θ to plotter

Output v and r to VIDAC cutting simulator

Wait time step ΔT

Fig. 8   Flowchart for digital profiler

Fig. 9 General arrangement for Hybrid Simulation of Tunnel Profile and Cutting Load Systems