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Modelling the Steering Characteristics of Coal Face Conveyors by Random Signal Testing

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ABSTRACT

A quarter-scale model of a coal cutting machine and armoured flexible conveyor (a.f.c.) system is subjected to random signal testing in an attempt to fit a continuous elastic beam model to the piecewise rigid a.f.c. structure for vertical steering investigations. A roller track, over which the system is hauled at constant speed, is used to provide the random test signal and a systems identification package, SPAID, developed at Sheffield, used to generate the source signal noise and to compute the cross correlation functions relating input and output height profiles. An energy-minimising dynamic-programming model of the a.f.c. is used to guide the choice of experimental parameters.

The results yield an equivalent elastic beam stiffness for the loose-coupled a.f.c. structure that is considerably less than that originally anticipated, thus reconciling the hitherto opposing predictions for steering system stability produced by discrete and continuous models. A.f.c. segments are shown to require considerable lengthening to stabilise simple analogue steering controllers.

Modelling the Steering Characteristics of Coal Face Conveyors by Random Signal Testing

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1. INTRODUCTION

1.1. The Longwall Method of Mining

Fig.1 shows a power-loader of the ranging-drum shearer type, conventional in Europe and gaining popularity elsewhere, cutting along a short section of a coal-face. Such faces are typically 100 to 300m in length and seam thicknesses suitable for this method of extraction lie in the range one to three metres. Between successive passes of the shearer, the armoured-flexible-conveyor (a.f.c.) upon which the machine rides is snaked forward by hydraulic rams as indicated in Fig.2. The sequence of operations for a unidirectional system is illustrated in Fig.3.

In the thicker seams, double-drum machines may be necessary, one drum for cutting the roof-coal whilst the other cuts the floor. In medium seams, a single drum suffices, provided its diameter is chosen to equal the nominal seam thickness minus, perhaps, the thickness (2 to 10 c.m.) of a thin coal ceiling to be left up for roof-control purposes in situations where the strata overlying the seam is fragile.

1.2. Vertical Steering

Although seams undulate vertically, their thicknesses generally remain nearly constant over many 100's of metres so that the purpose of the ranging arm (boom) is to steer the single cutting drum vertically, with respect to the a.f.c., in an attempt to follow seam undulations and so maintain the preset coal-ceiling thickness. Sensors of the nucleonic (1,2) and natural radiation (3) type have been developed whilst pick-force types are under intensive development for the measurement of the drum position within the seam and thus the thickness of the coal ceiling or floor left behind. Figs. 1 and 4 illustrate a nucleonic floor sensor attached to the shearer. The sensors can clearly provide the basic feedback control signal for positioning the hydraulically-actuated drum arm.

1.3. Multipass Instability

The control loop naturally involves a significant transport delay, X, plus sensor and actuator lags X_1 and X_2 but control loops can readily be tuned for stable operation on any <u>single</u> pass of the machine along the face. Unfortunately these systems are unavoidably <u>resonant</u> due to sensor delay X and drum offset R (= horizontal distance between drum centre and rear skids of machine) and because the a.f.c. settles on each newly-cut floor after each pushover process, cut-floor oscillations transmitted by the a.f.c. structure redisturb the control system at its resonant frequency leading to the growth of oscillations over a sequence of passes (cuts) and hence to multipass instability. Fig.6 shows the step responses computed for the so-called <u>rubber conveyor assumption</u> under which the a.f.c. is assumed to reproduce perfectly the cut-floor profile beneath. The simulation is based on the following machine equation derived from the geometry of Figs. 4 and 5:

$$y(n,\ell)+z(n,\ell) = h(n,\ell+R)+W\alpha(n,\ell+R)+R\beta(n,\ell+R)+J(n,\ell) \qquad \ldots(1)$$

in which y, h and z denote floor, a.f.c. and seam heights respectively, J the drum deflection and tilts α and β are given by

$$\alpha(n,\ell) = \{h(n,\ell) - h(n-1,\ell)\}/W \qquad ...(2)$$

and $$\beta(n,\ell) = \{h(n,\ell) - h(n,\ell+F)\}/F \qquad ...(3)$$

W being the a.f.c. width (= drum width), F the machine skid spacing whilst n and ℓ denote cut number and drum distance. The 2-term control law was used, i.e.

$$J_d(n,\ell) = k_h\{y_r - y_m(n,\ell)\} - k_g W\alpha(n,\ell+R) \qquad ...(4)$$

where k_h and k_g are the controller height- and tilt-gains (set at 0.8 and 1.0 respectively), y_r = reference floor-coal thickness, and y_m is the measured coal-thickness given by:

$$y_m(n,\ell) = y(n,\ell-X)/(1+X_1D) \qquad ...(5)$$

whilst jack-demand J_d is related to J thus:

$$J(n,\ell) = J_d(n,\ell)/(1+X_2D) \qquad ...(6)$$

The parameters used were X = 1.25m, X_1 = 0.6m and X_2 = 0.165m but the obvious multipass instability is incurable (4) under the rubber conveyor assumption:

$$h(n,\ell) = y(n-1,\ell) + z(n-1,\ell) \qquad ...(7)$$

Indeed Fig.6 was computed for the more stable "fixed-drum" shearer for which R is effectively zero. The presence of drum offset R in ranging drum machines only worstens the problem.

2. A.F.C. SMOOTHING

Of course the a.f.c. structure, being composed of some 80 to 200 rigid steel trays, loose-jointed end-to-end, will not follow precisely the cut-floor profile beneath. It will, in fact, act as a sort of floor-filter and it is this filtering process that is the subject of the present paper.

If the steering system dynamics may be represented thus (setting disturbances y_r and z to zero)

$$\tilde{y}(n,s) = G_s(s)\tilde{h}(n,s) \qquad \ldots(8)$$

and the a.f.c. may also be modelled linearly, to a first approximation, thus:

$$\tilde{h}(n,s) = G_c(s)\tilde{y}(n-1,s) \qquad \ldots(9)$$

then, as has been shown elsewhere (4), for multipass stability:

$$|G_s(j\omega)||G_c(j\omega)| < 1.0 \quad , \quad \text{for all } \omega \qquad \ldots(10)$$

For the special case: $R = X_2 = 0$, $k_g = 1.0$, it is readily deduced from eqns. (1) to (6) that

$$G_s(j\omega) = \frac{1 + X_1 j\omega}{1 + X_1 j\omega + k_h \exp(-Xj\omega)} \qquad \ldots(11)$$

whilst, for the rubber-conveyor assumption, $G_c(j\omega) = 1.0$ for which it is readily proved analytically that stability criterion (10) can never be satisfied for any X, X_1 or k_h thus confirming the previous transient responses (Fig.6). Making R_2 and $X_2 > 0$ only worstens the situation.

2.1. An Elastic Beam Model

If it is assumed that the a.f.c. structure's behaviour will resemble that of a continuous elastic beam extending along the full face-length, L, and resting on a bed of fine coal (left behind by the cutting-drum or created by damage to the solid floor steps during a.f.c. pushover) then it can be shown that

$$G_c(s) = 1/(1 + X_c^4 s^4) \qquad \ldots(12)$$

producing the <u>two-sided</u> impulse response shown in Fig.7. Distance constant X_c is given by

$$X_c^4 = EI/k_f \qquad \ldots(13)$$

where E = Young's Modulus of the beam material, I the beam's second moment of area about its neutral axis and k_f = reaction force p.u. length of beam p.u. compression of the bed of fine coal. From Fig.7, $2X_c$ clearly also has an important physical significance being, very nearly, the distance over which the beam falls to steady-state in response to an impulse in $y(n,\ell)$.

Provided X_c is made sufficiently large compared to the lag distances X, X_1, X_2 and R, then it can be shown analytically that multipass stability criterion (10) can now be satisfied as the specimen responses of Fig.8 to an initial step disturbance in $y(0,\ell)$ confirm. Here X_c/X is set at 0.5 and $X_1 = X_2 = R = 0$, (k_h being set at 0.5 and $k_g = 1.0$ as before). The multipass stability criterion for this case, (independent of the k_h setting provided this is <1.0) is readily shown to be

$$X_c/X \geq 0.4 \qquad \ldots(14)$$

and, for more general application, the rule of thumb

$$X_c/X_T \geq 0.4 \qquad \ldots(15)$$

applies, where $X_T = X+X_1+X_2+R$ if X dominates.

2.2. Discrete, Rigid-Section Models

<u>2.2.1. Mechanical models.</u> Until recently no sound mathematical model of the rigid-tray a.f.c. structure existed and investigations of its behaviour were undertaken using a 20-30 tray quarter-scale model (7) of the a.f.c. and cutting machine, using expanded polystyrene to

simulate coal. The elastic beam model of Section 2.1 was developed on an attempt to eliminate the need for this mechanical simulator the operation of which is tedious, costly and does not readily allow exploration of system parameter variation. Unfortunately the stability possibility predicted by the elastic beam concept was not reproduced by mechanical model trials using conventional a.f.c. tray-lengths X_p. Instead, responses resembling rubber conveyor predictions (Fig.6) were produced.

2.2.2. Computer Models. To ease the simulation of rigid trays, the authors were commissioned by the UK National Coal Board's Mining Research and Development Establishment (M.R.D.E.) to develop a digital program to describe a.f.c. behaviour. The model (5) was based on General Dynamic Programming (D.P.) and computes the heights and tilts of each tray to minimise the total potential-energy of the entire sequence without contravening angular constraints at the joints and without allowing penetration of the cut-floor profile $y(n,\ell)+z(n,\ell)$ which is assumed to remain rigid and dust-free. Like the scale-model, using conventional tray-lengths, X_p and delays X, the D.P. model predicts multipass instability (5) no matter what values of X_1, X_2, R, k_h and k_g are employed.

A serious discrepancy between the elastic beam and multiple rigid-tray concepts was therefore apparent. Clearly, in the latter case, there is no tendency for the a.f.c. to "spring-straight" under the restoring force of inherent elasticity. To test this diagnosis, artificial spring joints were added to the D.P. model, now programmed to minimise total strain + potential energy, and sufficient joint resilience was shown to stabilise. (The conclusion confirms practical trials conducted underground in which the loose a.f.c. joints have been bridged by cable-trays and ramp-plates to stiffen the structure as a whole).

2.2.3. Analytical Models. There nevertheless remains the natural expectation that tray rigidity, by itself, should be a stabilising influence since it should blanket out cut-floor oscillations to some extent and so prevent re-excitation of the resonance of $G_s(s)$. An analytical investigation of the effect of tray rigidity was therefore required.

Unfortunately, there is no simple analytical means for representing the real-life system because the points of contact between cut floor $y(n,\ell)+z(n,\ell)$ and conveyor profile $h(n+1,\ell)$ are randomly spaced in general, depending on the precise shape of the cut-floor. The difficulty is avoided however if the trays are superimposed on short pedastels at each end so that floor contact occurs only at the ends and intermediate undulations are bridged. The a.f.c. structure now becomes a sampler and linear interpolator so permitting z-transform analysis.

The details of the analysis are reported in references (8) and (9) and the main prediction obtained is that, for $X_1 = X_2 = R = 0$, $k_g = 1.0$ then stability is obtainable provided tray length X_p is chosen such that

$$X_p \geq 2X \qquad \ldots(16)$$

This minimum tray length is somewhat larger than occurs in conventional practice where $X \leq X_p \leq 1.3X$ and of course in real systems the other system lags X_1, X_2 and R would demand an even larger value of X_p for stability. Result (16) prompted D.P. studies with larger values of X_p and indeed stability was eventually obtained, provided the rule of thumb

$$X_p \geq 4X_T \qquad \ldots(17)$$

was satisfied. (X_T again = total system delay). Such tray lengths are, of course, enormously larger than in conventional systems.

2.2.4. The need for Identification Models. Now although some reconciliation of elastic-beam and rigid-tray modelling has been accomplished in Sections 2.2.2 and 2.2.3 and, despite negative results from initial trials, tray-rigidity has been shown to stabilise, there remains an apparent discrepancy of considerable proportions between the predictions of the two approaches. Specifically the elastic-beam criterion:

$$X_c \geq 0.4X_T \qquad \ldots(18)$$

appears at first sight to be far more optimistic than condition (17) resulting from rigid tray studies if X_c and X_p are of comparable magnitude. An examination of the impulse response of Fig.7 would suggest that decrement distance $2X_c$ should correspond to one or two pan lengths X_p of the real structure depending on the amplitude of the disturbances and consequently the number of joints over which the free-play between joints is taken up.

The discrete models, like the real conveyor, are of course all nonlinear and the fitting responses originating from one type and size of disturbance may not therefore produce an equivalent elastic beam model suitable for stability prediction. A fit obtained by some form of averaging process accommodating the full spectrum of likely input signals is likely to be much more powerful as a performance predictor and, for this reason, a programme of random signal testing of the $\frac{1}{4}$-scale model has been undertaken in an attempt to produce an improved "equivalent elastic beam" representation.

3. RANDOM SIGNAL TESTS

Random-signal testing and the production system impulse responses from the cross-correlation of the random input and output data is well-known nowadays and its theory is therefore not reproduced here. Instead we concentrate on the practical difficulties that arose in this application and the results obtained.

3.1. Practical Details

To make these tests, the $\frac{1}{4}$-scale model machine and a.f.c. were supported on a track of random diameter rollers as illustrated in Fig.9. During testing, the entire system (machine and a.f.c.) was hauled over the roller track at constant speed. The roller diameters were determined initially from a pseudo-random-binary-sequence (P.R.B.S.) of mean frequency b^{-1} filtered by a first-order lag of distance-constant, X_f, where ideally

$$b \ll X_f \ll \hat{X}_c \qquad \ldots(19)$$

constraint (19) being necessary to ensure a reasonably smooth analogue noise input whose bandwidth would well exceed that expected for the system (a.f.c.) under test. Too low a value of X_f renders the test signal excessively rich in high frequencies to the detriment of the lower frequencies present needed to excite the system dynamics, so causing considerable noise in the final correlation results.

Pure unfiltered P.R.B.S. is, of course, totally unsuitable as a test-signal in this application because the trays merely adopt a constant level across the tops of the binary bits. A related observation is, of course, that, even for analogue inputs of zero mean, the output of the a.f.c. has a positive mean due to its tendency to rectify higher-frequency components: no floor penetration being allowed. The means of the output signals were therefore computed and removed before correlation.

A roller track length of some $20X_p$ was chosen on the grounds that this should be long compared to the anticipated value $\hat{X}_c$ ($\simeq X_p$) of X_c. Trays were added to either side of those supporting the machine until further additions produced negligible effect on the output signal recorded from a vertical height potentiometer attached to the model machine and following a horizontal datum surface above (Fig.9).

Roller diameters ranging from 2 to 8cm were used to produce a real life floor height variation 4 to 16cm, the 12cm spread being typical of gently undulating seams. All roller centres were fixed initially at a constant height for ease of rig construction whilst allowing the simulated floor profile to be changed merely by swapping roller diameters. The 8cm maximum diameter unfortunately constrained the number of rollers per tray to only four (X_p being about 1.5m conventionally in real life and therefore 37.5cm when $\frac{1}{4}$-scaled). Now whereas a floor sample spacing of $X_p/4$ would probably be adequate for the identification of an $X_c \geq X_p$ i.e. for the approximate value originally guessed, it is clearly inadequate should X_c turn out to be considerably less than X_p. Unfortunately, applying the test signal to the dynamic programming model using up to 10 samples/tray indicated a value for $X_c < X_p$.

Indeed it was now found impossible to reduce filter constant X_f sufficiently to aatisfy condition 19 and thus properly identify the true X_c. The P.R.B.S. source signal was therefore abandoned in favour of a proven pink noise signal derived digitally from a robust systems identification package (SPAID) developed earlier at Sheffield (10). The noise generation algorithm embodies filtration of far sharper cut-off than is possible with simple low order filters of the type used earlier. Additionally, the random diameter rollers were abandoned in favour of much smaller rollers of constant diameter but now arranged at varying centre-heights as prescribed by the computed test signal. These could now be arranged much closer together allowing up to 10 rollers per tray and, should this spacing prove too coarse, alternate tray joints could be made rigid to double the effective X_p. It was in fact found that, by varying sample-spacing, results converged for about 6 samples (rollers) per tray for both the $\frac{1}{4}$-scale and D.P. models.

3.2. Results

Fig.10 shows a typical cross-correlation function $\phi_{hy}(\ell)$ obtained between the pseudo-random test signal $y(n,\ell)$ and output $h(n,\ell)$ produced by the elastic beam model having an $X_c \equiv 5$ samples. The resulting curve was found to be widely tolerant of the value of X_c and number of samples constituting the test sequence. It clearly accords closely with the analytically predicted impulse response curve of Fig.7 though in Fig.10 and all following cross correlations only one side of the symmetrical response is shown. The result confirms the adequacy of the test signal.

Fig.11 shows the input signal $y(\ell)$ after sampling for the purpose of fixing the roller heights of the modified rig together with the responses $h(\ell)$ produced by both the D.P. computer model and the $\frac{1}{4}$-scale mechanical model itself. Two responses for the latter are given (a) with the rear skids of the machine located at the centre of a tray and (b) with the skids at an inter tray joint. The results illustrate the excellence of the D.P. fitting programme which however allows no rocking of the a.f.c. trays on floor undulations under the machine weight. The model results show a tendency to follow the floor profile in greater detail as a result of this tray movement and that this phenomenon is most pronounced in case (b) as would be expected.

The resulting cross-correlations for the three cases are shown in Figs. 12 and 13 from which the inferred values of X_c (the equivalent elastic beam parameter) are given in Table 1. These values are obtained by inspection of the position of the zero crossing of the smoothed curves which should occur at a value of ℓ slightly greater than $2X_c$ as Fig.7 shows.

Table 1

Equivalent elastic beam parameters from cross correlation curves

Model	X_c/X_p	X_c/X_T *
Dynamic programming (minimum potential energy model)	0.333	0.476
$\frac{1}{4}$-scale model (a) skids at tray centre	0.250	0.357
$\frac{1}{4}$-scale model (b) skids at joint	0.167	0.238

$^*X_T$ here assumed = X only = $0.7X_p$ for conventional systems and a.f.c. tray lengths

The three results for X_c/X_p are of similar order of magnitude, the elastic beam equivalent to the D.P. model clearly being the stiffest whereas X_c is least when the machine skids are located at the tray joints as expected. The R.H. column of Table 3 predicts that all three models are close to the critical stability ratio of $X_c/X_T = 0.4$ predicted by the elastic beam theory (inequality 15). The D.P. model is predicted as being just stable whilst the ¼-scale model is unstable as observed in practice. Indeed if total system delay $X_T > 0.7X_p$ as it is conventionally (e.g. $X = 0.7X_p$, $X_1 = 0.35X_p$, $X_2 = 0.1X_p$ making $X_T = 1.15X_p$ even in the case of $R = 0$) then the D.P. model is also predicted by beam theory to be unstable in multi-pass operation, as has been observed (5).

If, however, a.f.c. tray lengths are considerably increased from their conventional values such that now $X_p = 2X_T$ {condition (16) established for the stability of rigid section conveyors in Section 2.2.3} then the ratio X_c/X_T entered in Table 1 would become 1.36, 1.02 and 0.68: all now satisfying the elastic beam stability criterion $X_c/X_T > 0.4$.

4. CONCLUSIONS

The random signal identification tests have thus produced equivalent values for X_c that reconcile elastic beam model predictions, dynamic programming and ¼-scale model simulation trials and analytic z-transform model predictions for long rigid trays. It can be concluded that the effective stiffness of conventional a.f.c. structures using conventional tray lengths is insufficient for stability and indeed is so small as to permit the a.f.c. to be disregarded in stability studies. For tray lengths $X_p \geq 2X_T$ then tray rigidity becomes important and an X_c/X_p ratio in the range 0.167 to 0.333 may be used for an equivalent elastic beam representation for analytical stability studies.

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7. NOTATION

a.f.c.	=	armoured flexible conveyor
D.P.	=	(general) dynamic programming
E	=	Young's modulus for elastic beam material
F	=	spacing between machine skids along-face
$G_c(s)$	=	transfer-function of conveyor
$G_s(s)$	=	transfer-function of steering system
h	=	height of conveyor above flat datum
I	=	2nd moment of area of beam about neutral axis
J	=	upward deflection of cutting drum
J_d	=	demanded drum deflection
k_f	=	restoring force of fine coal p.u. length of a.f.c. p.u. bed compression
k_h, k_g	=	height and tilt gains of controller
ℓ	=	distance travelled along-face
L	=	face length
multipass process	=	process involving repeated passes over the process material with continuous interaction between passes
n	=	pass number
pass	=	cut
R	=	horizontal distance (along face) between drum centre and rear skids of cutting machine
tray	=	rigid segment of a.f.c. structure
W	=	drum and conveyor width
X	=	coal sensor delay distance
X_1	=	coal sensor time-constant x machine velocity

X_2	=	hydraulic system time-constant x machine velocity
X_c	=	characteristic distance of elastic beam model
X_p	=	tray length
X_T	=	total system delay distance
y	=	thickness of coal ceiling (or roof) left by drum
y_m	=	measured value of y
y_r	=	desired reference thickness
z	=	deviation of coal seam from flat datum
α	=	tilt of a.f.c. and machine in face-advance direction
β	=	tilt of machine in along-face direction
ω	=	angular frequency in radians p.u. length,
$\phi_{hy}(\ell)$	=	cross-correlation function between input floor profile $y(n,\ell)$ beneath a.f.c. and output height profile $h(n+1,\ell)$ of a.f.c.

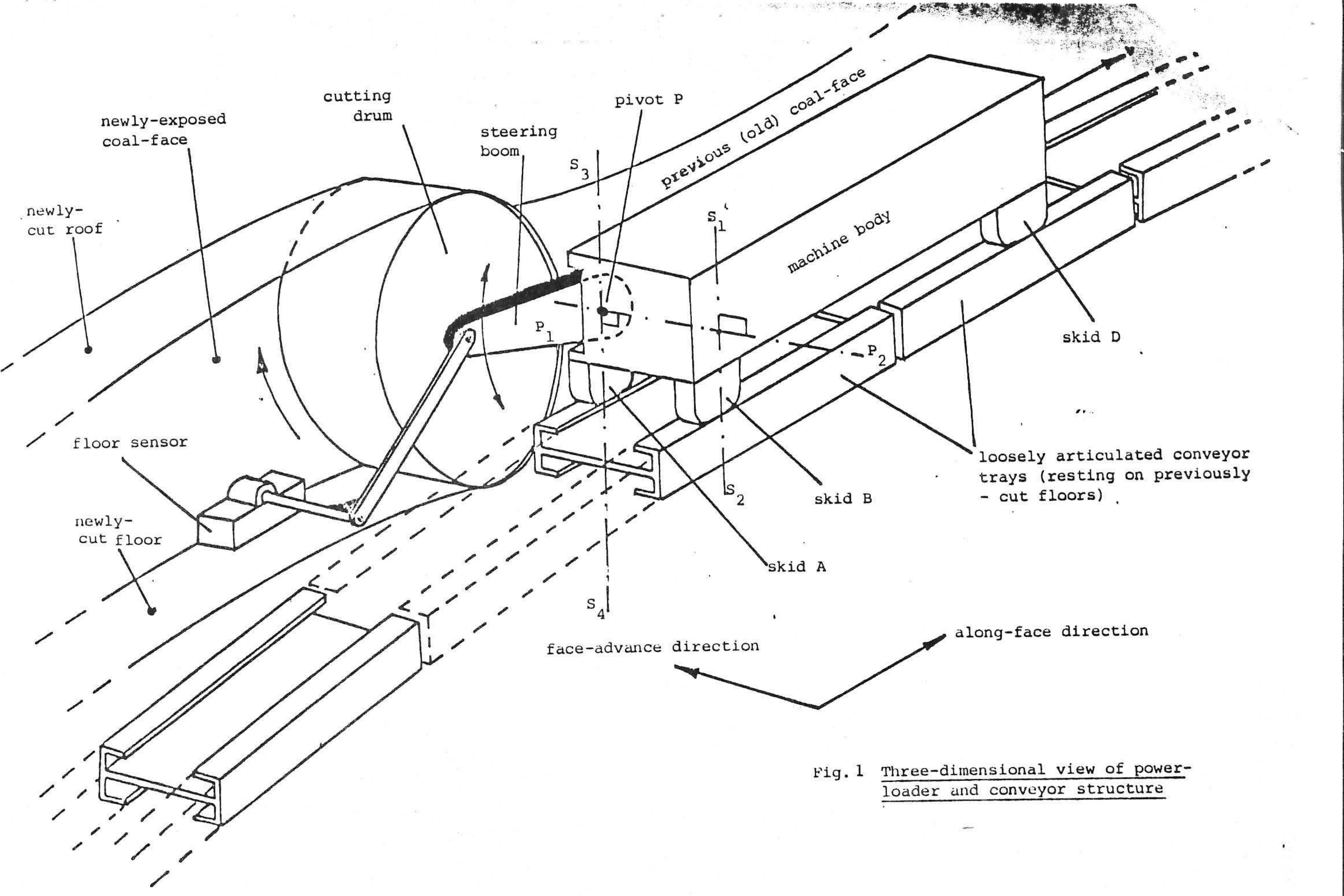


Fig. 1 Three-dimensional view of power-loader and conveyor structure

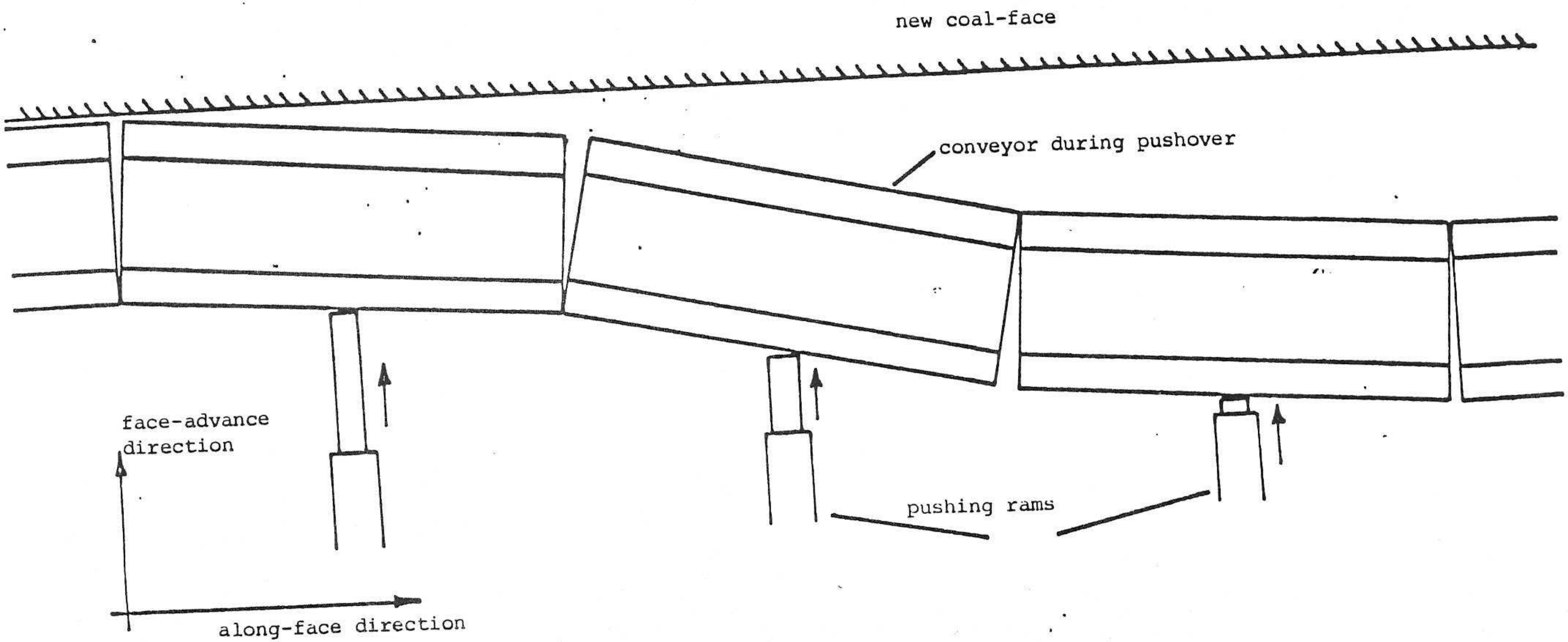


Fig. 2 Showing conveyor being advanced between cuts

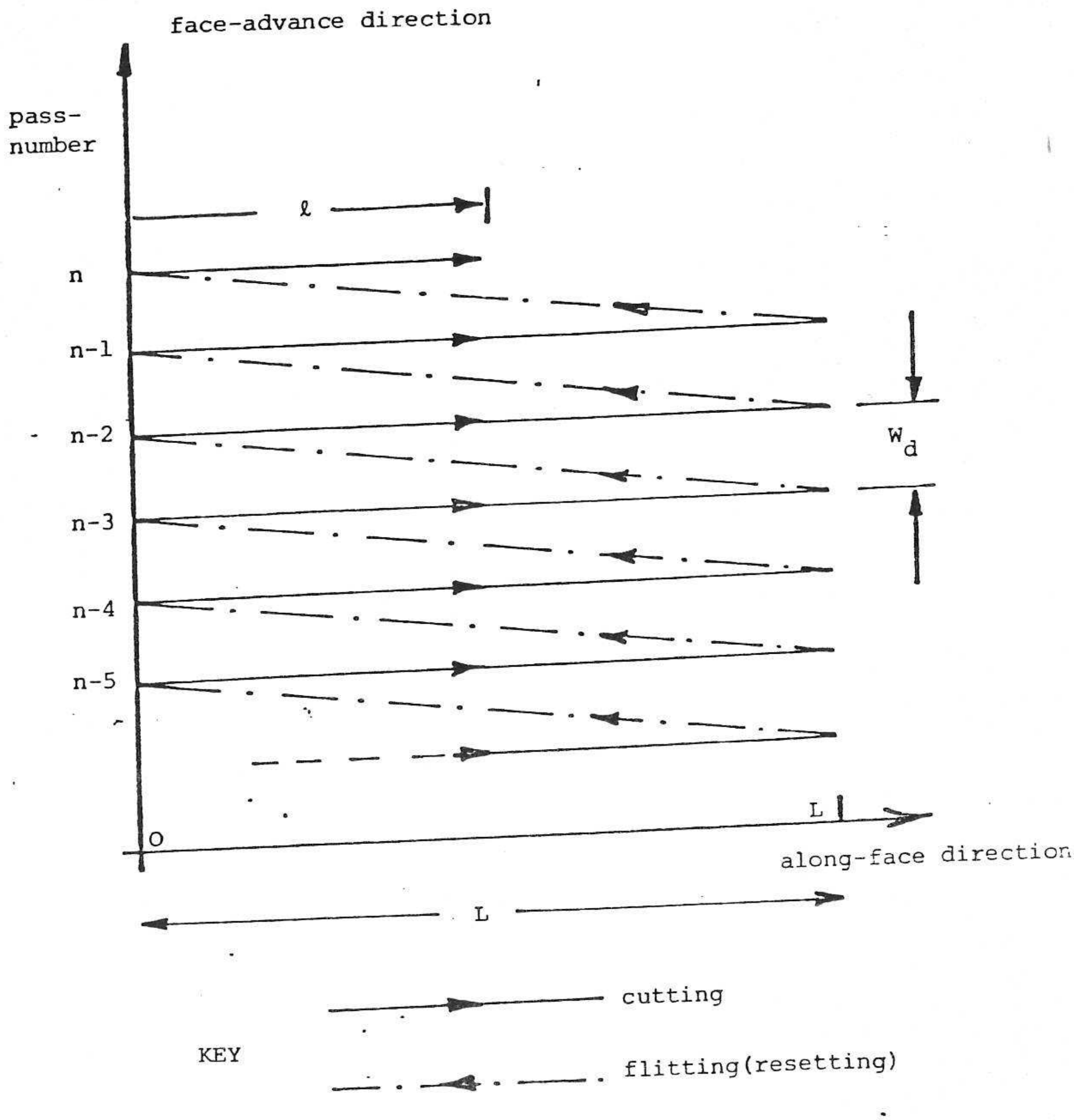


Fig. 3 The system of unidirectional longwall coal mining (schematic plan view)

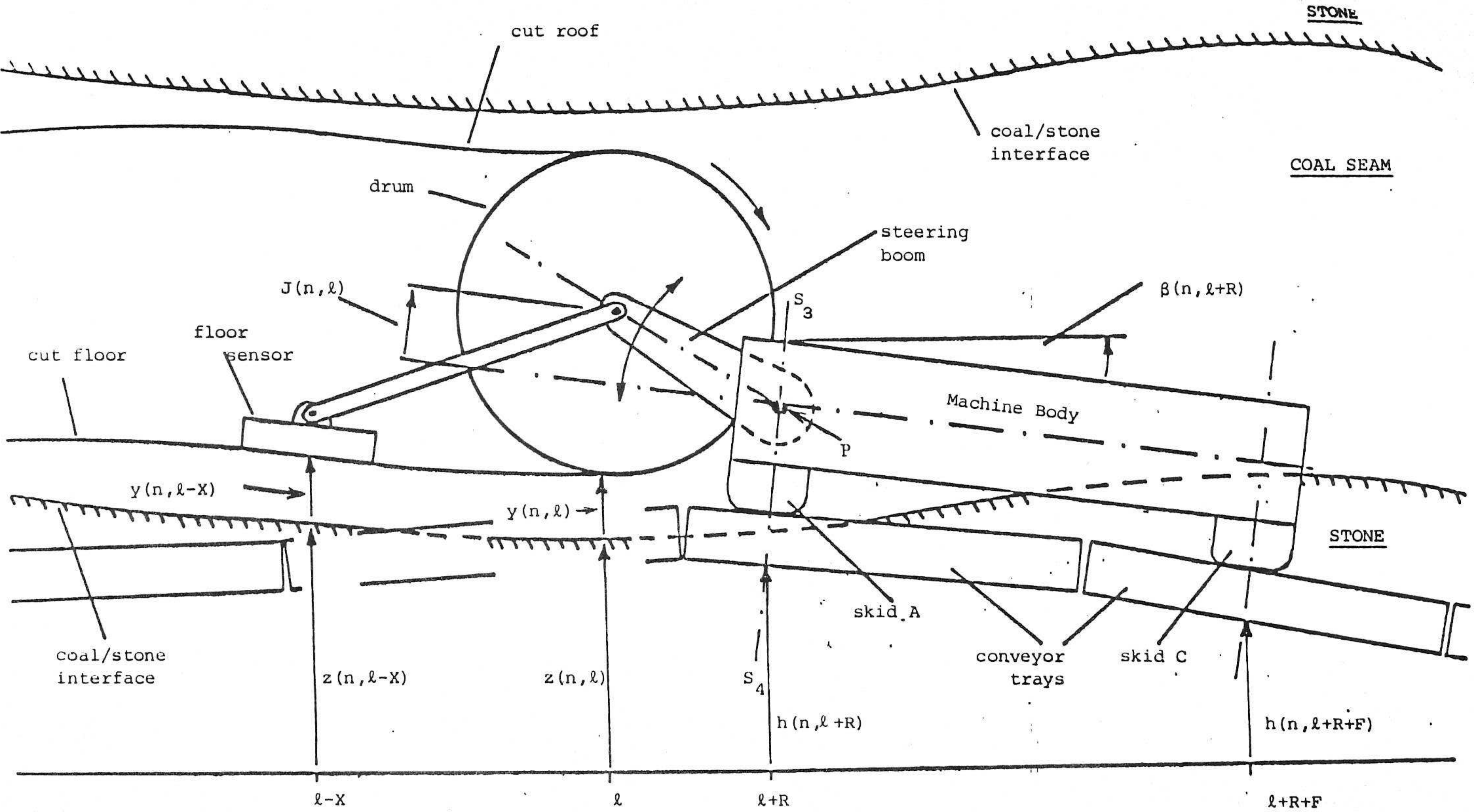


Fig. 4 Side-elevation showing process variables (all measured on the extreme face-side of the system)

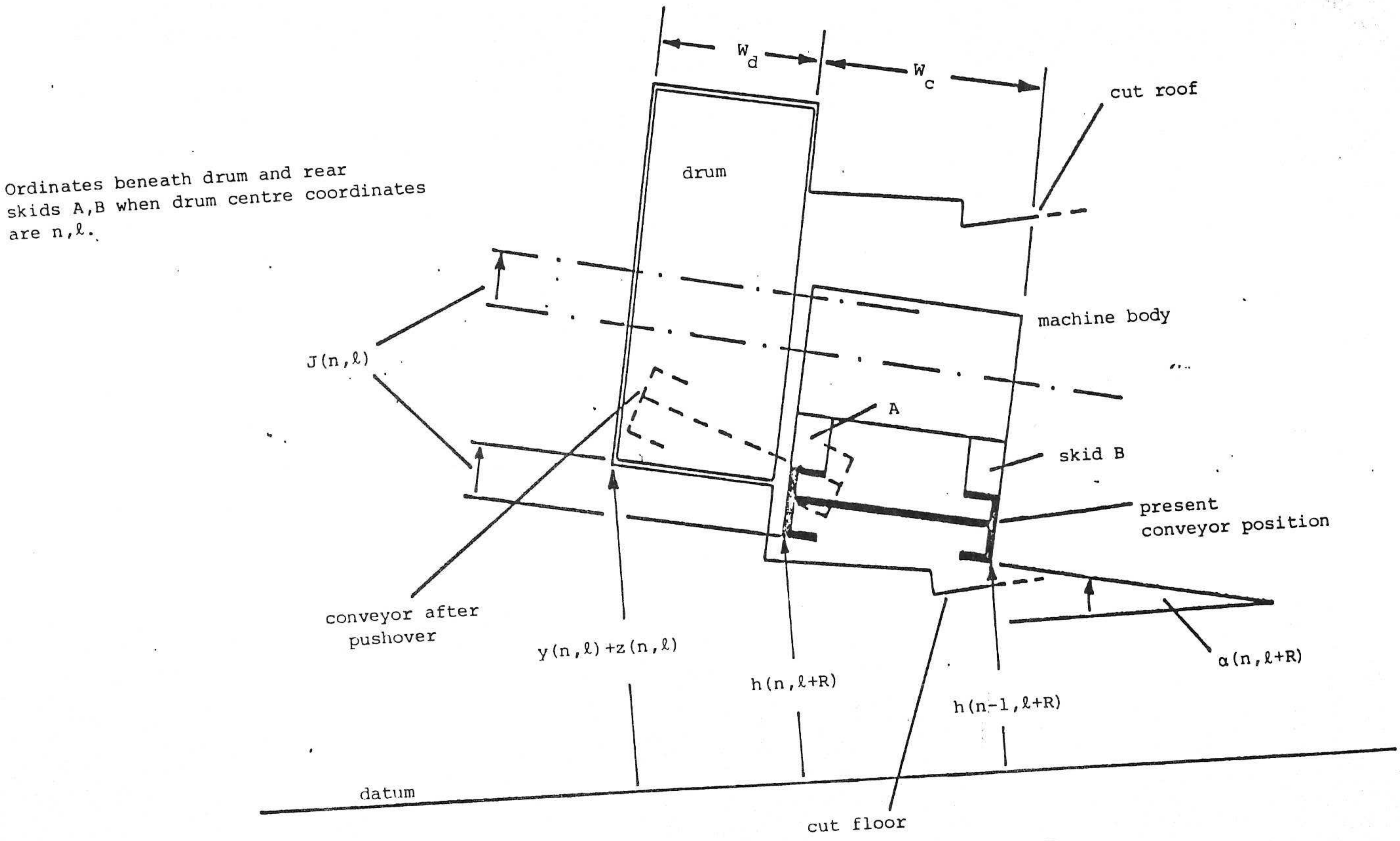
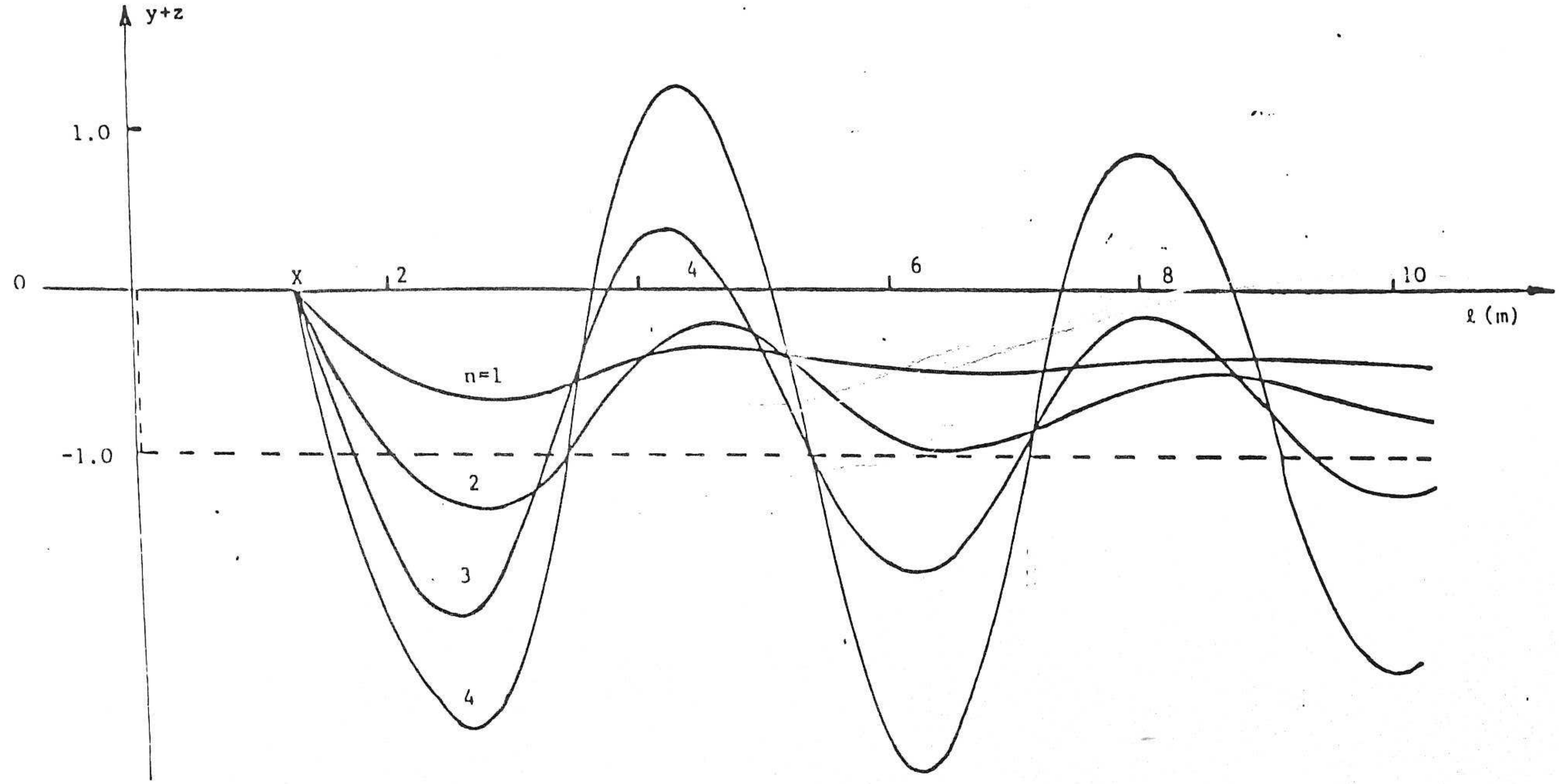


Fig. 5 End-elevation

Fig. 6 Simulated response of rubber-conveyor model to unit downward step in coal-seam



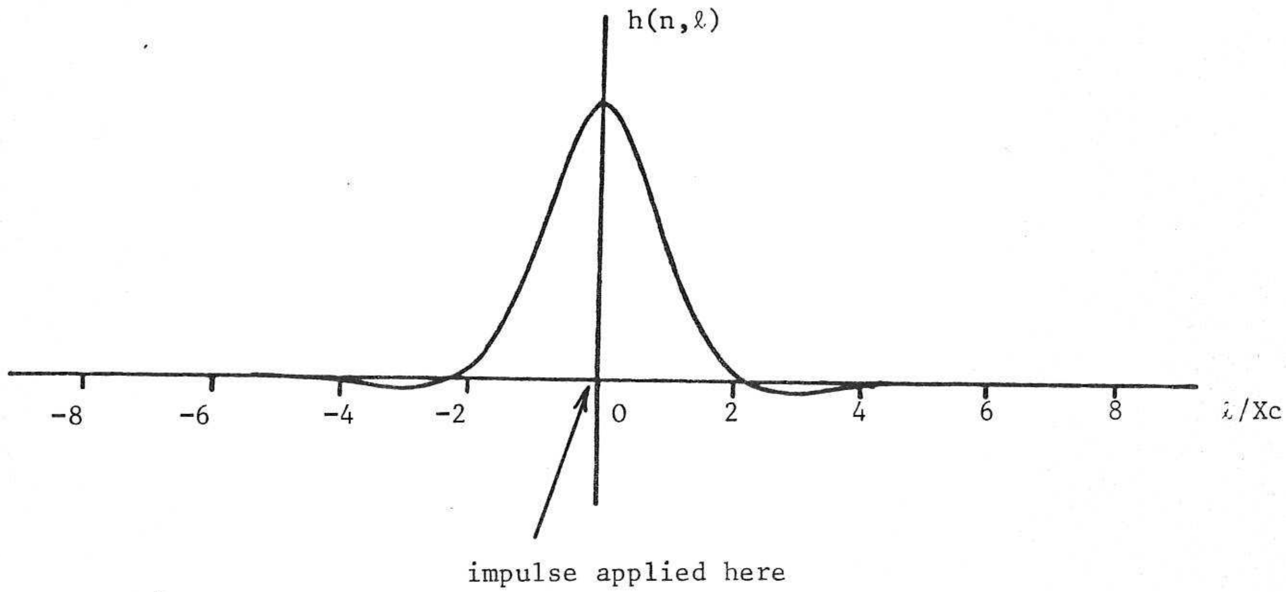


Fig 7. Impulse - response of elastic beam model

$x_c = 0.5\ X$

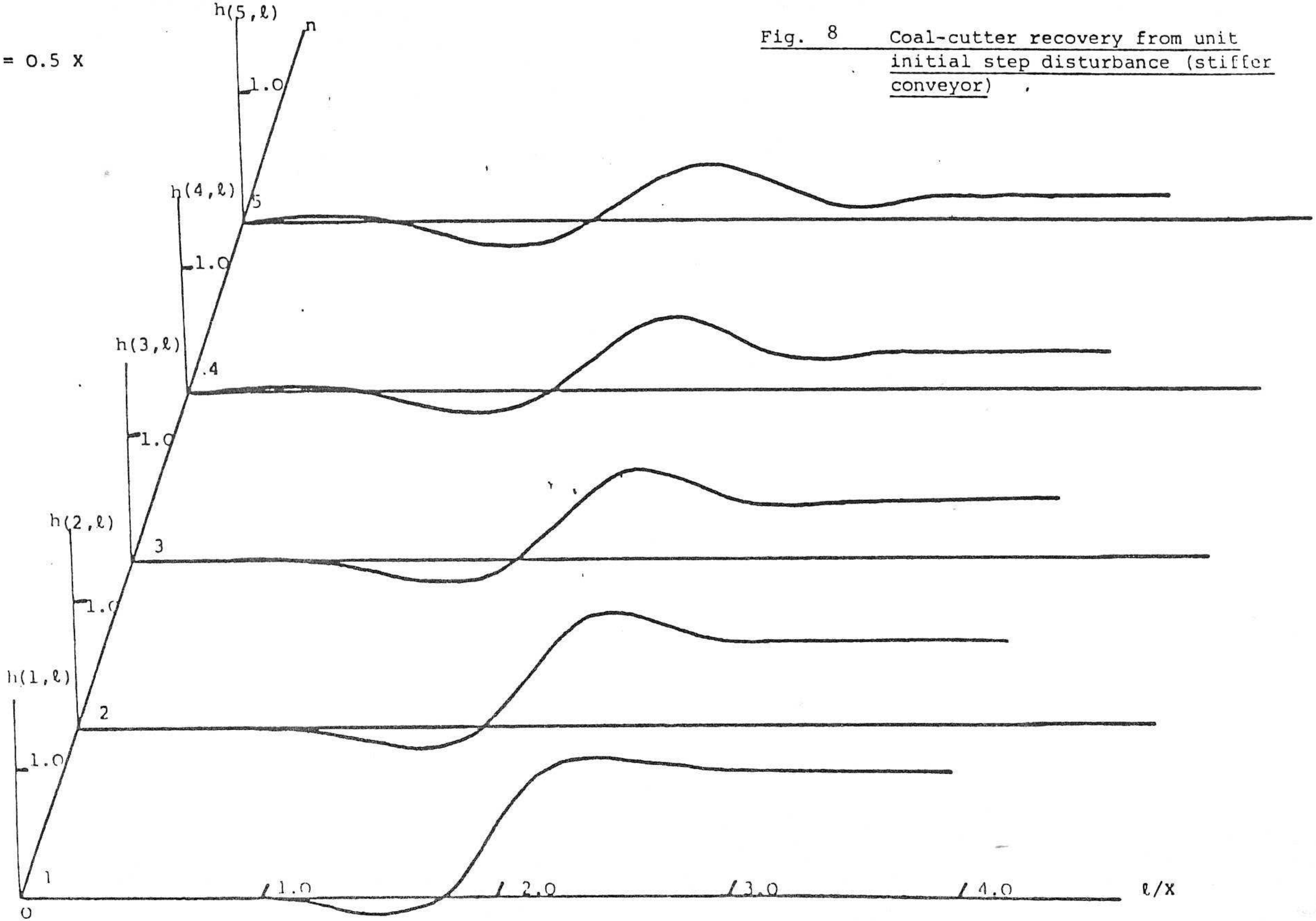
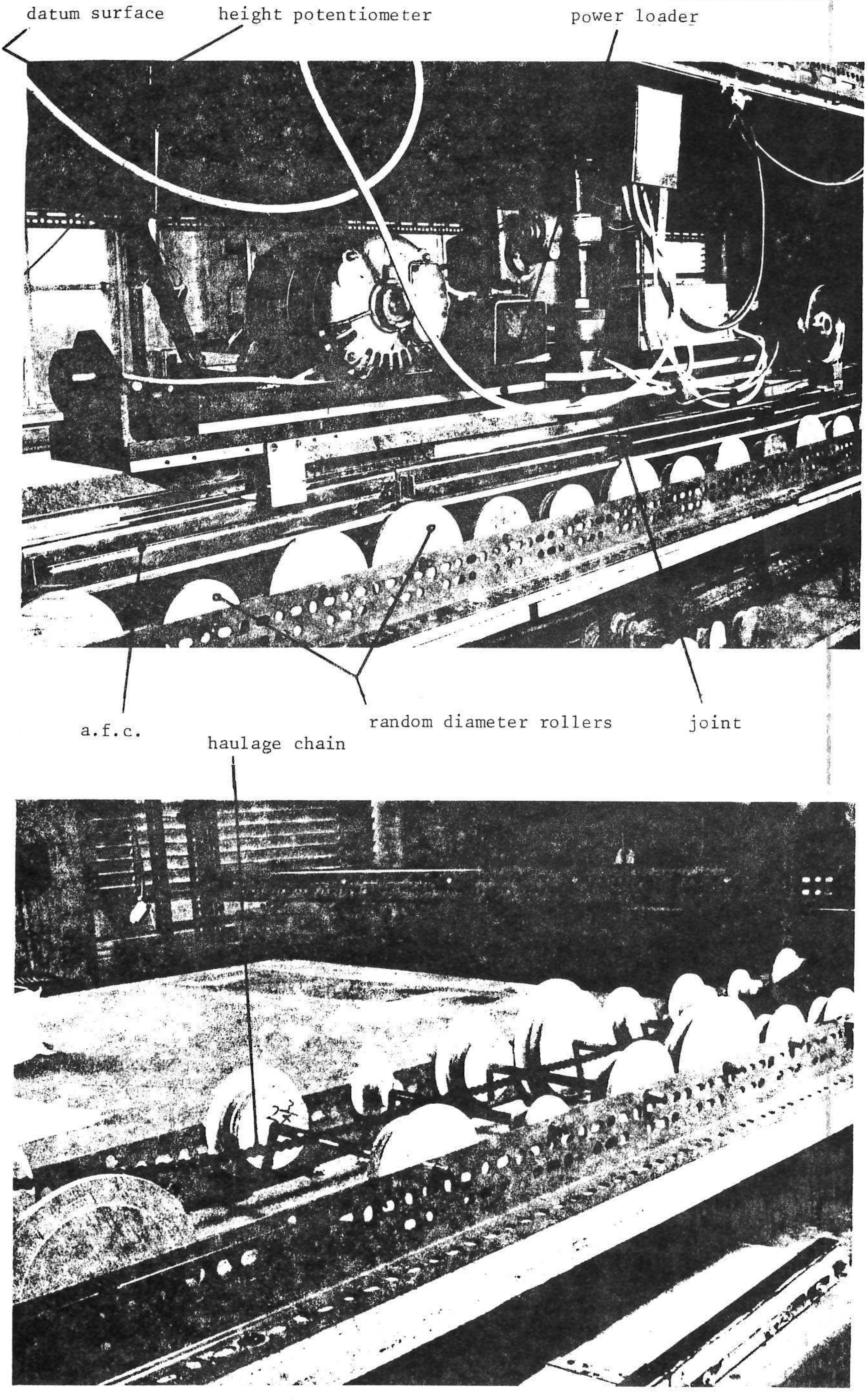


Fig. 8 Coal-cutter recovery from unit initial step disturbance (stiffer conveyor)

Fig.9. $\frac{1}{4}$-scale model machine and a.f.c. on random roller test track



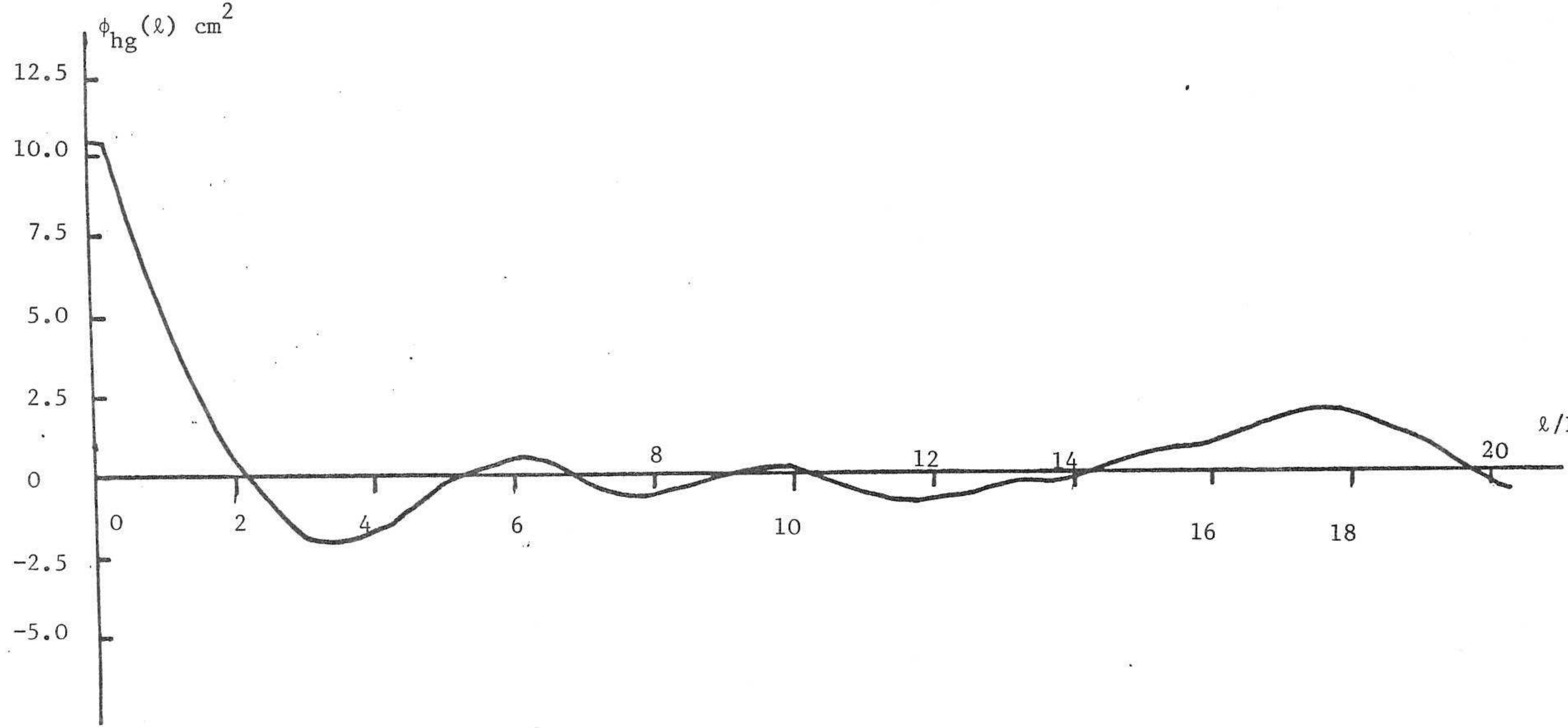


Fig 10. Cross-correlation function from elastic beam model

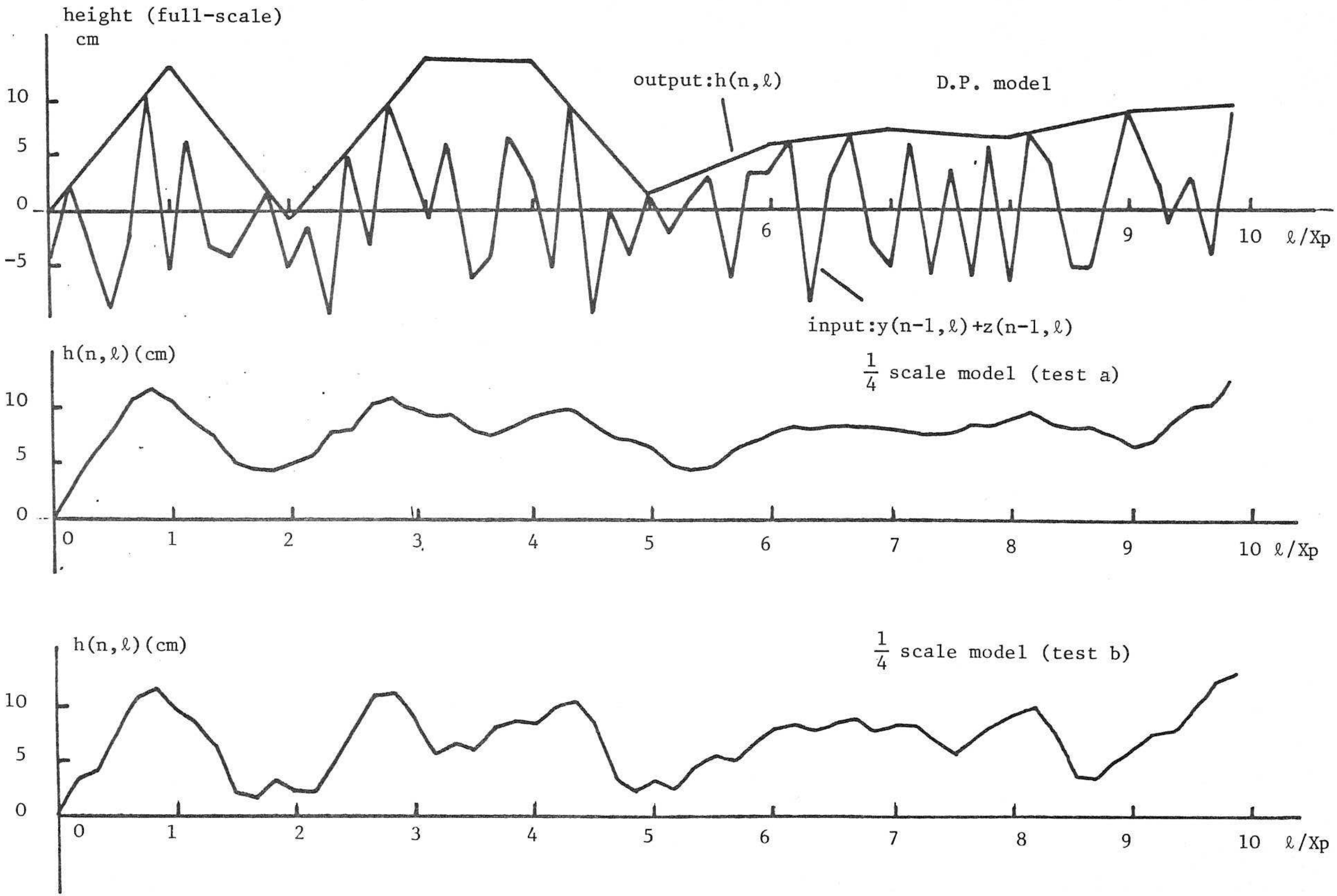


Fig 11. Responses of a.f.c. models to pseudo-random test signals

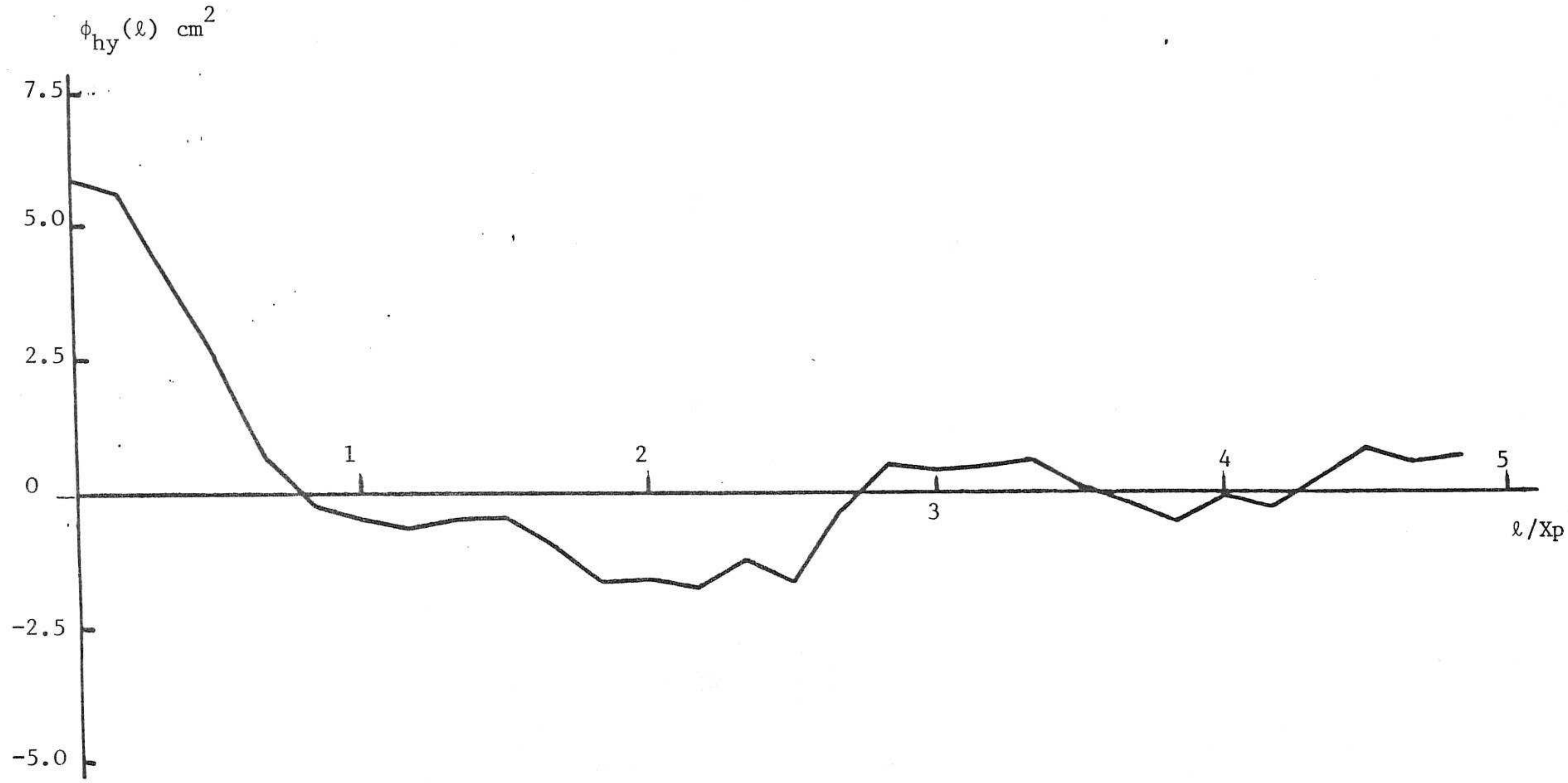


Fig 12. Cross correlation function from D.P. model

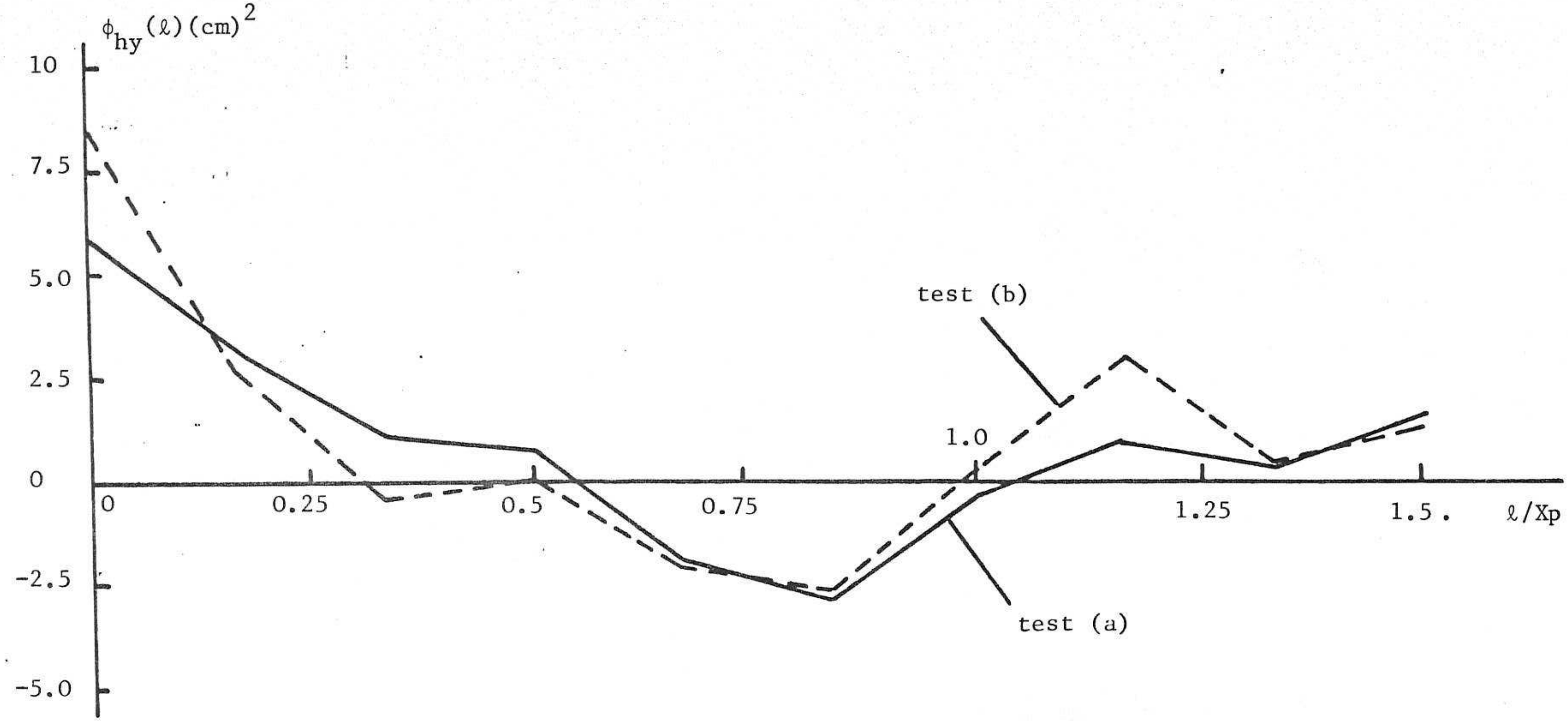


Fig 13. Cross-correlation functions from $\frac{1}{4}$ scale model