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THE EFFECT OF LARGE-SCALE STRUCTURES ON THE STABILITY OF COAL-FACE STEERING

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SYNOPSIS

Simplified models of piecewise rigid support structures for power-loaders operating on longwall coal-faces are shown to be amenable to analysis by z-transform methods. Such analysis predicts that increasing the length of the structure's subsections sufficiently (compared to the inherent delay within the machine's vertical steering system) should stabilise the vertical steering of the entire coal face. Increasing the width of the structure to embrace more than two consecutive cut floors is shown analytically to eliminate the need for electronic tilt-feedback in control systems.

In general terms, these analytical predictions are shown to hold good in detailed simulations of the system that eliminate the simplifications demanded by the analytical method. The general conclusion of the work is therefore that an increase in the size of support-structure segments can potentially reduce the complexity of steering control systems. The size-increase must be substantial, e.g. to 4 to 5 times the size of conventional structures.

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1. List of Symbols and Abbreviations

A	Function of k (eqn. 11), a-gain parameter $= k - 1 - M^{-1}$
a.f.c.	Armoured Flexible Conveyor
B	Function of k (eqn. 12)
Del(X)	Backward distance shift X
d.p.	Dynamic programming
G_s	Steering system transfer operator
H(z)	z-transfer function of G_s and a.f.c combined
h	height profile of machine support structure
i	Integer in Section 3
J	Deflection of floor cutting drum
k	Height-gain of steering system controller
k_c	value of k for critical multipass stability
k_g	Tilt gain of steering system controller (=1.0 in Section 3)
ℓ	Distance travelled in along-face direction (Figs. 1 and 4)
L	Length of coal-face
machine	Power-Loader
m	Integer $(= L/X_p)$
M	Integer (= Structure width/W)
n	Cut (pass) number
p	Integer used in Section 3
power-loader	Machine travelling along coal face for cutting and loading coal onto a.f.c.

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r	X_p/X = integer in Section 3
s	Laplace variable
Structure	Structure forming the track on which the power-loader rides.
W	Width of cut
X	Delay distance between coal sensor and cutting drum
X_1	Distance associated with coal sensor time constant
X_2	Distance associated withhydraulic steering system actuator
X_p	Length of discrete subsection (alongface) of machine support structure
y	Height profile of floor cut by power loader (= thickness of floor coal left in a flat seam)
y^*	y-function sampled at intervals X_p (Section 3) and W (Section 4)
z	Independent variable for z-transforms (= e^{+sX_p} in Section 3, = e^{sW} Section 4).
*	Indicates sampled function at intervals X_p (Section 3) and W (Section 4)
~	Transform
$\tilde{y}(s)$	Laplace transform of y
$\tilde{y}(z)$	z-transform of y
α	Tilt (in radians) of machine-support structure in the direction of face-advance (Figs. 1 and 4)
θ	Angle swept out by vector in z-plane in describing unit circle.

2. INTRODUCTION

2.1 Conventional Systems

The present paper is offered as a companion to two[1,2] previously published that were also devoted to the effect of the supporting structure of a longwall coal-cutting machine on the vertical steering of entire coal-face systems. The systems there examined were of the conventional type, illustrated in Fig. 1, in which the coal cutting machine rides on the semiflexible structure of a scraper-chain conveyor (a.f.c) extending along the entire 100 to 300 m. length of the coal-face. In those papers, the smoothing effect of the conveyor structure on the undulating floor cut by the vertically-ranging cutting-drum as the conveyor is pushed forward in the face-advance direction (fig. 1), between successive cuts, was examined. The broad conclusion reached was that, sufficient elasticity built into the a.f.c joints (i.e. between consecutive trays) should smooth out oscillations in the cut floor produced by resonant[†] vertical steering systems acting on the ranging cutting drum as so prevent repeated excitation of the system resonance from cut to cut, thus producing stable behaviour over a sequence of cuts. Detailed simulations conducted on conventional conveyor models lacking this elastic restraining force, and relying only on tray rigidity for any smoothing effect, failed to produce stable behaviour. These findings are in broad accordance with practical experience undergound.

2.2 Thick Seam Systems

With the discovery of coal seams of 4 to 5m. in thickness in recent
and
years, in the Selby coal field, the Doncaster/West Midlands Areas and else-where, it has become increasingly important to investigate and develop modified longwall systems for their extraction. Coal-cutting machines

[†]Machine steering systems are fundamentally resonant because of transport delays involved in (a) the sensing of the roof coal thickness behind the cutting drum and (b) the geometrical offset between drum centre and the rear skids of the machine.

(power-loaders) of progressively increases size and rating have naturally emerged over the past 10 to 15 years in response to demands for higher productivity and to extract bands* of coal that previously might have been left behind as uneconomic. Machines equipped with double cutting drums, one for roof- and one for floor-coal extraction have been one outcome of this trend. The increase in height of the excavation has, of course, also demanded the development and introduction of more massive hydraulic roof-supports to maintain the temporary walkway and passage for ventilation and service along the face. Figs. 2 and 3 allow a comparison of the scale of equipments needed for conventional faces extracting seams of 1 to 2 m in height and for the extraction of the very thick seams (4 to 5m) now contemplated.

The key point to note from Figs 2 and 3 is that whereas, in conventional systems, the a.f.c structure supports the weight of the power-loader and provides an anchor-beam for drawing forward the self-advancing roof-supports between cuts, in really thick seam extraction, the a.f.c can no longer perform these functions. Not only are the power-loader and roof-supports now greatly increased in scale relative to the a.f.c., but the power-loader body must now ride much higher to avoid the need for an impractically long boom for the roof-cutting drum. As shown in Figs. 3 and 4 therefore, large bench-type structures must now be introduced to provide the machine track and to anchor the supports. The a.f.c. now reverts to its basic function of coal conveying only. The system is termed a bench mining system and was first proposed by Robson[3] in the Doncaster Area.

Initial concepts involved a so-called full-face bench, i.e. a rigid bench extending along the full face-length, so apparently avoiding

* Some coal seams comprise distinct bands of coal of differing quality separated by dirt bands of thickness range from, say, 2 to 20cm.

the need for vertical steering other than at the face ends. Design studies [4,][5] rapidly revealed the impracticallity of constructing such a structure underground however, and instead the structure was subdivided into a number of loosely coupled subsections each some 4.5m in length (i.e. each accommodating three a.f.c. trays in-line) and again resting on the floor produced by the floor-cutting drum along the face. In basic principle therefore the machine track remains as a sequence of loosely articulated rigid subsections as with conventional a.f.c. systems. The subsections are now three-times longer, however and bridge many (typically five to seven) cut floors whereas a.f.c trays bridge only two.

This particular type of system therefore highlights the need to examine the effect of using larger machine support structures on coal-face control and on its vertical steering particularly. It should be re-emphasised, however, that the bench system is but one manifestation of a trend towards the adoption of larger units of mining machinery generally throughout the U.K. and abroad and there therefore exists a strong motivation for investigating the effect of machine size on mining control problems. The paper reports some of the findings of a collaborative theoretical study of this question sponsored at the University of Sheffield by the Mining Research and Development Establishment (M.R.D.E) of the National Coal Board (N.C.B).

2.3 Nature of the Investigation

The theoretical investigation reported here is divided into two parts, one concerned with increasing the length (along-face) of subsections of the support structure and the others with increasing structure width (in direction of face-advance). The analytical investigations, necessitating certain simplifying assumptions, are described with supporting simulation results in Section 3 and 4 respectively. In Section 5 the extent to which these 'ideal' system results carry over to a fuller system simulation is examined and general rules of thumb established for a stabilising structure size.

3. INCREASING STRUCTURE LENGTH

3.1 System Model Assumed

For initial analytical studies it is assumed that the vertical steering system is proportional, linear and that its only significant dynamics are due to the delay-distance, X, between cutting-drum and coal-sensor (see Fig. 1). Cut-floor height, $y(n,\ell)$ and the height profile, $h(n,\ell)$, of the machine-support-structure (where n denotes cut-number and ℓ distance cut along-face) are thus related as follows:

$$y(n,\ell) = G_s h(n,\ell) \tag{1}$$

where transfer-operator G_s is given by

$$G_s = 1/\{1 + k\ \mathrm{Del}(X)\} \tag{2}$$

Del (X) denotes the sensor distance shift and k the controller gain. The operator is identical to that used in the companion papers (eqns. 17 and 5 respectively) for analytic studies with the sensor time-constant neglected. Equations (1) and (2) are in fact derived from

$$y(n,\ell) = h(n,\ell) - k\ y(n,\ell-X) \tag{3}$$

For simplicity of analysis, we shall here consider the length, X_p, of each subsection of the support structure to be an integer number of delay distances, i.e.

$$X_p = r\,X\ ,\ r = 1,2,3,4\ldots\ldots \tag{4}$$

(The case of r = o corresponds to the so called 'rubber-conveyor assumption' examined in the previous papers and shown to predict system instability for all k).

To complete the process description, it is necessary to relate the structure profile, $h(n+1,\ell)$, to the cut floor* profile, $y(n,\ell)$, upon which

* In previous papers, the cut floor profile was denoted by $y(n,\ell) + z(n,\ell)$, y there representing floor coal thickness left by the machine and z representing the coal seam's natural undulations. For stability studies, external disturbance z can be set to zero and this is done in the present paper.

the structure rests after its pushover, in the face-advance direction, between cuts n and n+1. In the companion papers[(1),(2)] the a.f.c. was likened to two parallel chains of rods, one representing the leading edge of the structure (in the face-advance direction, Fig. 1) and the other the trailing edge. One pair of rods represented the side channels of one tray of the a.f.c. structure and the stiffness of the intervening deckplate was neglected such that the two chains of rods could undulate independently of one another. Thus all the structure stiffness was assumed to be concentrated in the side channels. In the analysis of this Section (3) we assume the machine support structure to have the same basic form, but we allow the length X_p of the subsections (rods) to be increased by alter ation of integer r.

In the first companion paper the rods were taken to be completely rigid and not to penetrate the cut floor. These assumptions together precluded analytical solution and demanded an expensive computed solution by dynamic programming. In the second paper, analytic solutions were obtained by allowing elasticity of the structure and some penetration of the cut floor profile by accepting the presence of a superimposed layer of fine coal. In the present investigation analytic solution has proved possible by retaining rod rigidity (and free angular play at the joints) but allowing some floor penetration, i.e. by a compromise. The type of floor fitting assumed is illustrated in Fig. 5 in which only the ends of the rods rest on the cut-floor. In between, valleys are bridged whilst hills are planed flat. (Alternatively the fit illustrated can be regarded as that produced on a hard floor by standing each rod on short pedestals at each end: their action being not unlike that of the beams shown in Figs. 3 and 4).

With this type of fit, the support structure merely samples the cut floor profile at distances iX_p, where i = 0,1,2...m and mX_p = face length L. Between samples the structure linearly interpolates between the two end-heights. Mathematically therefore the structure's response may be described thus:

$$h(n+1,\ i\ X_p) = y(n,\ iX_p)\ ,\ i = 0,1,2,\ldots m \qquad (5)$$

and

$$h(n+1,\ell) = \left[y(n,iX_p)\{(i+1)X_p - \ell\} + y\{n,(i+1)X_p\}(\ell - iX_p)\right]/X_p$$

$$i\ X_p \leq \ell \leq (i+1)X_p \qquad (6)$$

The composite system model comprising steering system (eqn. 3) and support structure (eqn. 6) may thus be regarded as a sampled-data system, the stability of which may be investigated by z-transform methods. It is first necessary to determine the z-transfer-function H(z) of this composite system illustrated in block-diagram form in Fig. 6.

3.2 z-Transfer-function

To find H(z) we must first determine the response of $y^*(n,\ell)$ (i.e. at point Q in Fig. 6 to a unit impulse applied at point P. The triangular impulse response of the structure alone is indicated in Fig. 6 and we must now determine its effect on steering system G_s at delay intervals X, i.e. at $\ell = i\ X$, $i = -r,\ -(r-1),\ -\ (r-2)\ldots 0,1,2,$ etc. where discontinuities will occur. For this purpose argument n is dropped for simplicity of notation, i.e. we set

$$h(i) = h(n,\ iX) \qquad (7)$$

and $$y(i) = y(n,iX) \qquad (8)$$

n and X being common to the arguments both variables. Knowing that h(i) follows the triangular pattern of Fig. 6, y(i) is readily calculated analytically by recursive use of equation 3, written in the new notation thus

$$y(i) = h(i) - k\ y(i-1) \qquad (9)$$

The procedure is carried out in Table 1.

TABLE 1 Calculation of unit impulse response of H(z)

i	$h(i)$	$y(i) = h(i) - k\,y(i-1)$ (eq^{n}. 9)
$-r$	0	0
$-(r-1)$	r^{-1}	r^{-1}
$-(r-2)$	$2r^{-1}$	$(2-k)r^{-1}$
$-(r-3)$	$3r^{-1}$	$(3-2k+k^{2})r^{-1}$
$\vdots$		
0	$rr^{-1} = 1$	$\{r+(r-1)(-k) + (r-2)(-k)^{2} \ldots\ldots (-k)^{r-1}\} = Ar^{-1}$
1	$(r-1)r^{-1}$	$\{(r-1) - k\,A\}r^{-1}$
2	$(r-2)r^{-1}$	$\{(r-2) + (r-1(-k) + A(-k)^{2}\}r^{-1}$
3	$(r-3)r^{-1}$	$\{(r-3) + (r-2)(-k) + (r-1)(-k)^{2} + A(-k)^{3}\}r^{-1}$
$\vdots$		
r	0	$\{(-k) + 2(-k)^{2} \ldots (r-1)(-k)^{r-1} + A(-k)^{r}\}r^{-1}$
		$= \{B + A(-k)^{r}\}r^{-1}$
r+1	0	$-k\{B + A(-k)^{r}\}r^{-1}$
r+2	0	$(-k)^{2}\{B + A(-k)^{r}\}r^{-1}$
$\vdots$		
2r	0	$(-k)^{r}\{B+A(-k)^{r}\}r^{-1}$
$\vdots$		
3r	0	$(-k)^{2r}\{B + A(-k)^{r}\}r^{-1}$
qr	0	$(-k)^{(q-1)r}\{B+A(-k)^{r}\}r^{-1}$

Now the desired output of $H(z)$ is $y^*(n,\ell)$ which is the function obtained by sampling $y(n,\ell)$, and therefore $y(i)$, at intervals X_p $(= r X)$, i.e. at $i = -r, o, r, 2r, 3r$, etc. and from inspection of Table 1 therefore we deduce that, if $\delta(\ell-Y)$ denotes a unit impulse at $\ell = Y$, then:

$$y^*(n,\ell) = Ar^{-1}\,\delta(\ell) + \sum_{q=1}^{\infty} (-k)^{(q-1)r}\{B+A(-k)^r\}r^{-1}\delta(\ell-rX) \tag{10}$$

where the parameters A and B from the table are given by

$$A = \sum_{i=1}^{r-1} (r-i)(-k)^i \tag{11}$$

and

$$B = \sum_{i=o}^{r-1} i(-k)^i \tag{12}$$

On taking Laplace transforms in s w.r.t. ℓ gives

$$\tilde{y}^*(s) = Ar^{-1} + \frac{\{B+A(-k)^r\}}{(-k)^r r}\sum_{q=1}^{\infty} \{(-k)^r e^{-rXs}\}^q \tag{13}$$

and on substituting z for e^{rXs} (recalling that the structure samples at distances $X_p = rX$) we obtain the z-transform, $\tilde{y}(z)$, of $y^*(n,\ell)$ and hence z transfer function $H(z)$. Hence:

$$H(z) = \tilde{y}(z) = Ar^{-1} + \frac{\{B+A(-k)^r\}}{(-k)^r r}\sum_{q=1}^{\infty} \{(-k)^r z^{-1}\}^q$$

$$= -B(-k)^{-r}\,r^{-1} + \frac{\{B+A(-k)^r\}}{(-k)^r r}\sum_{q=o}^{\infty} \{(-k)^r z^{-1}\}^q$$

$$= -B(-k)^{-r}r^{-1} + \frac{\{B+A(-k)^r\}z}{(-k)^r r\{z-(-k)^r\}} \tag{14}$$

Further simplification of equation 14 gives finally

$$\boxed{H(z) = \frac{zA + B}{r\{z-(-k)^r\}}} \tag{15}$$

3.3 Multipass Stability

The long delay term z^{-m} ($\equiv e^{-mX_p s} = e^{-Ls}$) in Fig. 6 accounts for the fact that the support structure rests on profile $y(n-1,\ell)$ whilst $y(n,\ell)$ is being formed by the cutter. Thus for the rapidly spiralling locus of the loop's z-transfer-function, $H(z)\ z^{-m}$, not to embrace the critical -1 point (in its own plane) as z describes the unit circle (i.e. the stability boundary) in the z-plane, it follows that, since m is a very large integer, the multipass stability criterion† must be

$$|H(z)| < 1.0, \quad z = 1 \angle \theta, \quad o \leq \theta \leq 2\pi \tag{16}$$

Now from equations (11) (12) and (15) it is clear that $H(z)$, for the simplified system here considered, is a function only of z and the system parameters k and r. For stability of the steering system itself on any single pass it is, of course, necessary that controller gain k be constrained such that

$$o < k < 1.0 \tag{17}$$

(6)

and under such circumstances it is readily shown from equations (11) and (12) that, for any chosen value of integer ratio r.

$$A + B > O \tag{18}$$

and $\quad A - B > O$

The locus of the numerator expression (15) for $H(z)$ as z varies round the unit circle is therefore itself a circle enclosing the origin. The origin is also included inside the circular denominator locus because of constraint (17). Thus, by substituting (15) in stability condition (16) it is clear that the necessary and sufficient requirements for multipass stability are simply

†The condition is similar to that used in the earlier papers(1),(2), viz $|G_s(j\omega)G_c(j\omega)| < 1.0$, $-\infty < \omega < +\infty$, for continuous steering systems $G_s(s)$ and continuous conveyor models $G_c(s)$.

$$A + B < r(1-k^r) \text{ and } A-B < r(1+k^r) \;, \; r = 2,4,6, \text{ etc.} \tag{19}$$

whilst

$$A + B < r(1-k^r) \text{ and } A-B < r(1-k^r) \;, \; r = 1,3,5, \text{ etc.} \tag{20}$$

Although a more general treatment is possible[6] in the following section (3.4) we confine ourselves to varying r between a few values of practical interest. Comfirmatory simulation results of the system analysed are also presented or stated. Fuller results have been reported by Edwards and Yazdi[6].

3.4 The effect of varying r

Setting $r = 1$ (i.e. $X_p = X$ in condition (2) reveals that for stability we must have that

$$k^2 + 2k + 1 < 0 \tag{21}$$

which is clearly impossible for any gain $k > 0$. Multipass stability is therefore unattainable in this situation and the conclusion is confirmed by the traces of Fig. 7 showing the attempted recovery of the simulated system over four passes (cuts) from a disturbed initial condition towards a flat desired horizon.

When r is set at 2.0 however, (i.e. $X_p = 2X$), condition (19) demands merely that

$$1 + 2k + k^2 > 0 \tag{22}$$

which is clearly satisfied for all k within the allowed range (17). Lengthening the structure subsections twofold has thus produced stability in this simplified system as confirmed by the traces of Fig. 8. {The general treatment mentioned above, in fact, reveals that stability is attainable for any even value of $r \geq 2$ for all k in the allowed range $0 < k < 1.0$ since k must merely satisfy the condition

$$(r-2) + kr + (r+2)\, k^r + r\, k^{r+1} > 0 \tag{23}$$

of which condition (22) is but a special case}

Setting r = 3 in conditions (20), however, shows that, for stability:

$$3\,k^2 < k+1 \tag{24}$$

which is satsified only if

$$0 < k < 0.768 \tag{25}$$

and indeed, whereas values of k within this range produce a stable, though less rapid recovery in the simulated responses (when compared to Fig. 8), setting k above 0.768 is found to produce the unstable behaviour typified by Fig. 9. This result again demonstrates that incrreasing X_p beyond the value X aids the stability of the multipass system. The somewhat inferior behaviour noted with r = 3, or indeed with any odd integer value, compared to cases in which r is even (and non-zero), may be attributed to the fact that odd-integer sampling picks up both the peaks and the troughs of the somewhat oscillatory response of G_s whereas even-integer sampling detects only one or the other. With r even therefore, no natural oscillation of G_s is transmitted from pass to pass, whereas a proportion of the oscillation energy is transmitted when r is odd.

The principle that increasing structure length aids multipass stability has been tested in fuller simulation of the true steering process and support structure as reported in Section 5. For the moment we turn to a separate analysis of the likely benefits of increasing structure width.

4. INCREASING STRUCTURE WIDTH

For this investigation, all dynamics in the along-face direction are ignored so reducing the problem to one in two dimensions only, viz: the face-advance direction and the vertical. All lags and delays in G_s are thus neglected and each discrete subsection of the machine support structure is assumed to be free to move quite independently of its neighbours. The two-dimensional system to be studied is illustrated in Fig. 10 and differs primarily from the real-life bench system of Figs. 3 and 4 in as much as the idealised system of Fig. 10 has a cutaway base (beam) allowing cut-floor steps to be bridged. Thus only the leading and trailing ends of the beam are here assumed to contact the cut-floor. [There appears to be divided opinion in practice as to whether flat or cutaway bases for mining machinery are superior and both types do exist[3,7]. By keeping the leading edge in better floor-contact, a cutaway base can avoid fine coal, left by the cutter, from penetrating beneath the base so causing an upward and sometimes cumulative bias on the machine's steering. On soft floors, however, such a narrow leading edge can severely tear up the floor leading to a loss of climbing ability. Floor strength is clearly a crucial parameter in these considerations].

Inspection of the geometry of Fig. 10 reveals that, for small angular changes the floor heights, y, tilt, α and cutting drum deflection, J, are interelated thus:

$$y(n+1) \simeq y(n) + W\,\alpha(n) + J(n) \tag{26}$$

and

$$\alpha(n) \simeq \{y(n) - y(n-M)\}/MW) \tag{27}$$

where W denotes the cutting drum width and MW the width of the structure base (i.e. the beam length). Integer n again denotes cut number, M is also assumed to be an integer but argument ℓ has been dropped since the study is here only two-dimensional as stated earlier .

We here assume a control law of the form

$$J(n) = k\,\{y_r - y(n)\} - k_g W\alpha(n) \tag{28}$$

Whereas in the preceding companion papers, and implicitly in Section 3 of the present paper, k_g was set at 1.0 to remove tilt effects from the process equations (26), we here, however, set this electronic derivative action to zero by putting k_g = 0.0. The object of this is to investigate the stabilising power of structure size i.e. the effect of increasing M, as an alternative to electronic compensation using a tilt transducer since the device is always vulnerable to damage in the arduous coal-face environment. Combining equations we therefore arrive at

$$y(n+1) = k\, y_r + y(n)\{1+M^{-1} - k\} - y(n-M)M^{-1} \tag{29}$$

and, on putting $y_r = 0$ for stability studies and taking z-transforms of this discrete system, gives

$$z\, \tilde{y}(z) = \tilde{y}(z)\{1+M^{-1} - k\} - z^{-M}\, M^{-1}\tilde{y}(z) \tag{30}$$

and cancellation of $\tilde{y}(z)$ {the z-transforms of y(n)} produces the characteristic equation

$$z^{M+1} + (k - 1 - M^{-1})z^{M} + M^{-1} = 0 \tag{31}$$

The effect of various values of M on systems stability is now readily determined by examination of the root loci derived from this characteristic equation. For this purpose, equation (31) is more conveniently expressed in terms of a ratio of polynomials of z thus

$$a\, z^{M}/(z^{M+1} + M^{-1}) = -1 \tag{32}$$

where

$$a = k - 1 - M^{-1} \tag{33}$$

The parameter 'a' may therefore be treated as the variable 'gain parameter' whose variation causes the roots of equation (32) vary in the z-plane. Critical values of 'a' may subsequently be interpreted in terms of true gain k for any preselected value of M using equation (33).

The general effect of increasing M, however, can be assessed merely by considering the position of the open-loop system poles and zeros (at which the loci start and finish). The M-zeros are clearly located at the origin whilst single <u>distinct</u> poles occur at positions given by

z .

$$z = \sqrt[M+1]{M^{-1}}\ \angle \pm \frac{i\pi}{M+1}\} ,\quad \begin{matrix} i=1,3,\ldots M\ (M\text{-odd}) \\ i=1,3.,,,M+1(M\text{-even}) \end{matrix} \tag{34}$$

The poles (i.e. the starting points of the loci) clearly lie on the unit circle when M = 1 (i.e. for conventional support structures such as a.f.c's) but the poles move progressively deeper inside the region of stability as M is increased. This suggests that increase of M does represent a powerful potential stabilising force under these circumstances. Loci computed from characteristic equation (32) using the Sheffield root locus package are illustrated for M = 1,2,3,4 and 5 in Figs. 11 (i) to 11(v) respectively. In the case of conventional systems, (M=1), the locus follows the boundary of the unit circle for $o < k < 4$ indicating its well known critical stability[8] in this range. The system is completely unstable for higher gains in the absence of tilt feedback. For the larger values of M the loci remain well inside the unit disc for substantial ranges of gain, critical stability not being reached until the values k_c set out in TABLE II are attained.

TABLE II Critical stability gains for various base widths

M	k_c
1	0.0 (to 4.0*)
2	2.0
3	2.67
4	2.0
5	2.40

A dramatic improvement in critical gain k_c therefore occurs immediately M is increased to values exceeding unity but, thereafter, little improvement in this parameter occurs. Due to the poles progressively nearing the origin as M increases however, the system damping, for a given gain in the range $o < k < k_c$) improves progressively and significantly as M is increased. The predictions are confirmed by the computed step responses illustrated in Fig. 12 which clearly show the improvement in damping. At a fixed value

*In the case of M=1 the system can never be better than critically stable for $k \geq o$ and therefore the value of 4.0 in Table II is of only academic interest.

of k however the speed of response (in terms of number of cuts taken to complete a response) deteriorates and some scope therefore exists for optimising M and k.

5. IMPROVEMENTS PREDICTED BY MORE-DETAILED MODELS

The models employed in the preceding Sections of the paper have been simplified versions to permit analytic solution. These solutions have indicated that stability improvements should be derived by increasing the length X_p (= rX) and/or the width MW of the loosely-articulated subsections that, together, comprise the support-structure for the power-loader. The question now arises as to whether or not these findings can be expected to carry over to the real-life situation. To generate confidence in possible future designs and to avoid an unnecessary number of prototypes (which in mining are enormously expensive) it is therefore necessary to test these preliminary conclusions on simulations close to reality.

Confining attention still to the uncoupled rod model of Section 3 the first step in this exercise was to relax the constraint that ratio $r = X_p/X$ should be an integer value as z-transform analysis dictates. Simulation in fact revealed that for r set at any value exceeding or equal to 2.0 would produce stability. We are thus reassured that the stability observed in Section 3 for r = 2,3,4 etc., is not merely a quirk of synchronism between the response of G_s and that of the structure. A more stringent test however, involves elimination of the floor penetration allowed in Section 3 thus demanding use of the potential-energy-minimising, dynamic programming (d.p.) simulation fully described in the first companion paper[1]. It was found that for stability, the subsection length needed to be increased to $X_p \geq 4X$ under these circumstances and Fig. 13 illustrates the stable behaviour achieved with r = 4 and k = 0.5. Convergence to the flat horizon is obviously somewhat slower than with the simplistic model (Fig. 8) so that floor penetration, as well as section length, is clearly a stabilising factor. Fig. 13, however, represents the first stable performance ever achieved with the d.p. model since all earlier studies had been confined to smaller values of r and the stable solution would not have been pursued nearly so vigorously without the incentive provided by first obtaining the stable analytic solutions of Section 3.

Inclusion of sensor and actuator lag distances X_1 and X_2 in the dynamics of G_s increases the critical value of X_p still further and the following rule of thumb emerges (from these later d.p. studies) for multipass stability, viz :

$$X_p \geq 4(X + X_1 + X_2) \quad , \quad (X_1, X_2 < X) \tag{35}$$

Probably for this reason, d.p. simulation of a chain of structures resembling more closely the benches of Fig. 4, failed to produce multipass stability since these structures, although three terms longer than a.f.c. trays, each have a length, X_p, of only about $2(X + X_1 + X_2)$: the minimum practical values of X, X_1 and X_2 being of the order of 75, 60 and 20% of the conventional tray length ($X_p/3$ in this case).

The idealised structure used in the latter d.p. investigation is shown in Fig. 14. It clearly resembles the bench structure more closely than the twin, uncoupled rod model used earlier, but the beams are assumed to have cutaway bases as in the two-dimensional studies of Section 4. Because floor penetration (or bridging) appears to aid stability, it was thought important to conduct two-dimensional simulations using flat based beams. Indeed, using colour graphics, such a flat-based two-dimensional model, embodying extensive logic[9] for the fitting of both the bench and its associated roof-support, to the cut floor has been developed into an interactive process-trainer for future operators of the system. Fig. 15 shows a typical response of the simulation in automatic mode (as opposed to the primary manual control mode for operators) using control law (28) with $M = 5$, $k = 1$, and $k_g = 0$ (i.e. again no tilt feedback). Clearly a stable response is still obtained with $M > 1$ (as also predicted by the simpler cutaway base model of Section 4) but rather greater overshoots tend to result with flat bases on hard floors because of their greater tendency to topple as the centre-of-gravity passes over an intermediate supporting step.

6. CONCLUSIONS

It has been shown that increasing length, X_p, of the power-loader support-structures with given steering system dynamics, G_s, involving a delay distance, X, can stabilise the vertical steering of the multipass longwall coal-cutting process. Using z-transform methods, simple analytic models predict that $X_p \geq 2X$ should give multipass stability. More realistic models based on dynamic programming methods, however, show that a practical rule of thumb stability criterion is $X_p \geq 4(X + X_1 + X_2)$, X_1 and X_2 being the sensor and actuator lag distances respectively. It therefore, appears that loosely articulated piecewise-rigid structures for the track of the power-loader can eliminate repeated excitation of the resonance of G_s but a very considerable increase in the subsection length, X_p, on conventional designs is needed. This probably means that the bench system (Fig. 4) will require elastication of the joints between benches or the use of pretensioned cables, as proposed in the preceding companion paper[2] to ensure stability despite the three-fold increase in X_p that the present bench system represents. Much depends on the strength of the cut-floor however. Analysis and detailed simulation have shown that increasing structure width eliminates the need for electronic tilt feedback in steering systems so aiding their robustness but slowing their response. A more gradual response for wide-based roof support structures (Fig. 15) is probably what is needed however, to ensure good roof contact and therefore good roof-control. The stability of most systems can be achieved either by means of <u>control</u> of adequate intelligence (complexity) or by means of adequate system inertia. The paper has demonstrated this general principle in this particular application. The paper has also demonstrated the need for analytic solutions of simplified systems to guide the choice of parameters in detailed process simulations for results of real value to be obtained. This, in the authors' view, is another important principle sometimes neglected.

7. ACKNOWLEDGEMENTS

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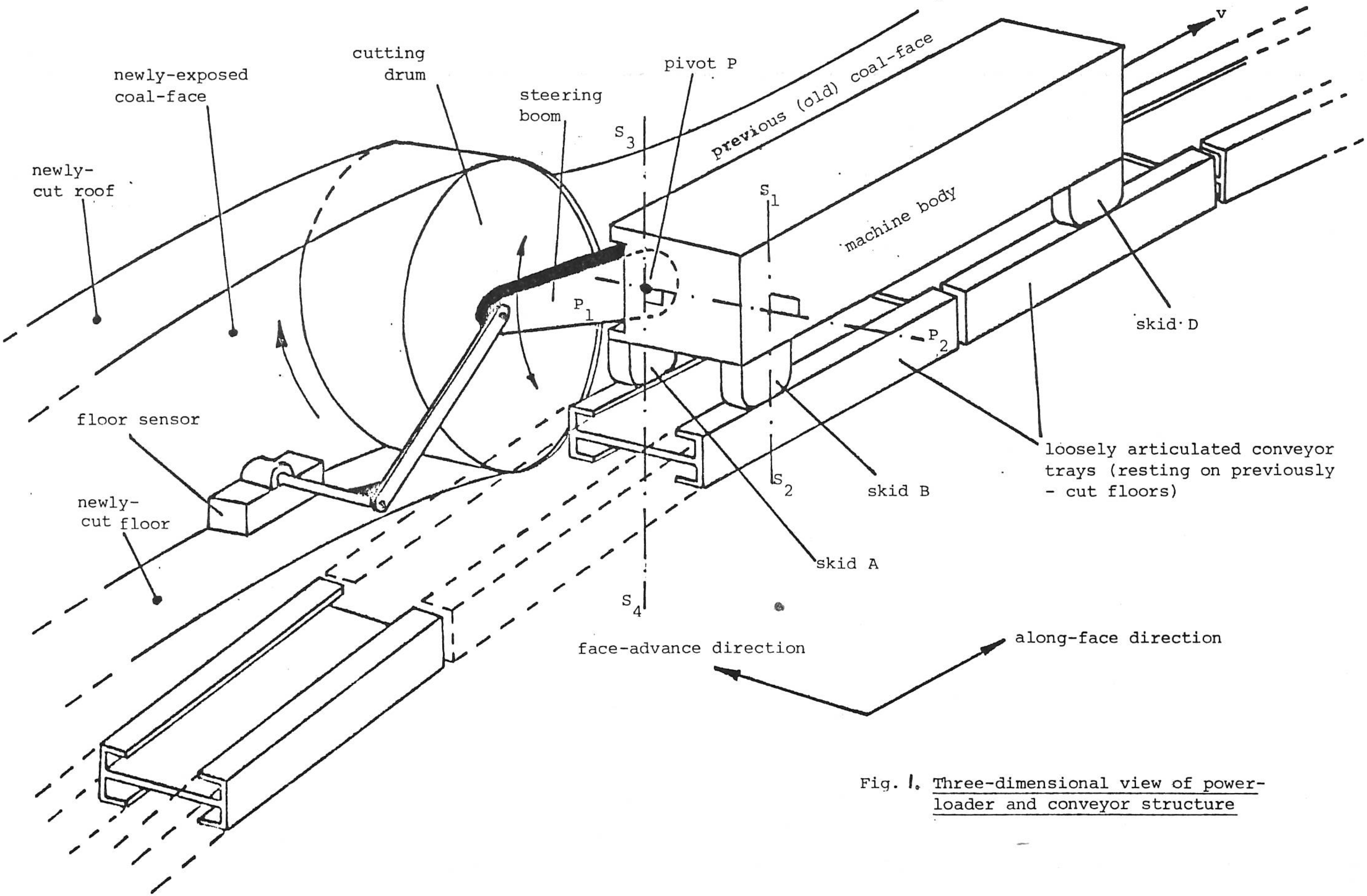


Fig. 1. Three-dimensional view of power-loader and conveyor structure

Fig. 2 Conventional longwall system in end view

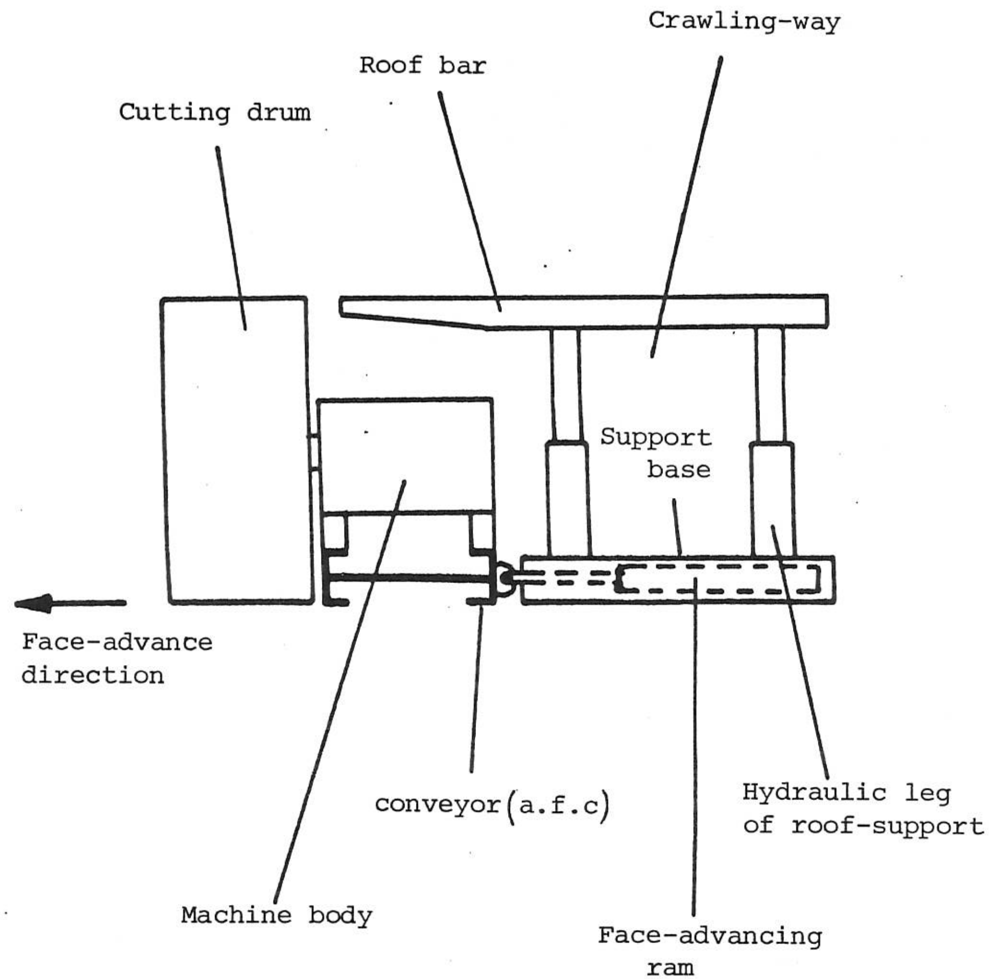


Fig. 3 Bench System for Thick Seams (Drawn to same scale as Fig. 2)

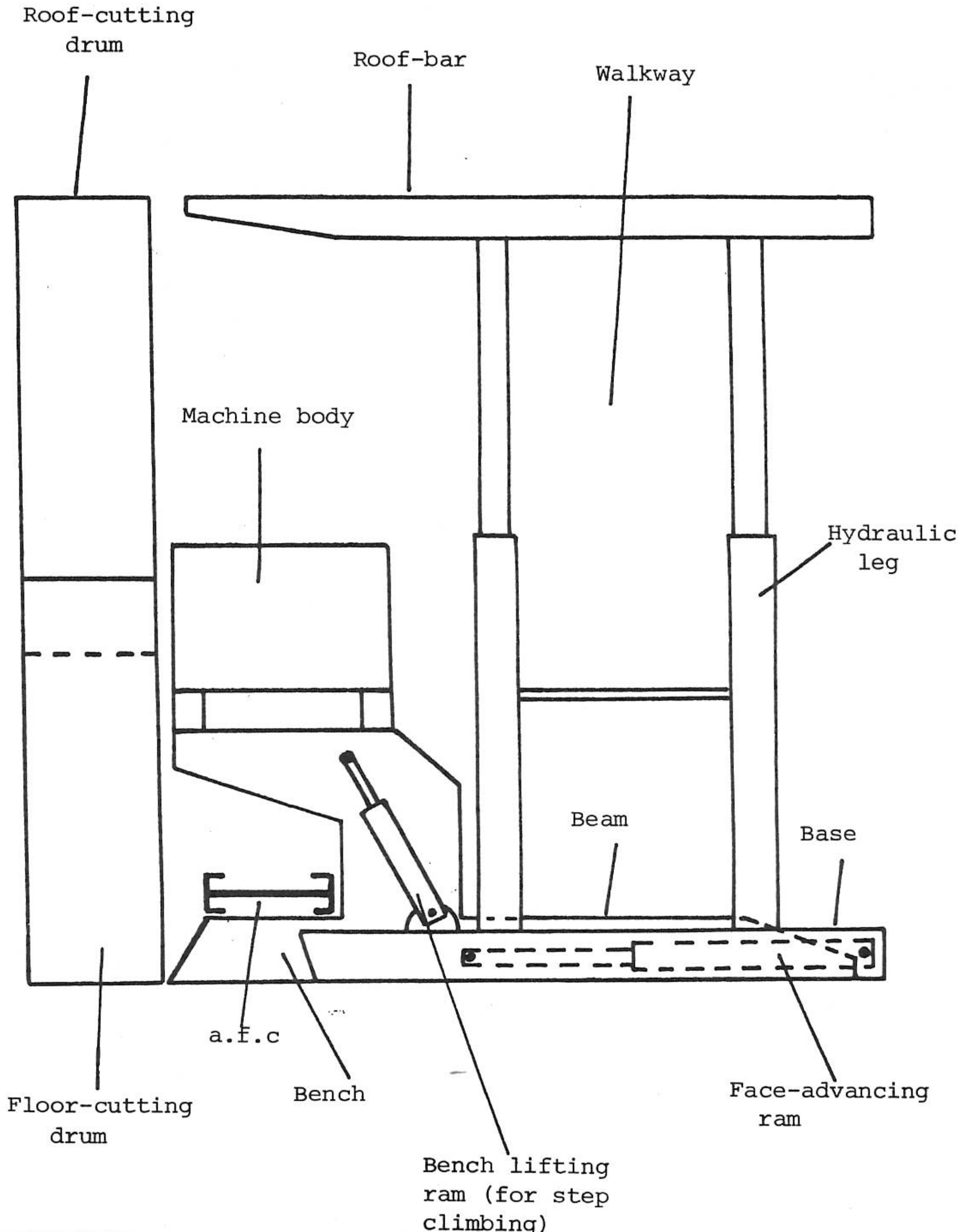
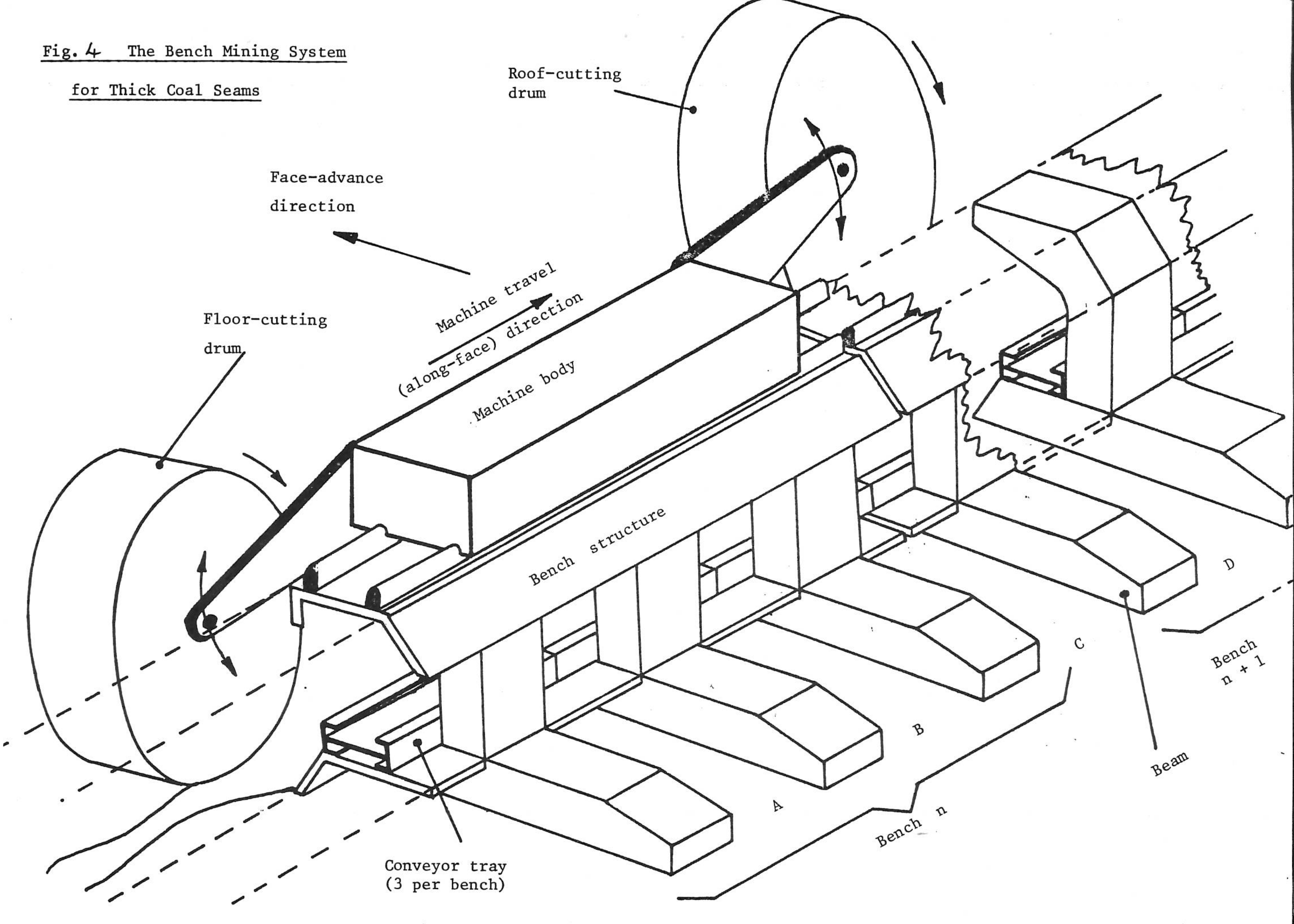


Fig. 4 The Bench Mining System for Thick Coal Seams



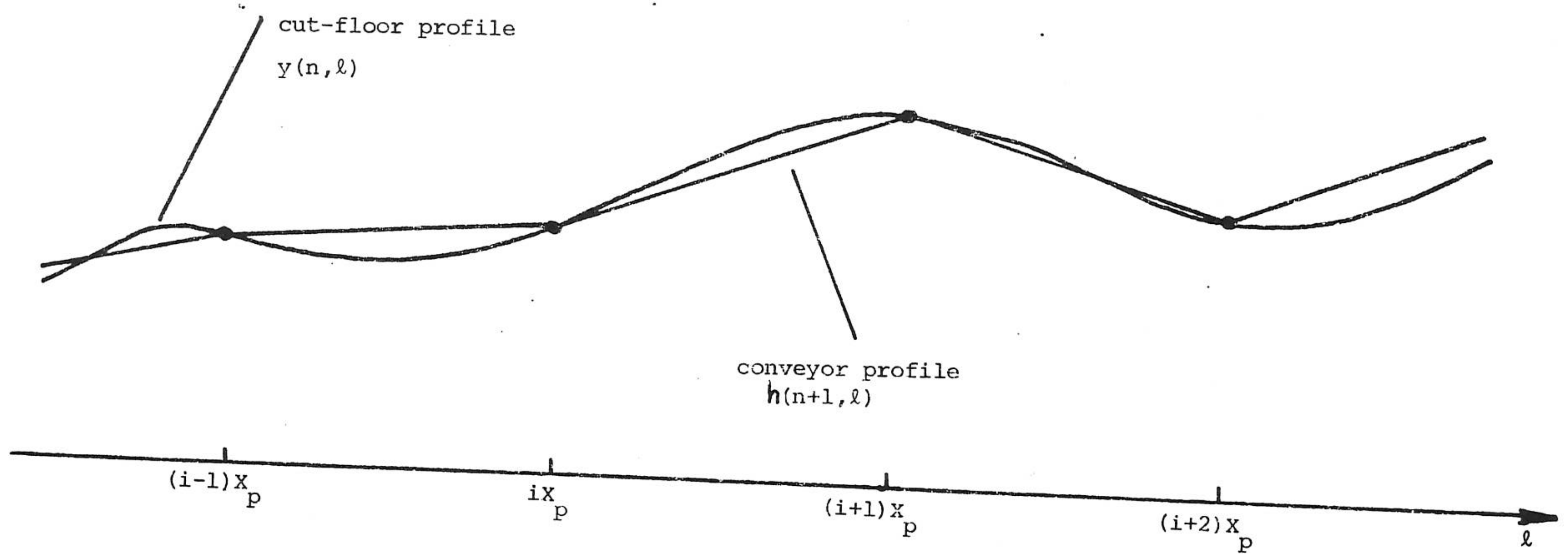


Fig.5 Showing assumed behaviour of piecewise rigid conveyor

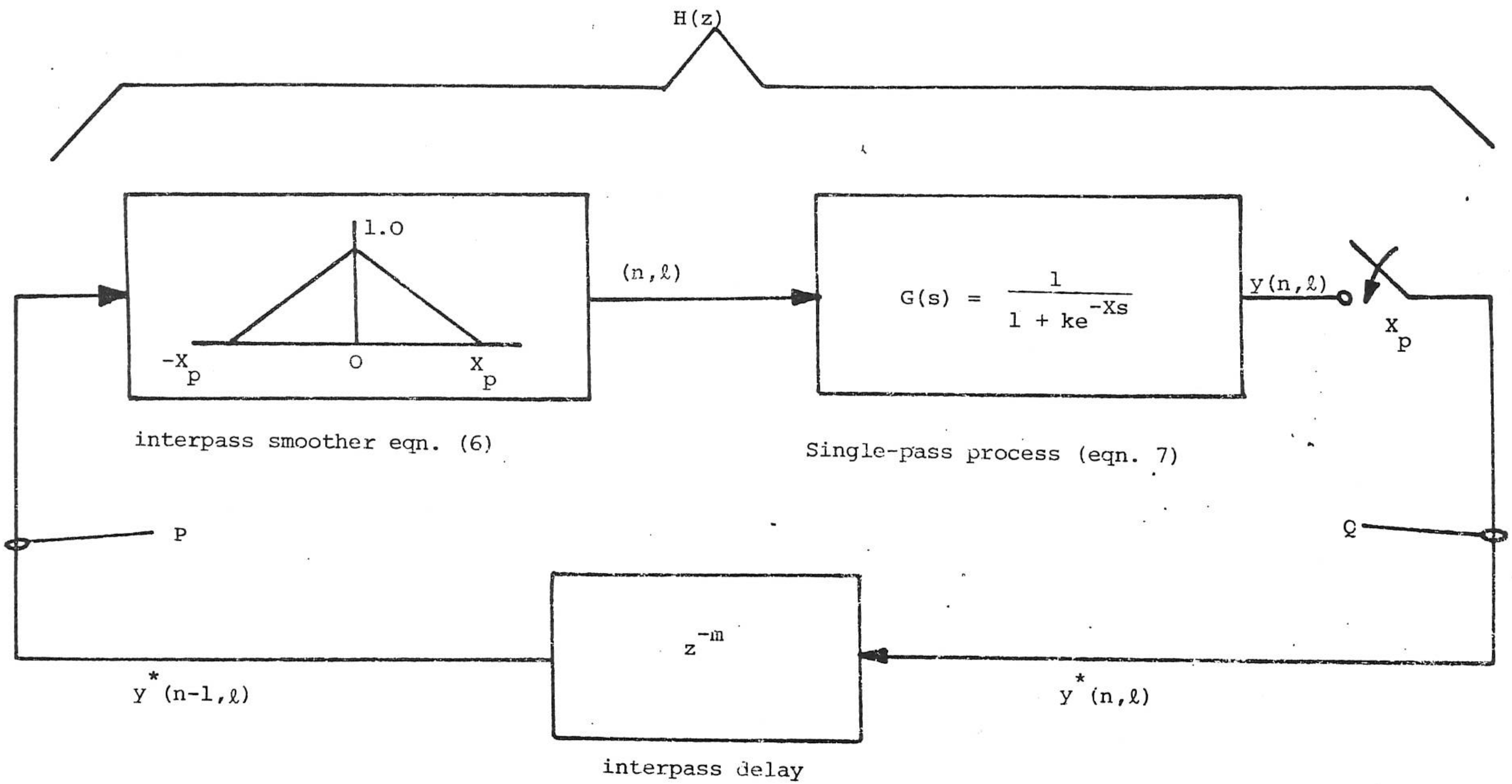


Fig. 6. Complete system block-diagram for calculation of H(z)

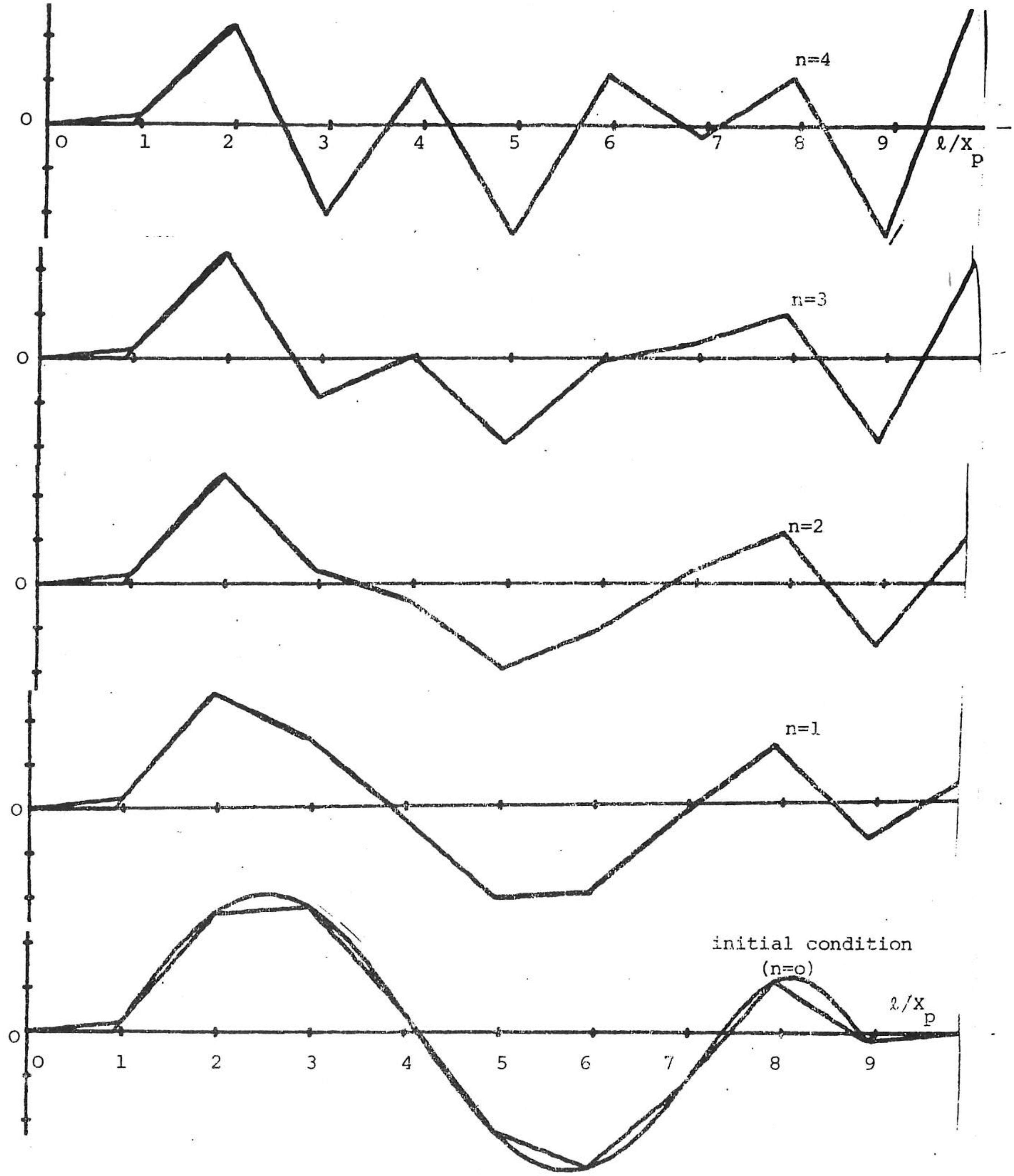
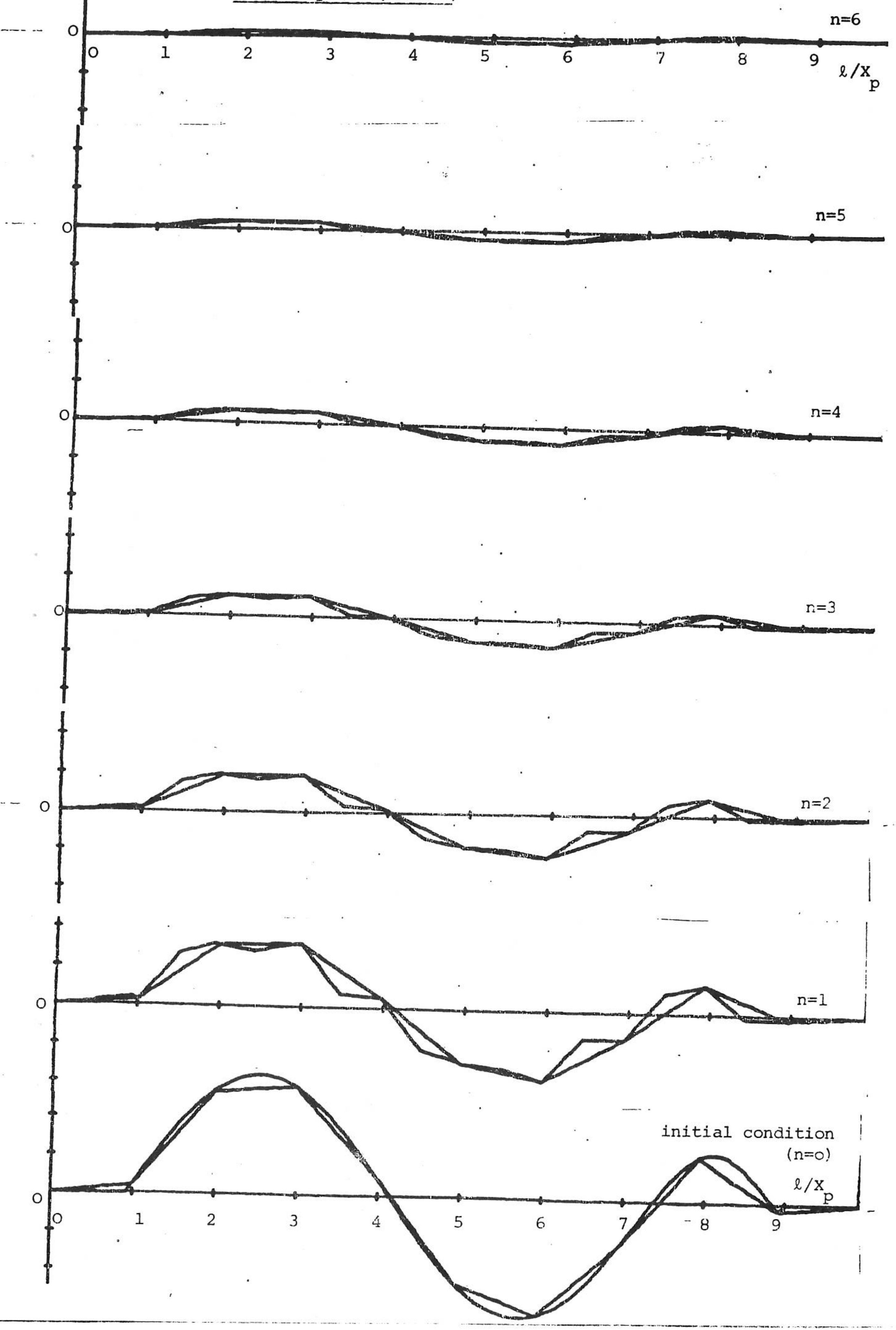


Fig.7 Traces of $y(n,\ell)$ and $h(n+1,\ell)$ showing multipass instability

($r=1$, $k= 0.5$)

Fig. 8 Traces of $y(n,\ell)$ and $h(n+1,\ell)$ showing multipass stability ($r=2$, $k=0.8$)



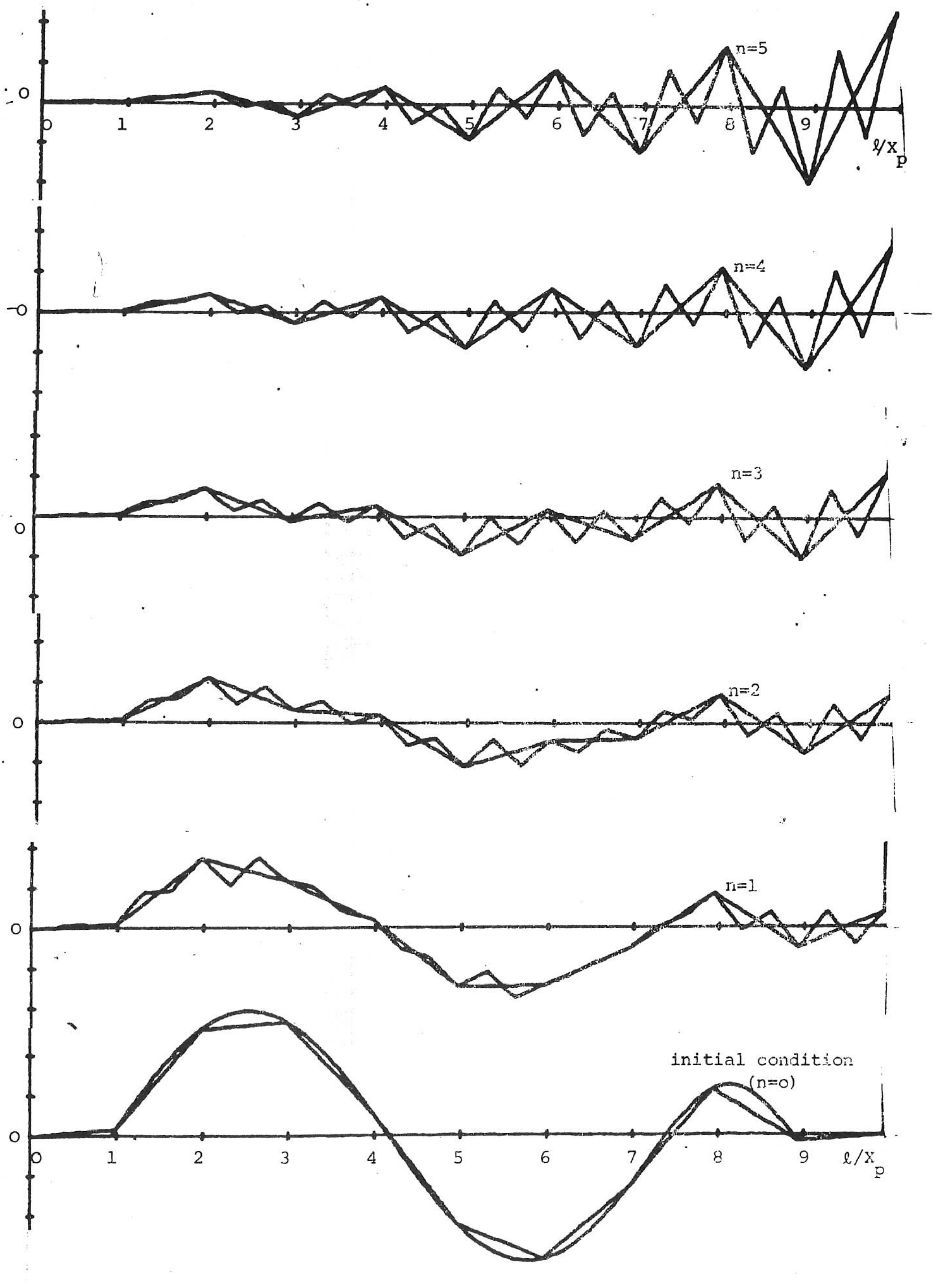


Fig. 9 Traces of $y(n,\ell)$ and $h(n+1,\ell)$ showing multipass instability ($r=3$, $k = 0.95$)

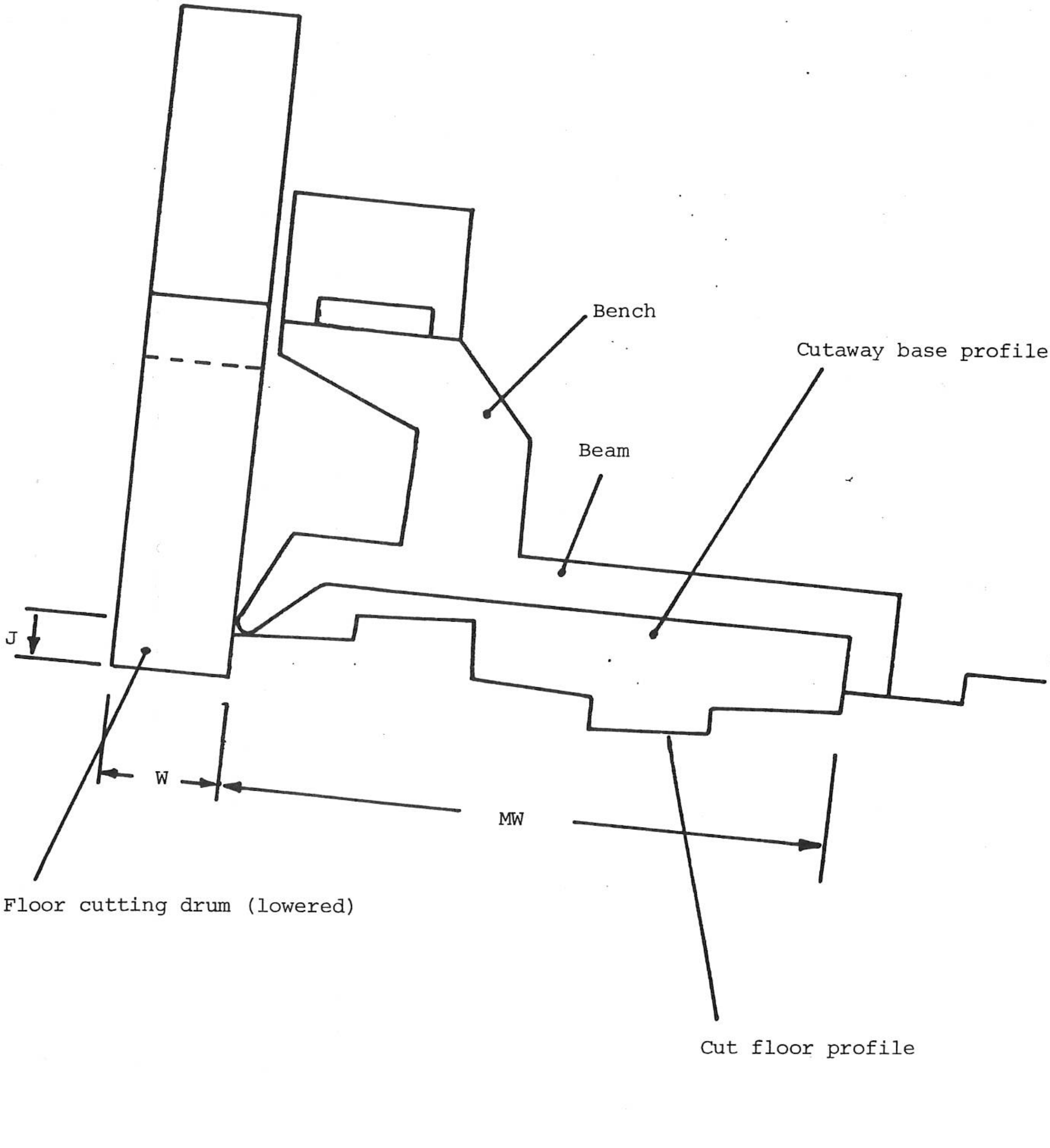
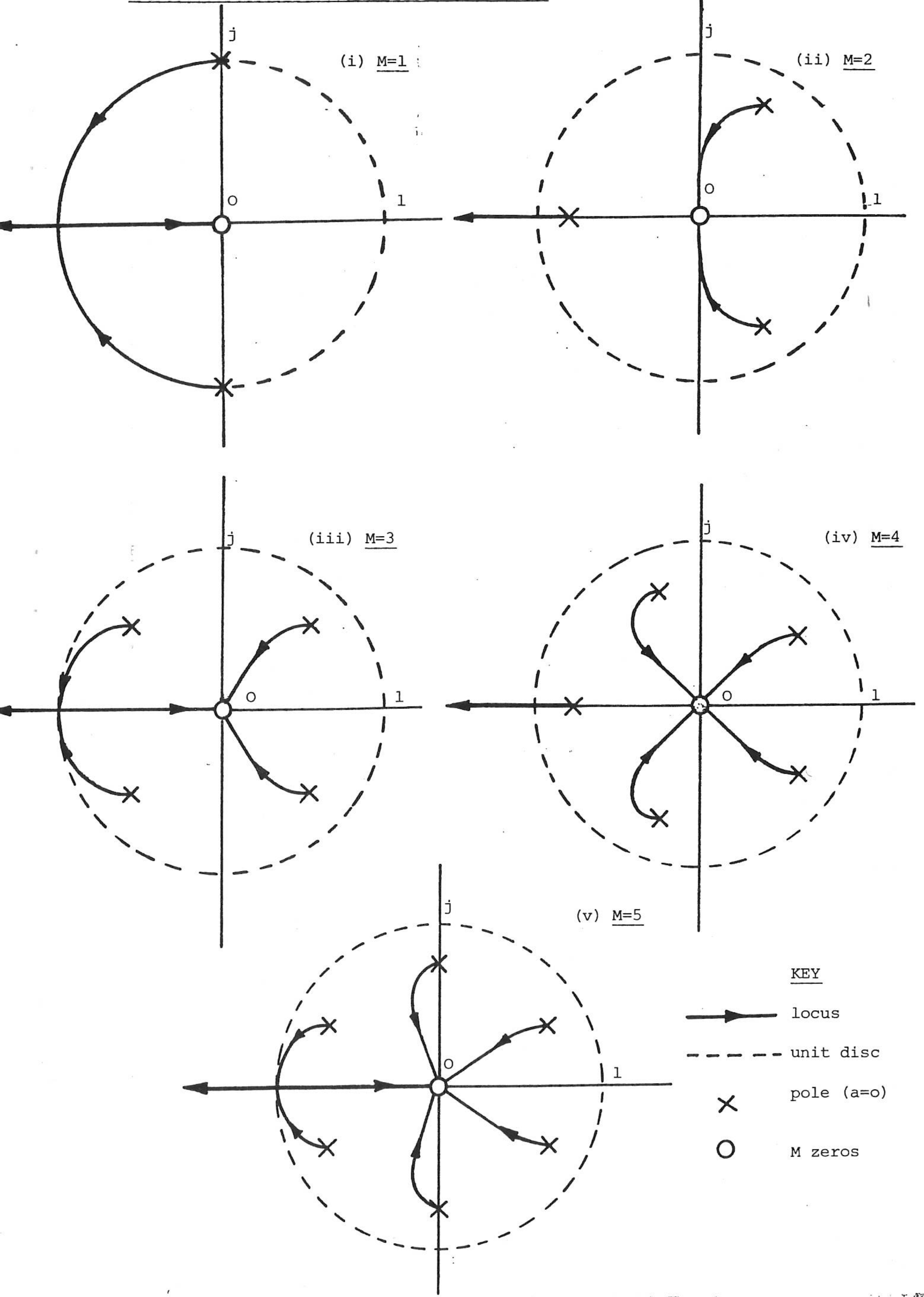


Fig. 10 Showing cutaway base to allow bench to bridge floor steps

Fig. 11 Root loci for various values of M



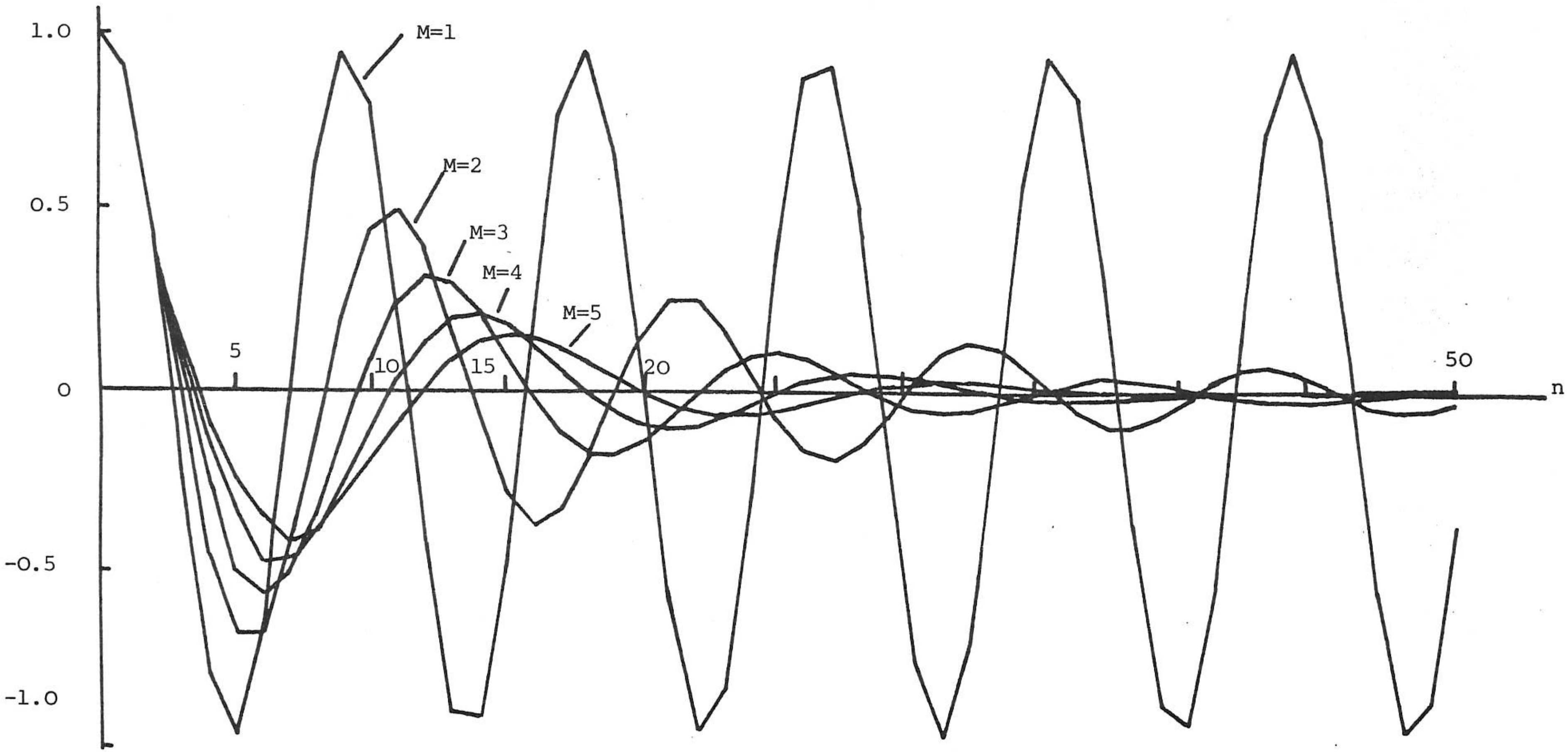


Fig. 12 Transient responses of simplified two-dimensional face-advance model for various values of M ($k = 0.5$, $k_g = 0.0$, $y_r = 0$).

Fig. 13 Traces of $y(n,\ell)$ and $h(n+1,\ell)$ showing stability of rigorous dynamic programming model ($r=4$, $k=0.5$)

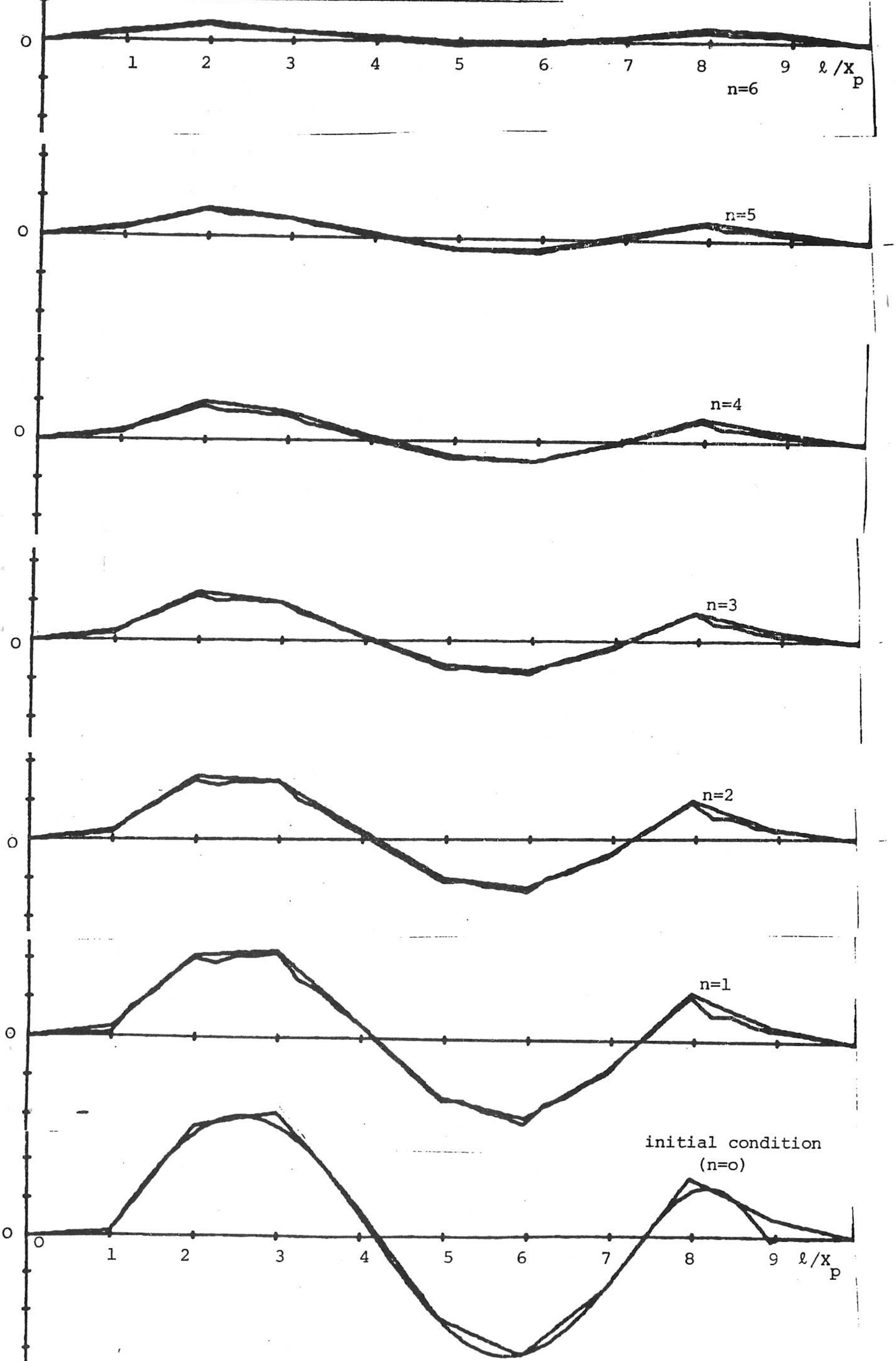
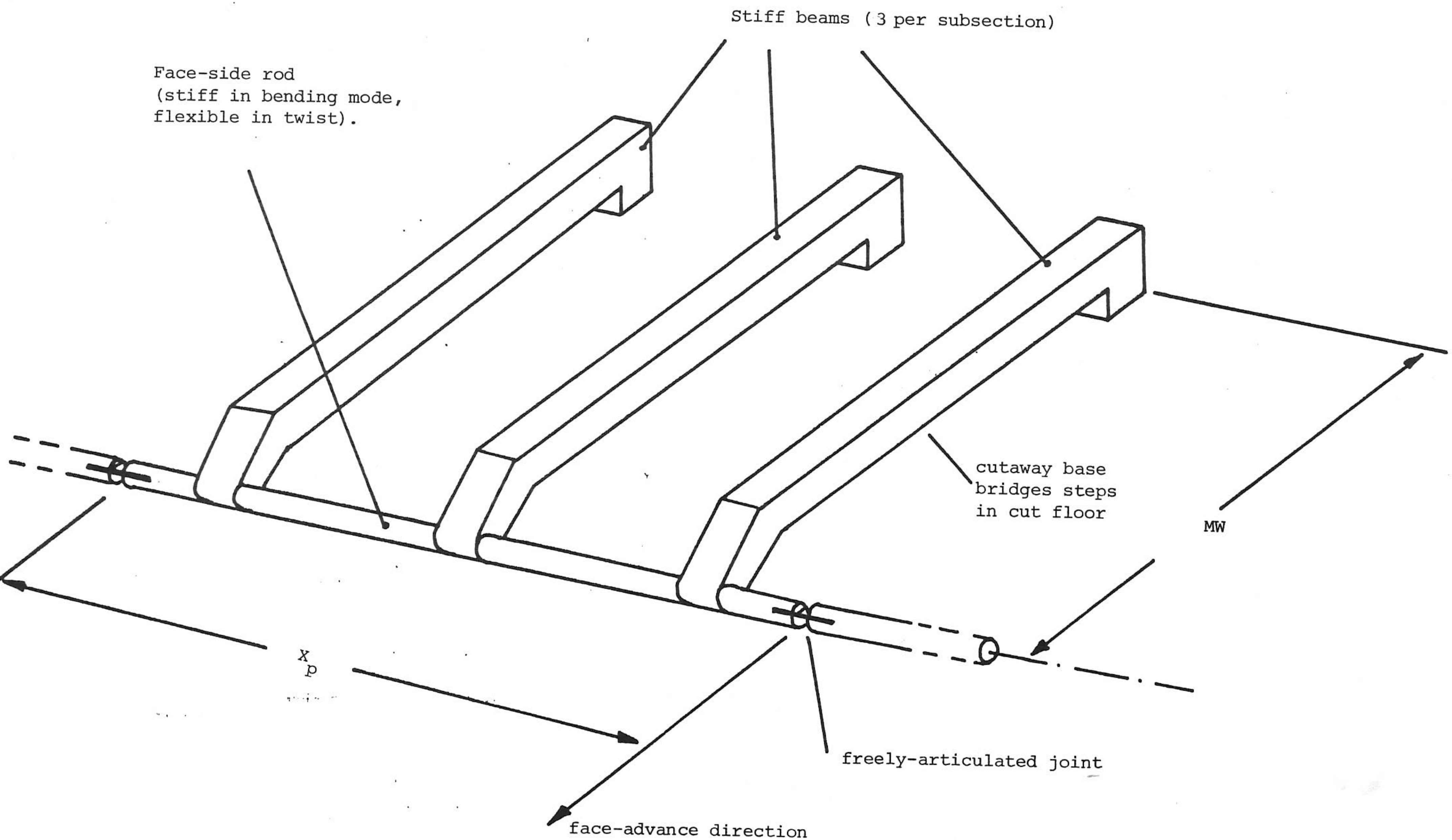


Fig. 14 Idealised structure assumed for dynamic programming model of bench system



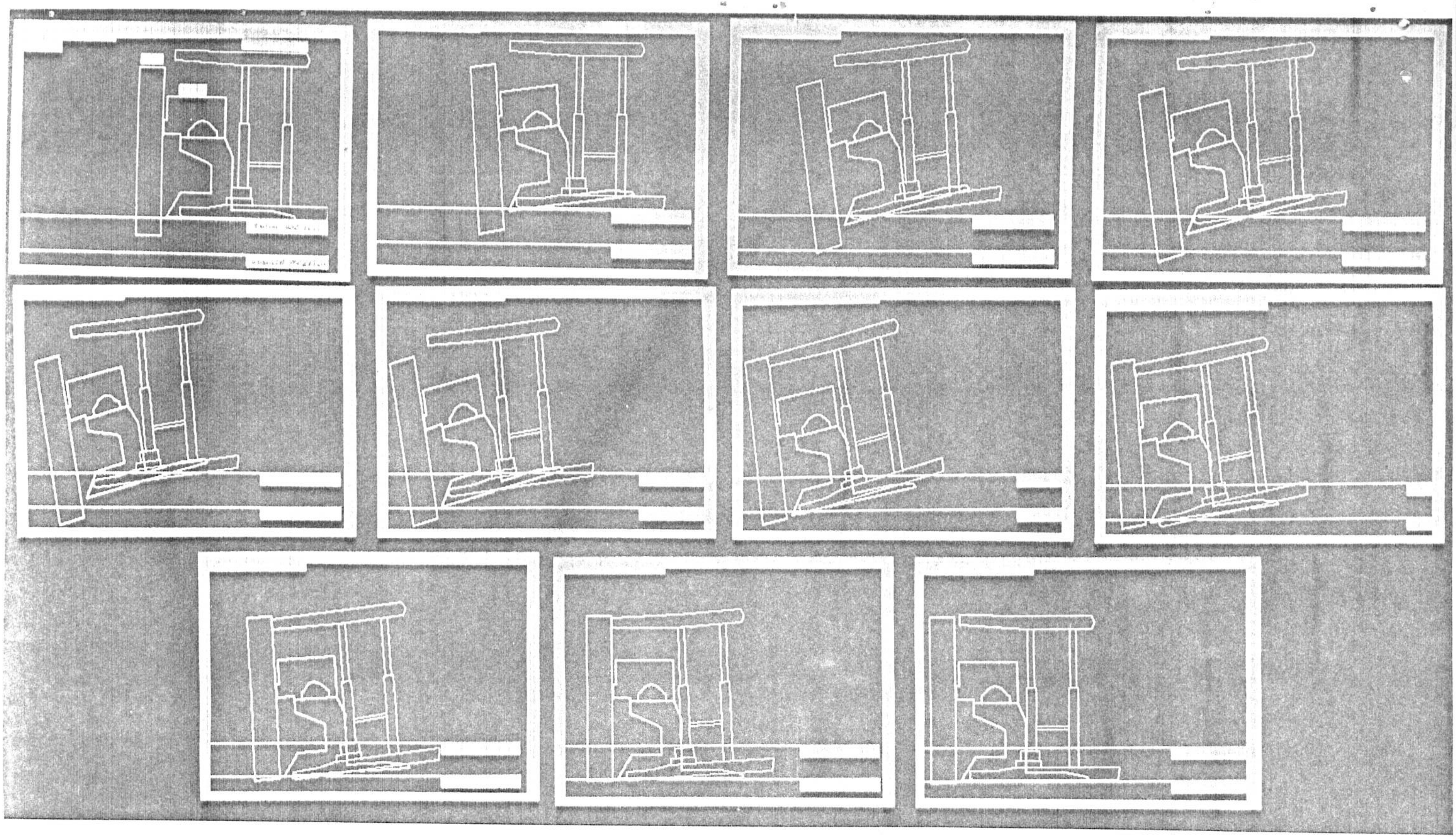


Fig. 15 Step-Response from 2-Dimensional Bench System Simulator with Flat Base ($k=1$, $k_g = 0.0$, $M=5$)