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SYMPOSIUM ON MODERN CLEAN ROOM TECHNOLOGY
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THE USE OF COMPUTATIONAL MODELLING TECHNIQUES FOR CLEANROOM DESIGN

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1. INTRODUCTION

This paper presents the results of a computational and experimental study to assess the possible benefits of using mathematical modelling techniques for cleanroom design. Specific attention is paid to the separate problems of flow within HEPA filter plenums and the general flow environment within a class 100 cleanroom.

A two-dimensional modelling visualisation has been used for each of the cases studied. This has the advantage that its relative cheapness allows more design variations to be analysed than would be possible with a three-dimensional method. Full-scale measurements were also taken in the clean room which was the subject of the modelling exercise. The predictions of the computational model were found to be in good agreement with the measurements taken.

The results of this brief study suggest that significant cost saving may be achieved by incorporating methods of computational analysis in the cleanroom design procedure. The analysis technique used in this work can also provide a better understanding of the performance of existing cleanroom facilities.

2. TYPES OF PLENUM AND CLEANROOM

2.1 Plenum

According to Marsh and Quinn (1981) the HEPA filter is the key element in the laminar flow clean air system. It can be shown that the greatest inefficiency in the traditional laminar flow clean air module is in the method of air distribution from the fan to the HEPA filter. Figure 1 shows four of the many plenum designs which have been proposed to facilitate the necessary air distribution. Such designs are all very well in theory, but many cleanrooms are constructed in existing buildings where space is at a premium. In such circumstances a much simpler plenum design may be attractive, such as that shown in Figure 2.

2.2 Cleanrooms

Figure 3 shows that the two major classifications of clean room as defined in BS5295 (1976).

Figures 3a and 3b are laminar cleanrooms in which the entire 'body' of air within the room moves, ideally with uniform velocity, along parallel flow lines which may be either horizontal or vertical. It is worth noting in passing that whilst the flow of air in such clean rooms may be laminar when the room is completely empty, the introduction of equipment, work benches and personnel undoubtedly lead to turbulent air flows being generated. Figure 3c is 'defined' as a 'conventional flow cleanroom', i.e. a cleanroom in which the flow is not unidirectional or of a uniform velocity.

The cleanroom design which was used for the modelling exercise is shown in half-section in Figure 4. This design was chosen because an existing cleanroom was available for use in obtaining full-scale measurements.

3. THE MATHEMATICAL MODEL

The computational prediction work was conducted using the Atkins program CAFE (Computer Aided Flow Evaluation). This program is designed to solve flow problems which exhibit areas of recirculation.

Although this was the first instance of using CAFE to predict air flow within cleanrooms, the program has been extensively used for related problems (see, for example, Broyd et al (1983)).

It is neither desirable nor necessary to present a detailed description of the program CAFE in a paper such as this. The following variables may be solved in either two or three dimensions (depending on the visualisation used):-

- Momentum in each dimension.
- Continuity
- 2 turbulence equations (unless the flow is genuinely laminar)
- Concentration.
- Temperature.

It is possible to model flow domains of any desired geometry, and internal blockages may be used to simulate work benches, items of equipment, personnel, etc. The effect of heat-emitting (or absorbing) equipment or personnel may be modelled by introducing sources (or sinks) of heat energy arbitrarily throughout the model domain, (although it should be pointed out that it is difficult to accurately simulate the effects of people moving around). Likewise the shedding of particles by plant or personnel may be modelled by making the analogy between small particles and concentration of gases. For example, Figure 5 indicates that there is little loss of accuracy by assuming that particles of $< 10\ \mu m$ remain associated with individual "parcels" of air flowing through the cleanroom.

A full description of the solution technique used for CAFE may be found in Dean and Moult (1978).

4. COMPUTATIONAL PREDICTIONS

4.1 Supply Plenums

Two tests are presented, representing the following:-

Run 1. A baseline test which uses the physical geometry and fluid flow specification given in Figure 2 and Table 1 respectively.

The filter is represented by an expression, (equation 1) derived from the manufacturers' flow characteristic (Figure 6). The relationship calculates a value of equivalent filter porosity (on average = 0.0325) based on the filter local exit flow velocity (Vo) and the pressure drop (ΔP) across the filter

$$\Delta P = 296.2\ Vo^{1.1062} \qquad (1)$$

Run 2 A test to represent a dirty filter at its maximum recommended pressure drop of 450 N/m^2. Equation 1 was modified to represent a dirty filter by applying a pressure axis shift to give equation 2.

$$\Delta P = 968.8\ Vo^{1.1062} \qquad (2)$$

The results obtained are illustrated as streamline plots on Figures 7 and 8. In addition the outlet vertical velocity profile is also given for each test case in Figure 9.

The streamlines plotted represent the streamlines at 10%, 20%, 30% etc. of the mass flow rate.

The results may be summarised as:-

a) The presence of a 'low porosity' filter causes a high plenum pressure and strong recirculation within the supply ductwork. The use of a single plenum results in a non-uniform velocity distribution along the filtered area, with variations of around 15% from the mean velocity.

b) Reduction of the filter porosity simulates an old, or dirty filter. The effect of this is to further increase the variation of outlet velocity along the filtered area.

Results from the helium bubble flow measurement technique were used to confirm the hot wire anemometer results. The hot wire results were then used to plot the velocity vectors and profiles given overleaf.

Results

The results of the experiment measurements are shown on Figure 10, which shows velocity vectors of the experimental measured resultant flow velocity superimposed on the stream line plot, for the baseline test model. Measurements were taken at 0.5m, 1.0m, 1.5m and 2.0m above the floor, and at four positions across the half-section of the room.

Comments

Comparisons of experimental and predicted results in general show a reasonable degree of correlation at all measuring heights. The correlation is best where velocities are largest. This is to be expected since measurements at these low velocities are at the lower limit at which a hot wire anemometer will operate accurately.

In general the experimental results confirmed the flow predictions made with a relatively simple computer model and therefore show the possibilities afforded by this type of flow model in design studies for controlled environments such as cleanrooms.

Computer models similar to that used in this work can be developed to give greater degrees of correlation, by including other factors into the model, eg. 3 dimensionality, temperature effects, etc. These would however increase run time costs, but would further help to improve design procedures.

Differences between Predicted and Experimental Results

The differences experienced between predicted and experimental results are attributable to two different sources of inaccuracy

a) Measurement technique errors.

b) Limitations in the computer model.

The first source of inaccuracy includes errors normally associated with hot wire anemometry, eg. temperature drift, contamination of the measuring sensor etc. This source also covers human errors of judgement in alignment and location of the sensor and instrument error.

The second source of inaccuracy includes assumptions made in the flow model, for example the assumption of planar flow. The assumption of planar flow was shown by experiment to be well-founded for the cleanroom studied herein, but cases of more complex geometry may need to be handled somewhat differently. Greater accuracy in flow modelling is possible with more sophisticated models. Also included here are the differences in the flow model domain shape as mentioned earlier and the difference in inlet flow velocity. These are not strictly accuracy problems however, as they could be eliminated by redefining the prediction model.

6. CONCLUSIONS

It has been shown that the air flow through a cleanroom may be well predicted by a mathematical model. The purpose of this paper has been to point out that such a capability exists, and the two components studied have not been refined to optimise their performance, as would happen in a full design study.

The benefits of using computational methods of flow analysis for cleanroom design may be summed up as follows:-

- Prediction of flow patterns
- Prediction of temperature effects
- Prediction of particle flows
- Bad design points picked up at design stage
- Cost effective design.

The disadvantages of using such an approach lie mainly in the factors of cost and time involved. However, such considerations will generally be far outweighed by the knowledge that a given design will perform up to its required standards of operation.

4.2 Cleanroom Environments

Four tests cases are presented overleaf representing the following:

Run 3. A baseline test case based on the specification given in Table 2 and the geometry illustrated in Figure 4.

Run 4. The same specification as the baseline Run 3 above, except that flow is eliminated from the inlet filter, nearest the wall.

Run 5. The same specification as the baseline Run 3 above, except that flow is eliminated from the two end inlet filters, nearest the wall.

Run 6. The same specification as the baseline Run 3 above, except that the outlet duct is reduced in height from 450 mm to 250 mm.

The results of the four test cases are illustrated as streamline plots on Figures 10, 11, 12, 13 respectively. In addition, profiles of vertical and horizontal components of velocity at a working height of 1 m above the floor are given in Figures 14 and 15.

The streamlines plotted represent the streamlines at 10%, 20%, 30% etc. of the mass flow rate, as before. Extra streamlines are plotted in regions of recirculating flow, for example Figure 11. In these diagrams a 'dividing' streamline is drawn, between the main flow and the recirculating region, and also four other streamlines are drawn which represent equal intervals between the dividing line and the stagnation point or line. Thus it should be noted that equal spacing between the streamlines either side of the dividing line do not signify equal mass flows. In reality the recirculating air flows are a very small percentage of the total air flow within the room.

Whilst the presence of recirculation flows within cleanrooms is generally considered unwelcome, it is possible to consider each case on its own merits if the flow environment may be accurately predicted at the design stage.

The results may be summarised as:-

a) Reduction of the filtered air supply from part of the ceiling area (eg. Figure 11) causes significant localised effects, but has little effect on the bulk air flow through the room. Thus the prime effect of reducing the filter area by 28%, the difference between Runs 3 and 5, is to produce a small area of weak recirculation.

b) Reduction of the size of the air outlet duct has little effect on the bulk air flow through the room, but does of course produce larger outlet velocities.

5. EXPERIMENTAL MEASUREMENTS

Measurement Techniques

Low velocity flow under atmospheric conditions presents a difficult measuring environment for most common flow measurement techniques; eg. pitot tubes, hot wire anemometry etc. Two different flow measurement techniques were used in this work, enabling comparisons of the results from different methodologies to be made.

Measurements were made with a single wire hot-wire probe yawed through 90° to locate a maximum reading when the wire was aligned perpendicular to the local flow direction. Both the magnitude and direction of the local velocity was obtained with this technique.

The second measurement technique involved tracking neutrally buoyant helium bubbles introduced into the flow. Knowledge of the camera shutter speed used to obtain pictures of tracks formed by individual bubbles enabled the magnitude of the velocity to be calculated.

7. REFERENCES

1. Broyd, T.W., Dean, R.B., Moult, A., Oldfield, S.G., (1983) "Safe Ventilation of Industrial Buildings", to be presented at the International Conference on Heat and Fluid in Nuclear and Process Plant Safety; I. Mech. E., London, 17-18 May 1983.

2. BS5295 (1976) "Environmental Cleanliness in Enclosed Spaces, Part 1", British Standards Institution.

3. Dean, R.B., and Moult, A., (1978) "The CAFE2ST Computer Program for the prediction of two-dimensional, recirculating flows", Atkins Research and Development (internal report), Epsom, Surrey.

4. Marsh, R.C., and Quinn, T.B. (1981) "Energy Savings in Laminar Flow Clean Air Systems". Solid State Technology, July 1981.

Table 1

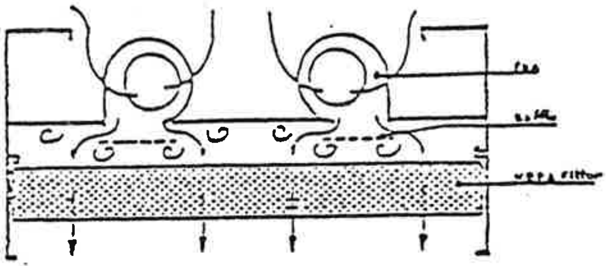
FLOW MODEL SPECIFICATION
DUCTWORK MODEL

Geometry:	
Length of duct (m)	12.20
Width of duct (m)	0.762
Height of duct (m)	0.940
Filter thickness (m)	0.078
Inlet duct (m)	0.7 x 0.762
Filter porosity (average)	0.0325
Fluid Properties:	
Inlet mass flow rate (m^3/s)	4.25
Inlet pressure (bar)	1.00
Inlet temperature (K)	288
Inlet density (kg/m^3)	1.293
Inlet viscosity (Ns/m^2)	1.85×10^{-5}
Inlet velocity (m/s)	7.96

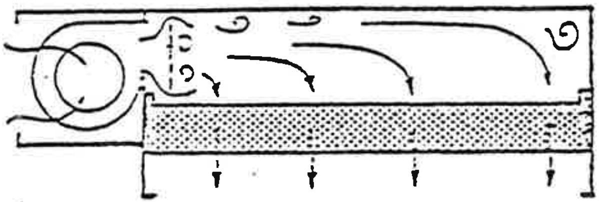
Table 2

FLOW MODEL SPECIFICATION
CLEANROOM MODEL

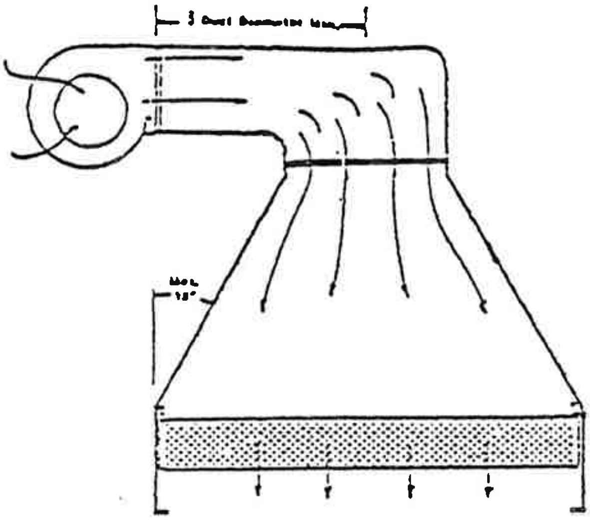
Geometry:	
Width of room (m)	9.011
Height of room (m)	2.825
Inlet Filter size (m)	1.247 x 0.623
Gasket separating inlet ducts (m)	0.003
Distance between centres for inlet ducts (m)	0.626
Size of dead space in roof (m)	0.122
Height of outlet duct (m)	0.450
Fluid Properties:	
Inlet air velocity (m/s)	0.3
Density of inlet air (kg/m^3)	1.293
Inlet pressure (bar)	1.0
Viscosity of inlet air (Ns/m^2)	1.83×10^{-5}
Inlet air temperature (K)	288



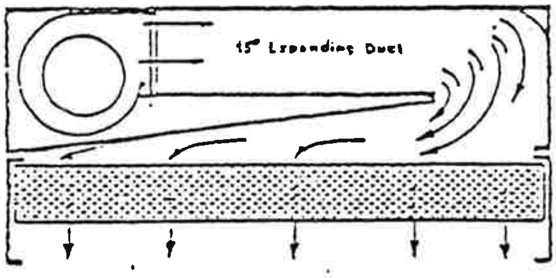
a) —Class 100 laminar flow air module with small multiple fan units.



b) —Class 100 laminar flow module with larger fan unit.



c) —Ideal Class 100 laminar flow module to minimize air distribution turbulence losses.



d) —Experimental "energy efficient" module resulting in 53% energy savings.

Figure. 1 Proposed filter plenums

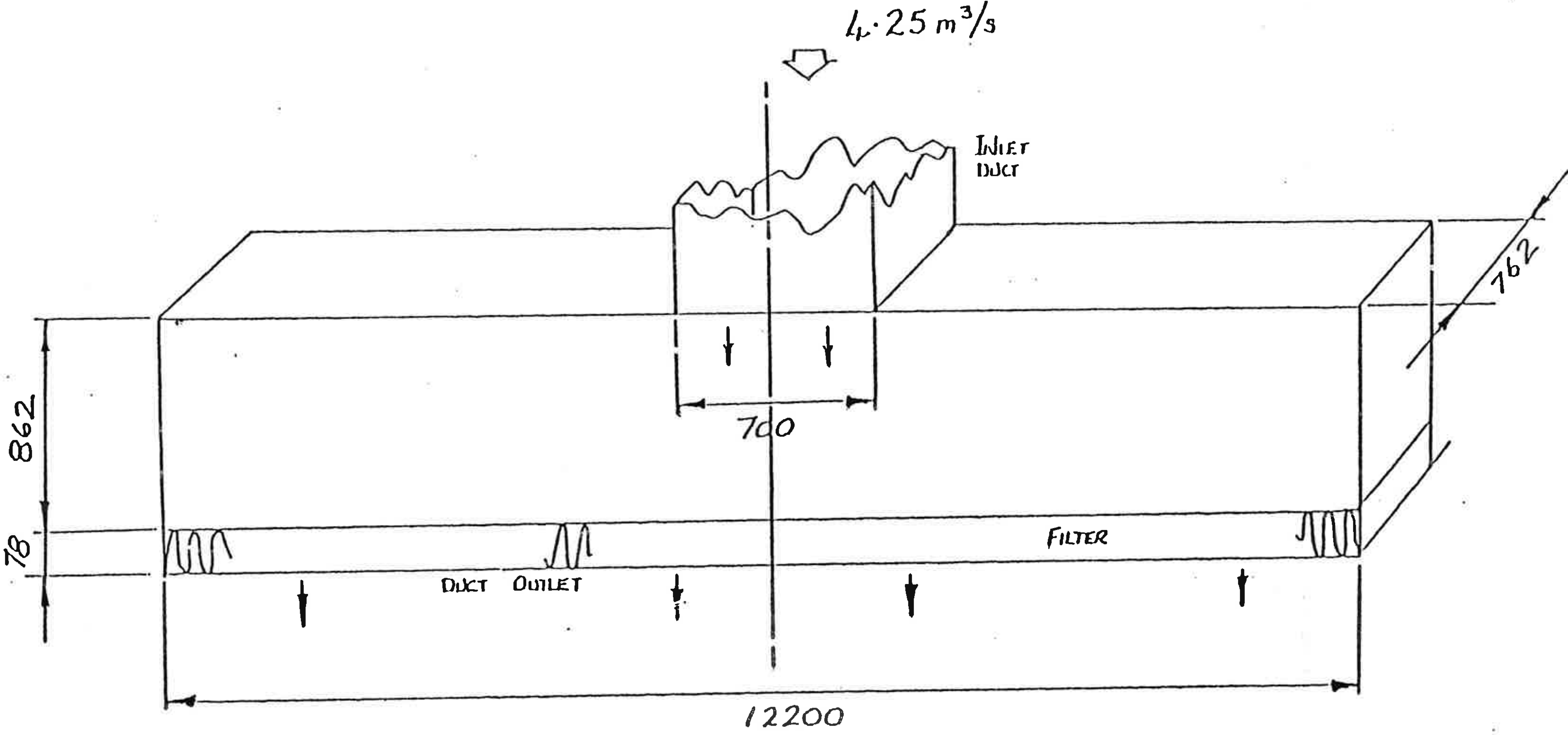
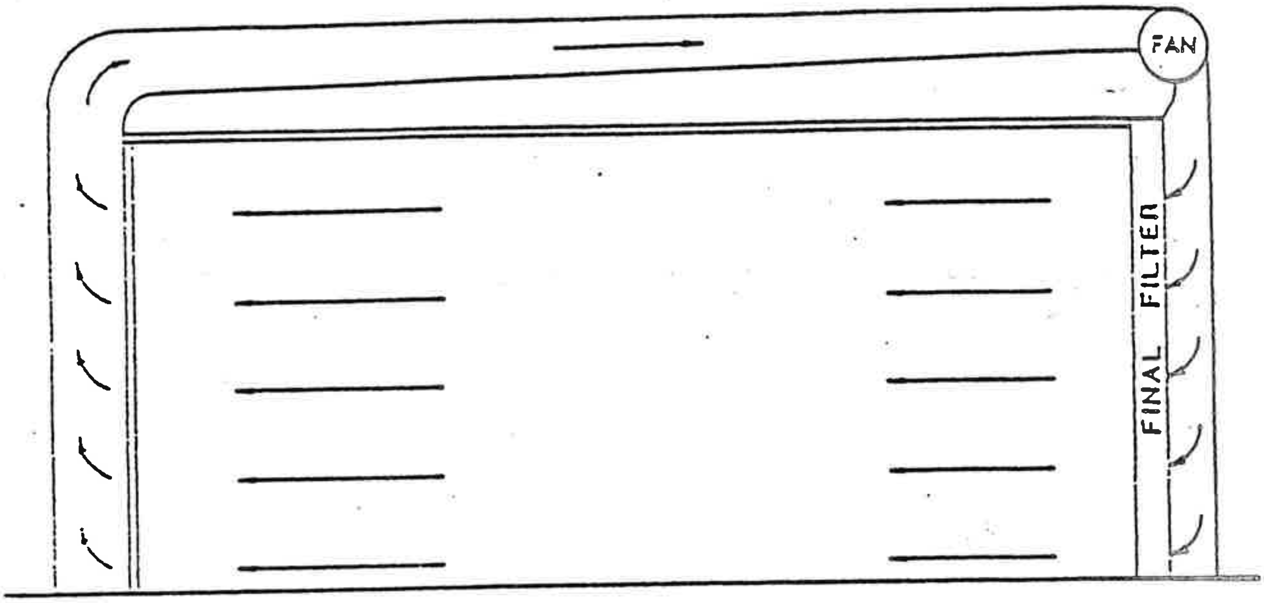
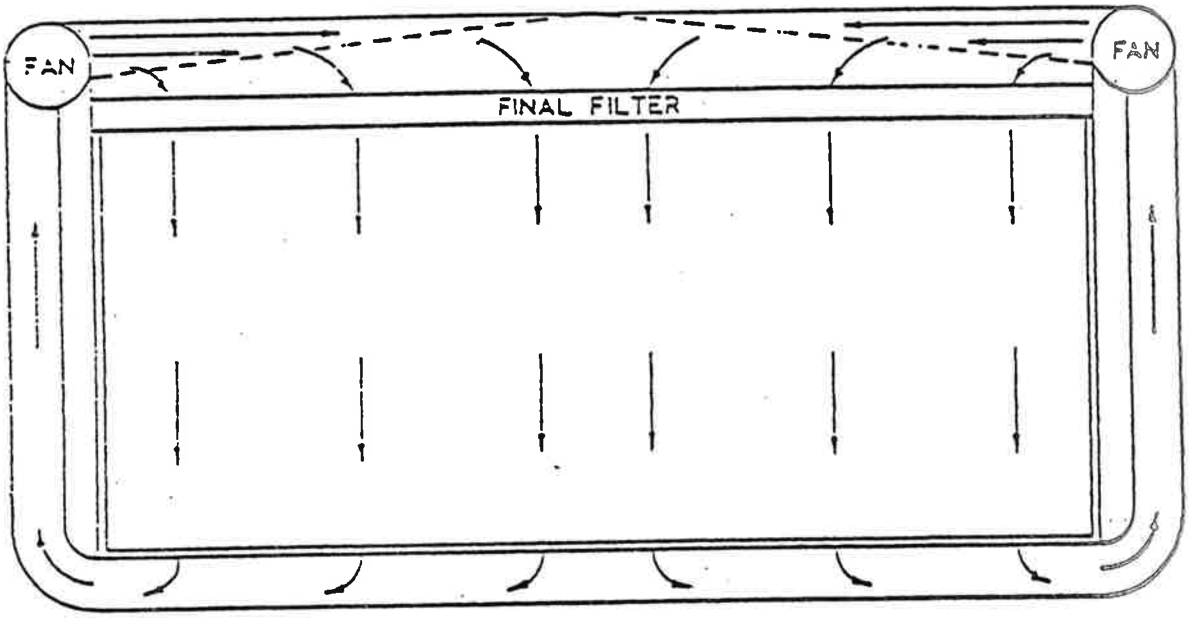


FIGURE 2 GEOMETRY OF SUPPLY DUCTWORK MODEL

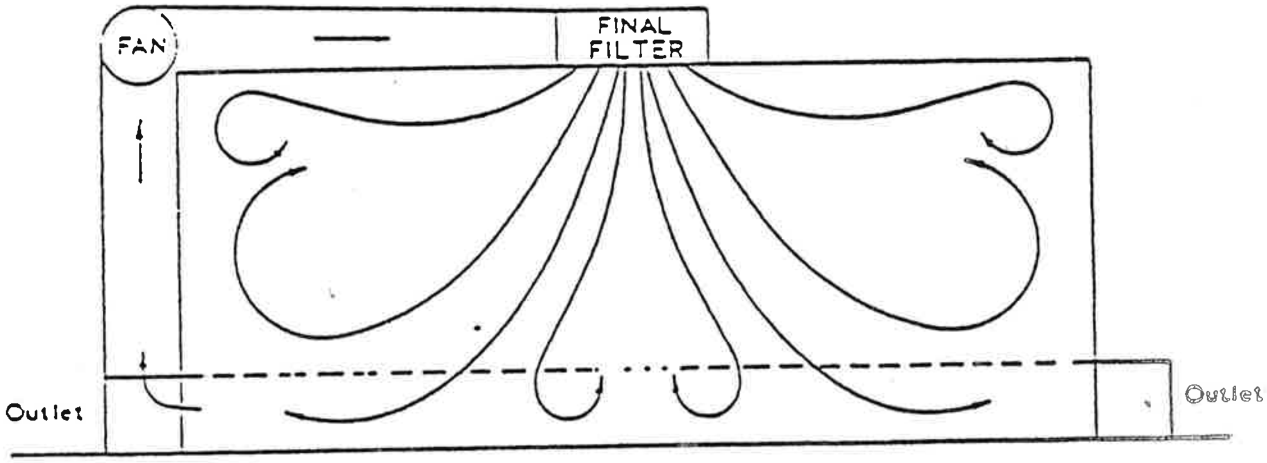
NOT TO SCALE
ALL DIMENSIONS ARE IN mm.



(a) Unidirectional horizontal flow clean room



(b) Unidirectional vertical flow clean room



(c) Conventional flow clean room

Figure 3 Diagrams showing typical air flow (schematic layouts)

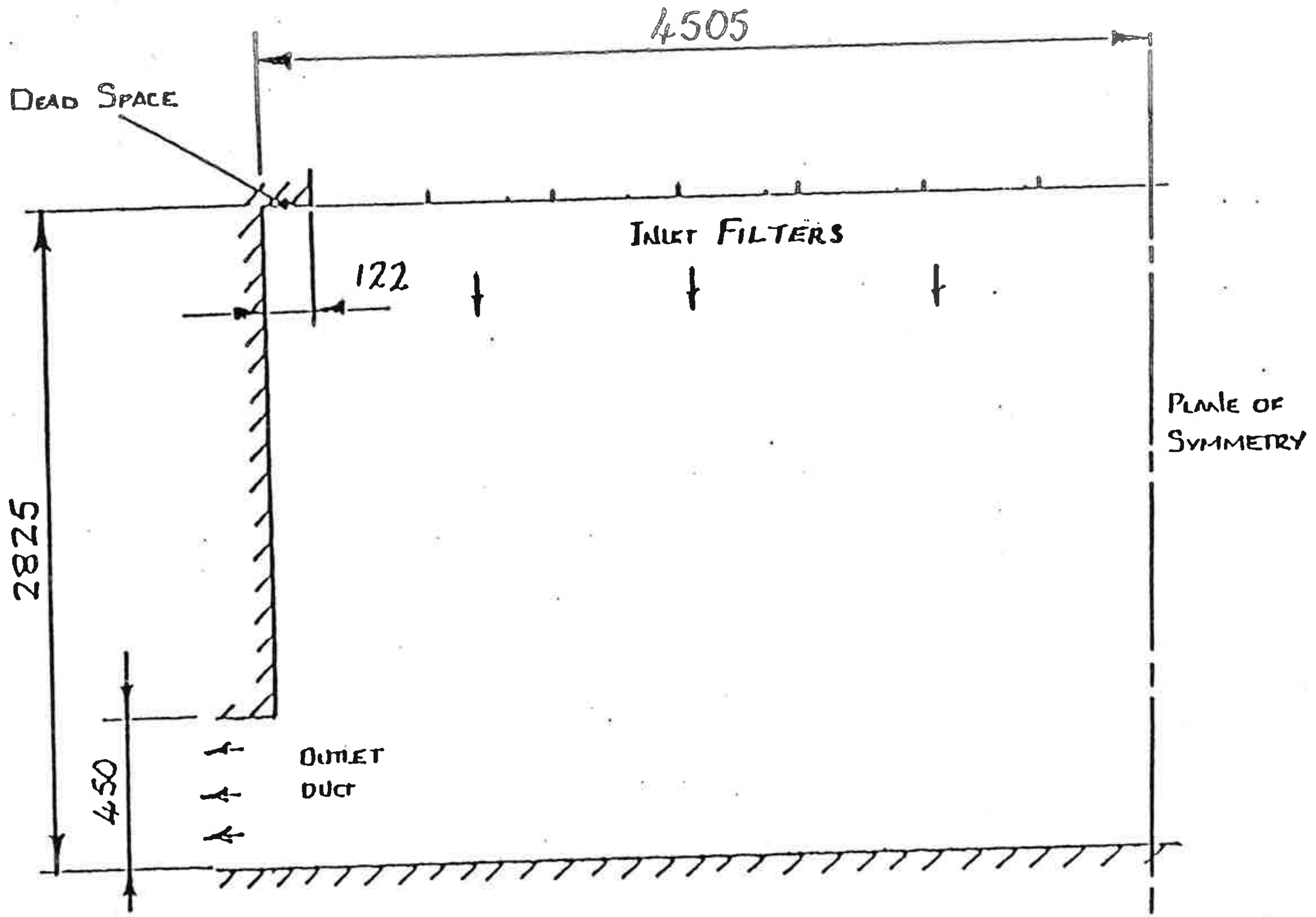


FIGURE 4 GEOMETRY OF CLEANROOM MODEL (baseline test case)

NOT TO SCALE
ALL DIMENSIONS ARE IN mm.

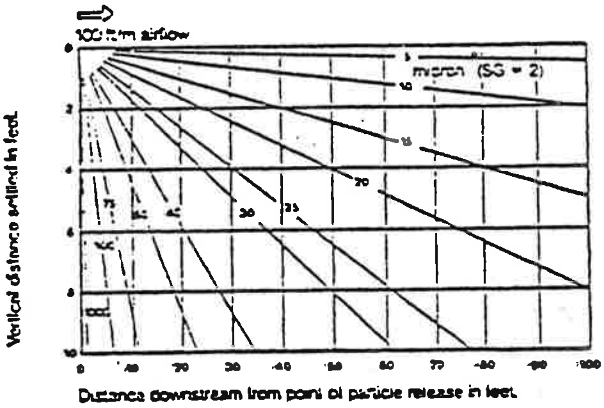


Figure 5 Particle fallout curves

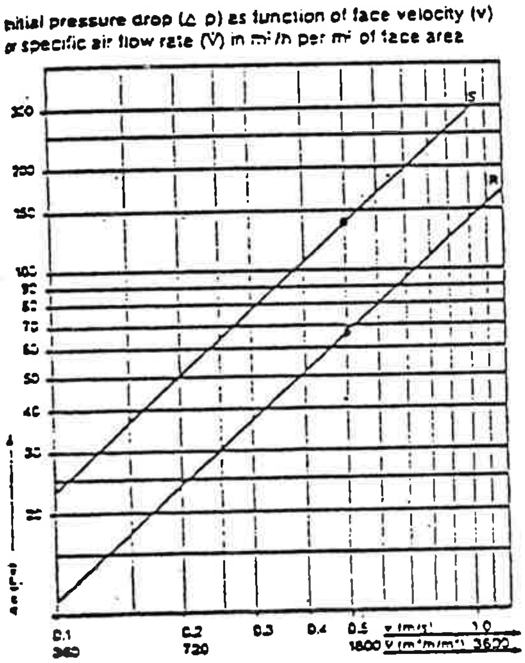


FIGURE 6 FILTER MANUFACTURERS FLOW CHARACTERISTIC
(S-TYPE USED).

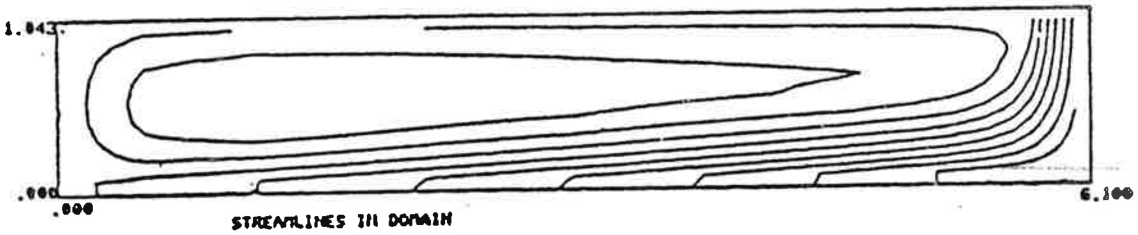


Figure 7 Baseline Test (Run 1)

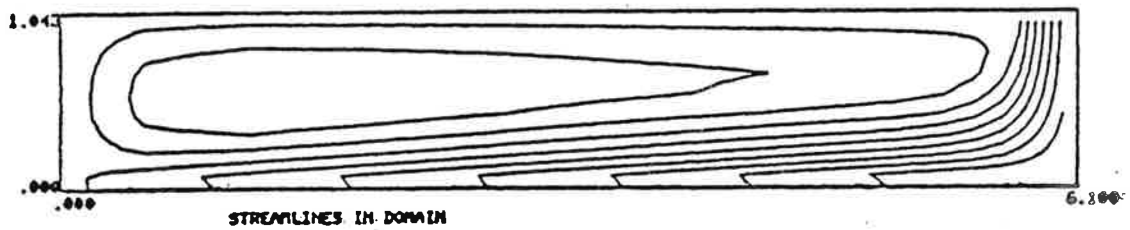


Figure 8 Test Case 2 (Run 2)

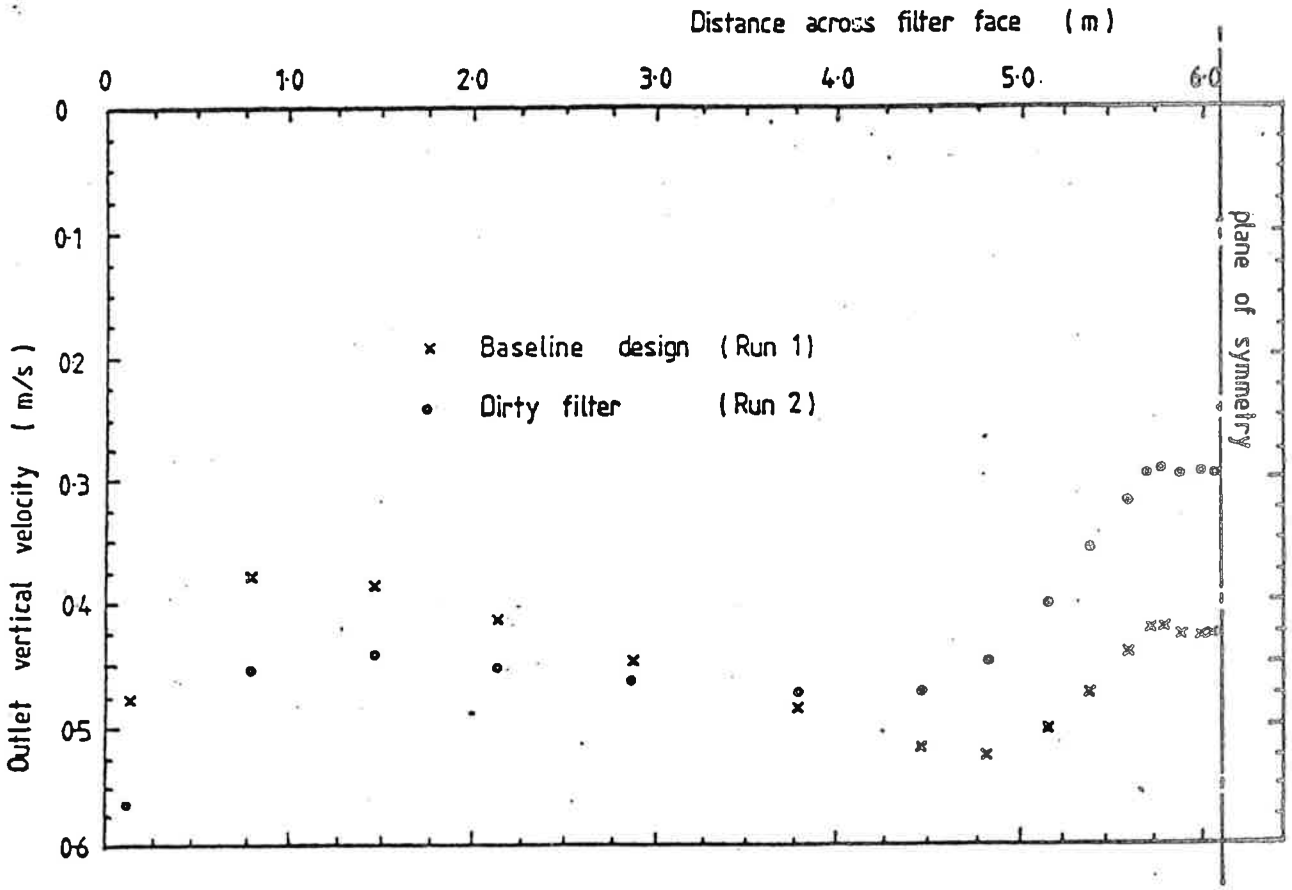
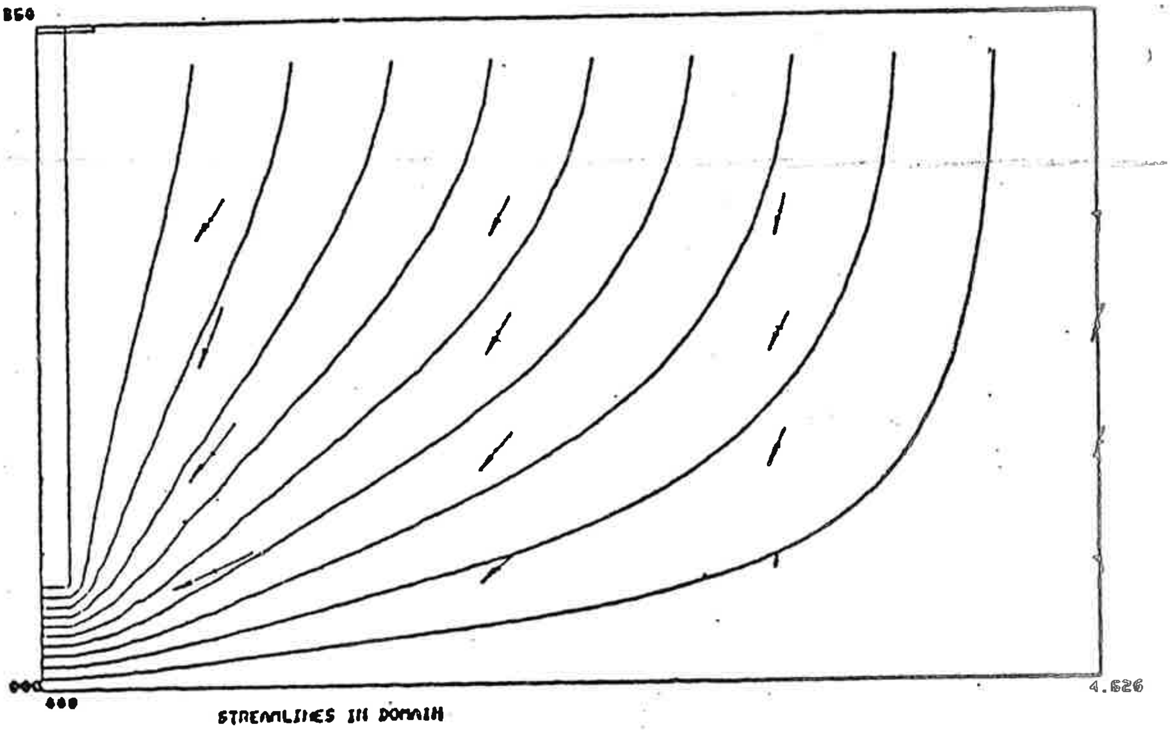


Figure 9 Exit vertical velocity profile for the supply ductwork



—— = 0·5 m/s
velocity vector (exp. data)

FIGURE 10 BASELINE TEST CASE
with experimental results

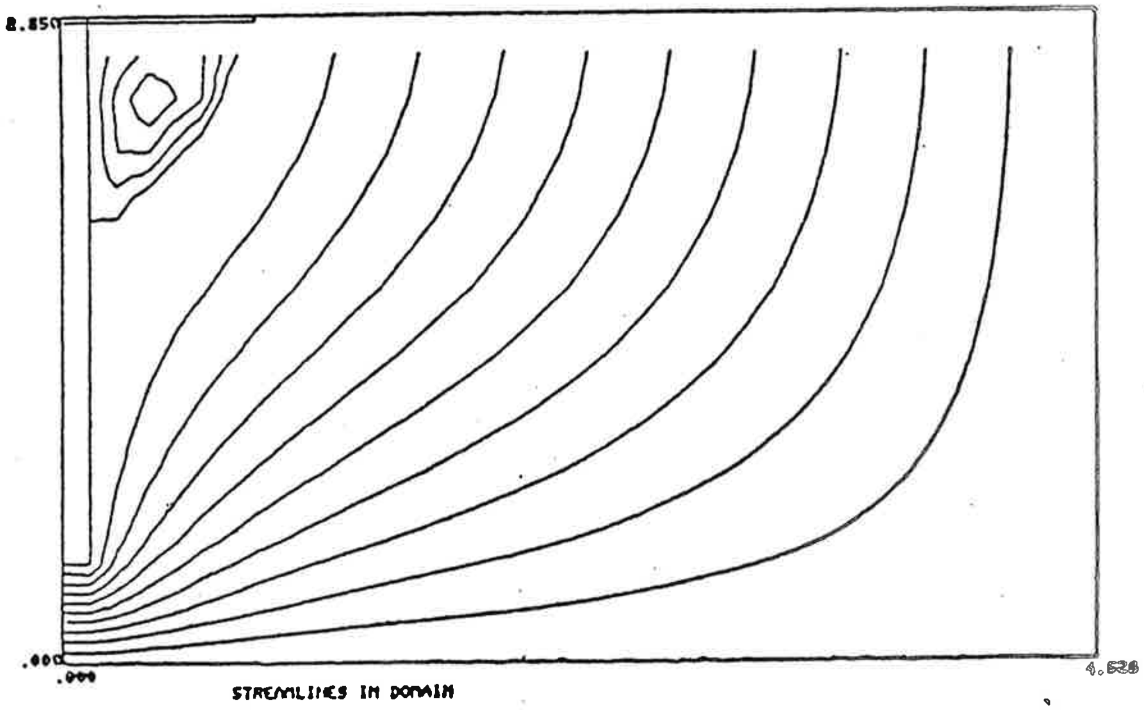


FIGURE 11 RUN 4

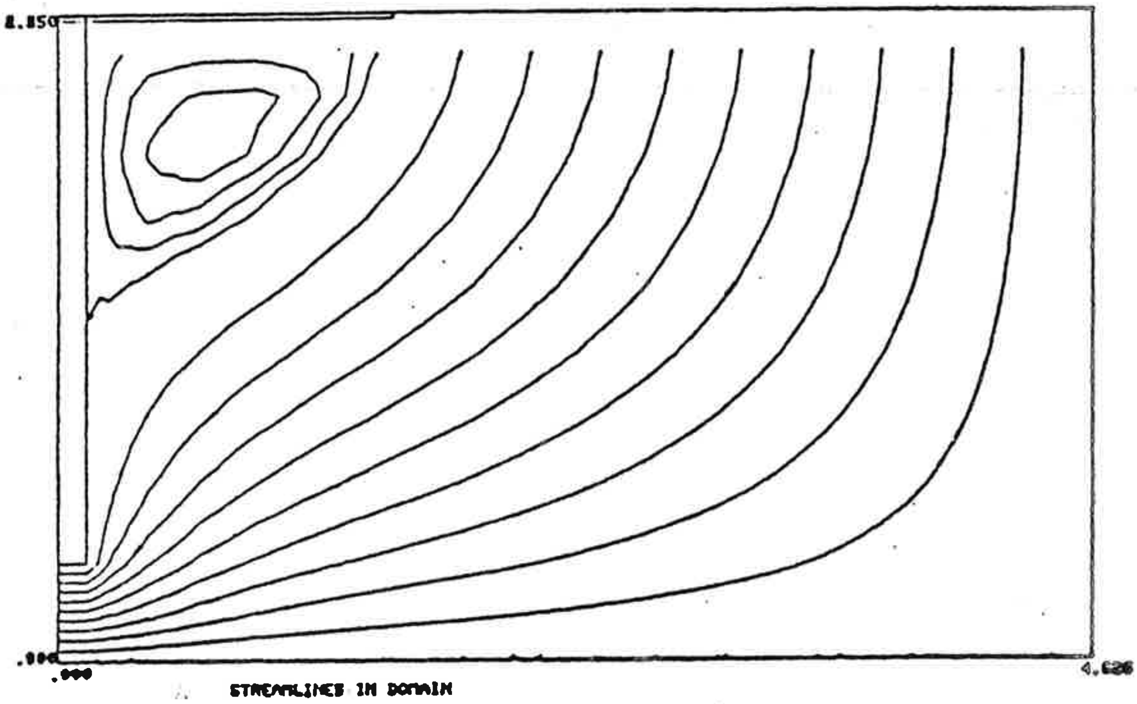


FIGURE 12 RUN 5

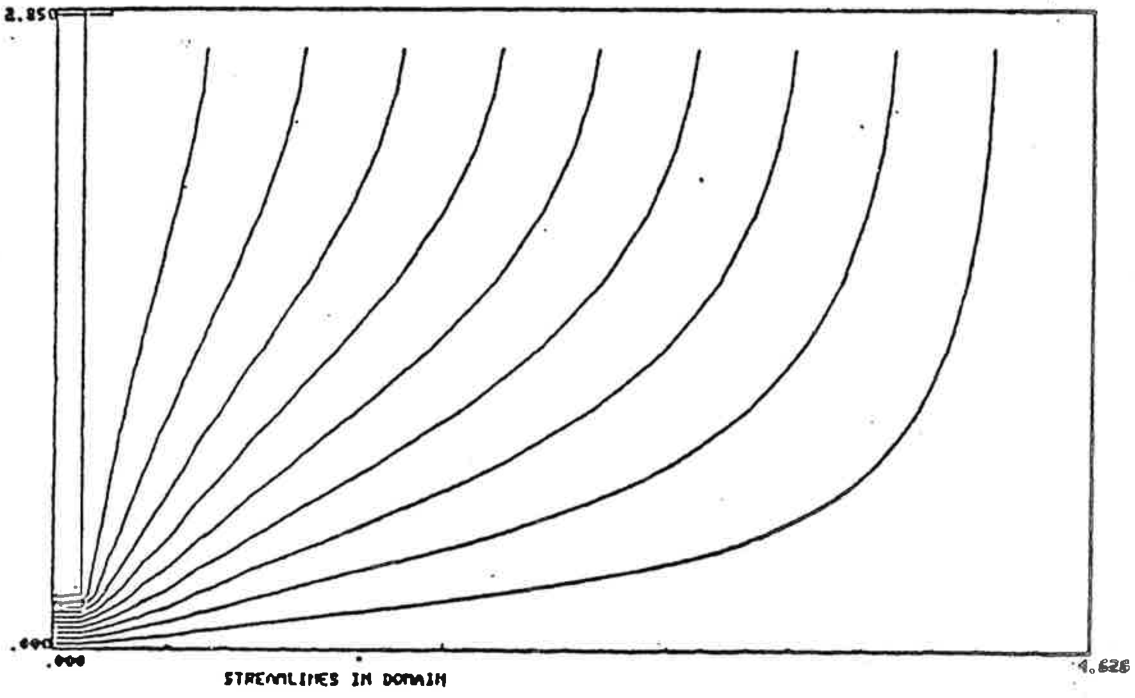


FIGURE 13 RUN 6

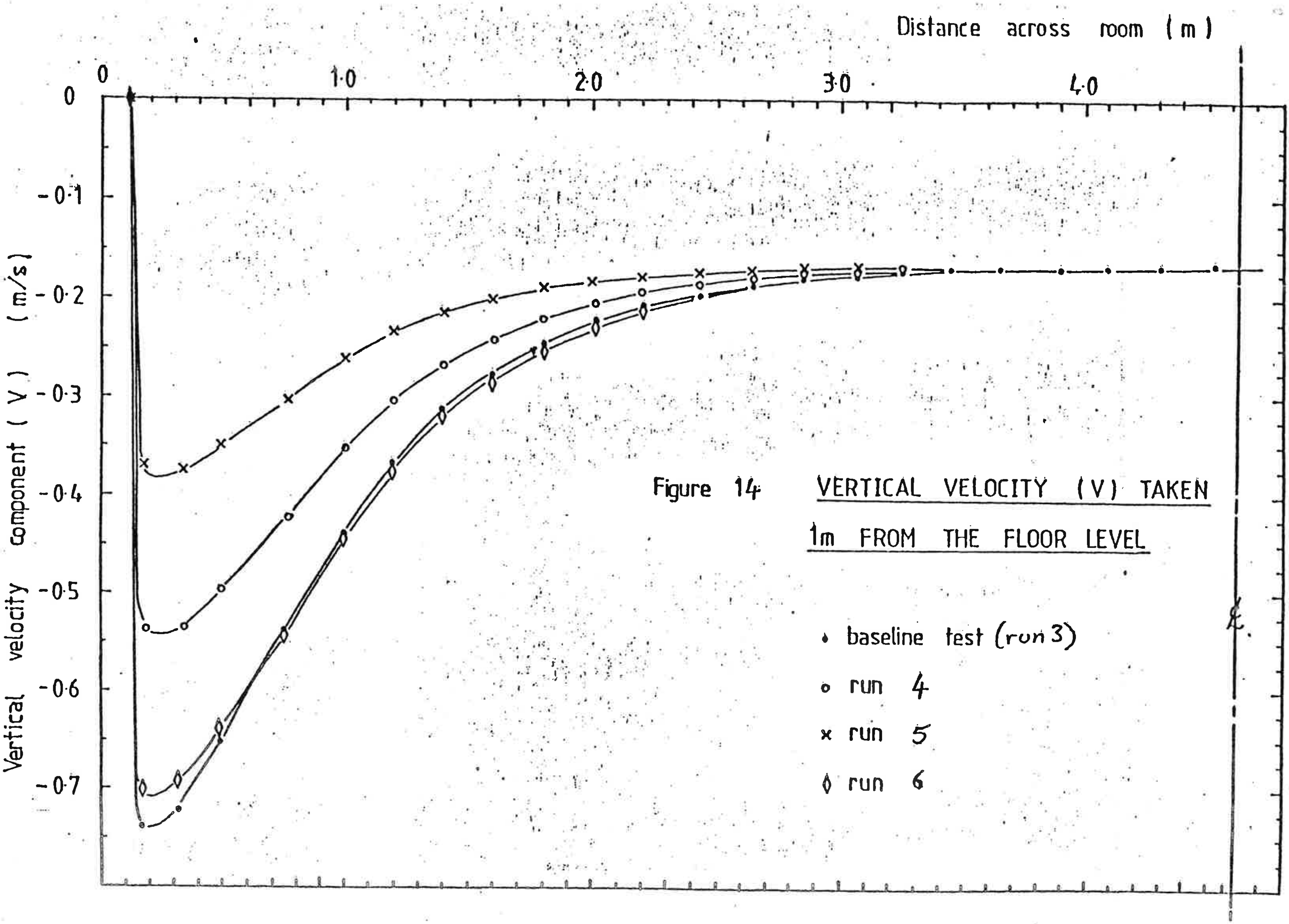


Figure 14 VERTICAL VELOCITY (V) TAKEN 1m FROM THE FLOOR LEVEL

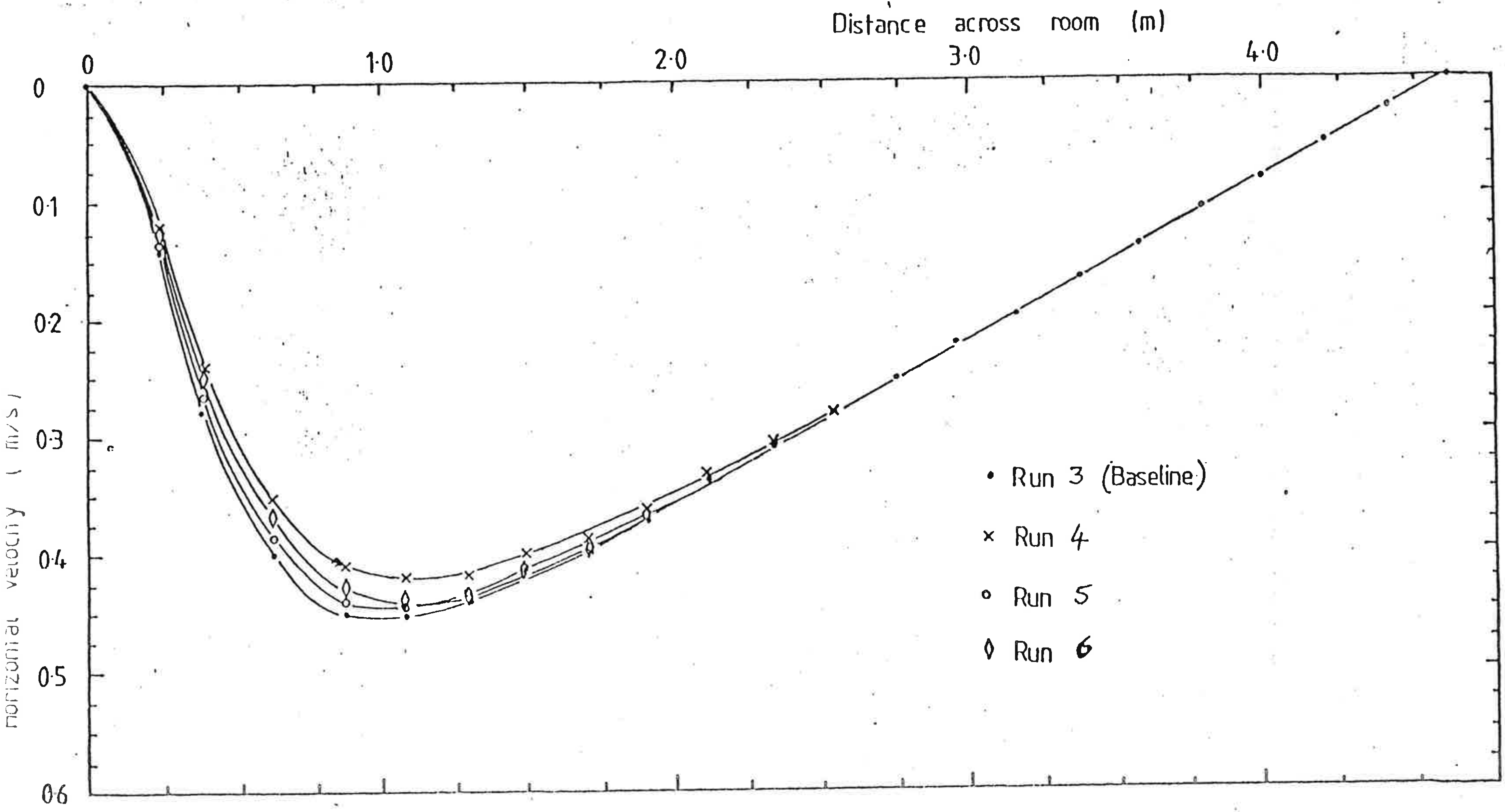


Figure 15 Horizontal velocity taken 1 m from floor level