HR Wallingford

PHYSICAL AND NUMERICAL MODELLING OF AERATORS FOR DAM SPILLWAYS

Technical Report

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ABSTRACT

This report describes experimental and numerical studies on the performance of aerators for protecting chute spillways from cavitation damage. The work was funded by the Construction Industry Directorate of the Department of the Environment and by HR Wallingford.

The experiments were carried out in a 0.3m wide tilting flume which was uprated to allow flow velocities up to 15 m/s. In the first part, a systematic study involving 322 tests was made of the factors affecting the performance of ramp aerators : ramp height ; ramp angle ; flow depth ; flow velocity ; turbulence level ; and head-loss characteristics of the air-supply system. A best-fit formula for predicting the air demand ratio was obtained from an analysis of the test results.

The second part of the experimental study investigated the effect of scale on the amount of air entrained by ramp aerators. Three sizes of geometrically-similar ramp were tested giving scale ratios of 1:1, 1.5:1 and 2:1. The results showed that the air demand ratios scaled correctly provided : (1) the head-loss characteristics of the air supply were reproduced correctly ; and (2) the water velocity in the test exceeded about 5.2 m/s.

In the third part of the experimental work a study was made of alternative types of aerator that would remain efficient while causing less disturbance to high-velocity flows. A new design of three-dimensional wedge was developed which entrained 50% more air than an equivalent length of two-dimensional ramp. The design could be manufactured in steel and would be suitable for installation in both existing and new spillways.

To assist in the design of aeration systems a numerical model called CASCADE was developed. This interfaces with an existing model named SWAN that predicts the development of flow along a spillway channel and also estimates the amount of self-aeration at the free surface. If a risk of cavitation damage is identified, CASCADE is used to determine the size of aerator and supply system needed to provide a specified amount of additional air to the flow. The model offers the option of five different methods of predicting air demand, and also gives an approximate estimate of the spacing necessary between adjacent aerators.

SYMBOLS

A_a	Outlet area of air supply duct
A_m	Cross-sectional area of bend, fitting etc
A_n	Cross-sectional area of duct
a	Non-dimensional pressure parameter - Equation (36)
B	Width of channel
B_a	Combined width of wedge aerators
b	Coefficient in Equation (7)
C_m	Head loss coefficient for bend, fitting etc
c	Coefficient in Equation (10)
d	Depth of flow normal to channel
E	Euler number - Equation (41)
E_{min}	Minimum value of E at start of air entrainment
F	Froude number - Equation (2)
F_*	Critical air entrainment parameter - Equation (8)
g	Acceleration due to gravity
h_1	Height of ramp aerator measured normal to channel
J	Conveyance parameter of air supply system - Equation (3)
K	Cavitation index - Equation (26)
K_i	Value of K for incipient cavitation
k	Entrainment coefficient - Equation (22)
$k_{z,zo}$	Loss coefficients for air jet in cavity - Equation (51)
L_c	Length of air cavity along channel
L_n	Length of duct
M	Overall conveyance parameter of air supply system - Equation (12)
n	Slope of chamfer (n units parallel to flow to one unit normal to flow)
P_n	Flow perimeter of duct
Δp	Relative pressure in air cavity (atmospheric pressure minus pressure in cavity)
$p_{0,1,2}$	Static pressures at points 0, 1, 2
p_v	Vapour pressure of liquid
Q	Volumetric discharge of water
Q_a	Volumetric discharge of air
q	Discharge of water per unit width
q_a	Discharge of air per unit width
T_i	Turbulence intensity - Equation (17)
t	Time
u_1	Height of offset measured normal to channel
V	Depth-averaged water velocity
V_l	Local mean velocity
V_o	Minimum effective value of V for air entrainment
v	Local root-mean-square velocity fluctuation
w	Total height of ramp and/or offset
X_u	Parameter defined by Equation (42)
x	Horizontal co-ordinate from tip of aerator
z	Vertical co-ordinate from tip of aerator; elevation above datum

SYMBOLS (CONT'D)

α	Parameter in Schwartz & Nutt method - Equation (35) ; energy coefficient
β	Air demand ratio (= q_a/q)
β_e	Estimated value of β
β_m	Measured value of β
β_r	Estimated value of β for ramp aerator
γ	Coefficient in Equation (45)
δ	Proportionate change in time-averaged velocity
ϵ	Proportionate fluctuation in instantaneous velocity
$\epsilon_{1,2}$	Coefficients in Equation (11)
θ	Angle of channel to horizontal
λ_n	Darcy-Weisbach friction factor for duct
ν	Kinematic viscosity of liquid
ν_a	Kinematic viscosity of air
ρ	Density of water
ρ_a	Density of air
σ	Surface tension of liquid
ϕ	Angle of ramp relative to channel
ϕ_i	Take-off angle of jet relative to channel
Ω	Turbulence factor - Equations (20) and (21)

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1 INTRODUCTION

Aerators are now often used to prevent cavitation damage in spillways and tunnels of high-head dams. Cavitation bubbles form within a liquid when the local static pressure falls close to the vapour pressure of the liquid (which for water is usually only slightly greater than zero pressure absolute). In civil engineering applications, this occurs most commonly in high-velocity flows and is the result of turbulence and flow separation at discontinuities in the boundaries. Damage to the perimeter of a channel is caused not by the formation of the cavitation bubbles but by their violent collapse when they enter regions of higher pressure. Damage therefore tends to be located downstream of the point at which the bubbles are generated.

Injection of air into water has been found to cushion the collapse of cavitation bubbles and, in sufficient concentration, to prevent damage ; note that the air does not itself prevent the formation of the bubbles. A spillway aerator consists of a ramp or offset (or combination of the two) that operates by creating a large air cavity within the water from which air can be entrained into the flow. The pressure within the air cavity is slightly below atmospheric pressure so that the air can be drawn in naturally via a system of ducts or wall slots without the need for injection pumps. It is important to distinguish between the cavitation bubbles, which are usually very small and filled mainly with water vapour at a pressure close to absolute zero, and the air cavity formed by the aerator, which is large and filled with air at a pressure only slightly less than atmospheric.

Research on aerators at HR Wallingford was started under a previous contract funded by the Construction

Industry Directorate of the Department of the Environment (DOE). The first stage consisted of a literature review on the general subject of cavitation and aeration in hydraulic structures ; the results were published in the form of a design manual in HR Report SR79 (see May (1987)). In the second stage, a high-velocity flume was specially built for the testing of spillway aerators. A systematic study of aeration ramps was carried out to investigate the relationship between air demand and the geometry of the ramp, the characteristics of the air-supply system and the flow conditions in the spillway channel. Results of these experiments were presented in HR Report SR 198 (see May and Deamer (1989)).

The present report describes two parts of a follow-on research study which was also funded by DOE. The first part was carried out using the high-velocity flume and extended the previous experimental work on aerators. Two aspects were studied : scale effects in model tests of ramp aerators and alternative configurations for new designs of aerator. The second part involved the development of a numerical model called CASCADE for designing aeration systems for spillways. The model uses as its starting point a computer program for self-aerated flows produced by Binnie and Partners. The model identifies the point on a spillway where cavitation damage becomes a danger, and enables the geometry of the required aerators and air-supply system to be determined.

Part A of this report describes and analyses the experimental results obtained on scale effects and new designs of aerator. Part B gives details of the methods used in the numerical model to predict the performance of aerators and contains a manual explaining the data requirements and how to use the program.

PART A - EXPERIMENTAL STUDIES

2 SCOPE OF TESTS

Model studies of spillway aerators are usually carried out with the same fluids (air and water) as in the prototype. Effects due to viscosity and surface tension therefore tend to be relatively too high in a model, and can cause it to underestimate the air demand that will occur in the prototype. Comparison of data from laboratory studies (see May (1987)) indicates that for a model of a spillway aerator the scale needs to be greater than approximately 1:15 if significant scale effects are to be avoided. A second requirement is that the velocity of the water in the model should be high enough to reproduce the air entrainment process correctly ; experiments suggest a minimum value of about 6-7m/s.

The tests on ramp aerators carried out at HR in the previous DOE contract (see May and Deamer (1989)) were limited to a maximum velocity of 6.8m/s by leakage problems in the flume. These were overcome at the start of the present contract by the construction of a new steel pressure box which enabled the flume to be operated at velocities up to 15m/s. The first stage of the present experimental work therefore extended the previous tests on ramp aerators to higher velocities and determined the extent of scale effects by means of comparative tests of different sizes of aerators having the same geometrical shape.

The second stage of the experiments investigated new designs of spillway aerator. Existing types appear to work effectively in preventing cavitation damage, but suffer from three main problems. Firstly, the ramps and offsets cause very major disturbances to the flow. The aim in a conventional dam spillway is usually to produce the smoothest possible flow profile. The provision of aerators creates a large amount of turbulence and bulking of the flow. The depth of the channel and the amount of freeboard therefore need to be increased, which can be expensive in terms of excavation and construction. Excessive spray from spillways is also undesirable and can cause erosion and slips in adjacent embankments. Secondly, aerators for wide spillways often require large ventilation shafts and ducts beneath the channel to supply air across the full width of the flow. These are costly and cannot easily be added as a remedial measure to an existing spillway which has suffered cavitation damage. Thirdly, large aerators are not a very efficient solution to the problem. A prototype study by Pinto (1986) of the spillway for the Foz do Areia dam showed that the turbulence created by the aerators entrained approximately three times as much air through the free surface as was entrained directly by the aerators. To prevent cavitation damage to the boundaries of a channel, the local air concentration needs to be above a certain minimum figure (often assumed to be about 7%, see May (1987)). The entrained air in the rest of the flow is not therefore effective as it tends to move upwards and away from the boundaries.

These factors suggest that a more efficient solution would involve the use of smaller aerators spaced more frequently along the spillway. These would create less disturbance to the flow and inject air close to the boundaries where it is required. The greater efficiency of the system would reduce the total amount of air added and thus cause less bulking of the flow. These benefits would allow the depth of the spillway channel to be reduced compared with that needed for a conventional aeration system. Smaller aerators could be prefabricated and would have the advantage of being easier to install on existing spillways where remedial measures are required. Preliminary experiments in the high-velocity flume were therefore carried out at a small scale on several alternative geometries. The selected design was then built to a larger scale and its performance compared with some of the conventional ramp aerators tested previously.

3 EXPERIMENTAL ARRANGEMENT

The experiments were carried out in the high-velocity flume constructed as part of the previous DOE contract. The layout of the flume is shown in Figure 1 and Plates 1 and 2 and its principal dimensions are:

length of test section	: 4.0m
width (variable)	: 0.30m (maximum)
depth	: 0.43m
angle of flume	: 0° to 45°
maximum discharge	: 0.21m³/s
maximum velocity	: ~ 16m/s

Full details of the design are given in May & Deamer (1989).

As mentioned in Section 2, the wooden pressure box used in the original design was replaced during this contract by a steel box of welded construction (see Fig 1). The depth of water at the upstream end of the flume was determined by a wedge block bolted to the roof of the pressure box. The thickness of this block was varied by adding or removing pvc spacers. The new design worked very satisfactorily and allowed the 20m head pump to be operated at full capacity.

Additional instrumentation was used in the new tests to measure the degree of turbulence in the flow approaching the aerator and the length of the air cavity downstream of the aerator. Other experimental studies have shown that increasing the turbulence can increase the efficiency of the entrainment process by which air is removed from the cavity. The turbulence level was therefore measured using a total-head pitot tube of the type originally developed by Arndt & Ippen (1970). The HR version consisted of a 2.0mm internal diameter pitot tube connected via an adapter to a flush-faced pressure transducer that registered the instantaneous fluctuations in velocity head. The tube and transducer were filled with water and sealed under a vacuum so as to prevent air bubbles in the flow entering the tube and causing errors in the measurement of the dynamic pressure. The output from the transducer was analysed to determine the mean and root-mean-square values of the pressure fluctuations.

Although the flume was equipped with perspex sides, it was not possible to determine reliably by visual observation where the flow re-attached downstream of the aerator. The length of the air cavity was therefore measured using an instrument specially developed for the purpose. The device was basically a conductivity probe consisting of two insulated wires with their exposed tips about 1mm apart. This was moved along near the floor of the flume until the large change in electrical reading occurring at the boundary of the cavity was detected.

As in the previous contract, a void-fraction meter was available for making point measurements of air concentration downstream of the aerators. The instrument uses a probe with a small insulated wire tip, and acts effectively as an on/off switch when the tip passes from water to air. More details of the meter are given in White & Hay (1975) and May & Deamer (1989).

4 TESTS ON AERATION RAMPS

The twin aims of the tests were to measure the air demand of ramp aerators at higher velocities than used in the previous study and to investigate how dependent the results were on the scale of the model.

Six different aerators of triangular cross-section were tested in the high-velocity flume. Each aerator spanned the full 0.30m width of the channel, with air being drawn in along the downstream edge of the ramp from a box beneath the flume. The rate of flow of air was measured using a Dall tube (truncated venturi) in the inlet pipe to the air box and was adjusted by means of a butterfly valve (see Fig 2).

Two shapes of ramp were tested, each in three different sizes, as follows:

Ramp No	Height, h_1 (mm)	Ramp slope, ϕ (degrees)
1	8	9.09
3	8	4.57
4	12	9.09
6	12	4.57
7	16	9.09
9	16	4.57

The symbols used in this report are defined at the beginning of the text. The following range of conditions was studied in the tests:

• mean flow velocity, V	1.7-14.3m/s
• flow depth, d	46-106mm
• Froude number, F	2.5-21.5
• relative flow depth, d/h_1	2.9-14.3
• angle of channel to horizontal, Θ	15.5°
• head loss characteristics of air supply system	5 No valve settings (0,1,2,3,4 with setting 0 being fully open)

The following quantities were measured in all the tests carried out :

- water discharge per unit width, q
- air flow rate per unit width, q_a
- pressure in air cavity, Δp
- temperatures of air and water
- air pressure (mm Hg)

The pressure in the air cavity produced by the aerator was measured in terms of mm head of water relative to atmosphere ; a positive value of Δp indicates that the pressure in the cavity was below atmospheric. The temperature and pressure of the air were needed to calculate the density of the air and its rate of flow through the Dall tube.

In certain of the tests, the following additional measurements were made:

- length of air cavity downstream of aerator, L_c

- local values of the root-mean-square velocity fluctuation v'

If scale effects due to viscosity and surface tension are not present, the non-dimensional air demand ratio $\beta = q_a/q$ should depend only on the following factors

$$\beta = fn_1 \left(F, \frac{d}{h_1}, \phi, \Theta, \frac{\Delta p}{\rho g d}, \frac{\rho_a}{\rho}, \frac{v'}{V}\right) \qquad (1)$$

where ρ and ρ_a are the densities of water and air respectively, and the reduced Froude number F is defined as

$$F = \frac{V}{(gd)^{1/2}} \qquad (2)$$

Note that, for convenience, the channel slope term is omitted from the precise definition of the Froude number because Θ needs to be included anyway as a separate variable in Equation (1). For a given air demand, the sub-atmospheric pressure in the air cavity is determined by the head-loss characteristics of the supply system. To a reasonable order of accuracy, these characteristics can be described by the equation

$$q_a = \frac{J A_a}{B} \left(\frac{\Delta p}{\rho_a}\right)^{1/2} \qquad (3)$$

where A_a is the cross-sectional outlet area of the ducts supplying air to width B of the channel. J is a conveyance parameter of the system, which is inversely related to the sum of the head loss coefficients associated with friction, bends, entrance, exit etc.

Equation (3) can be used to express the parameter $(\Delta p/\rho g d)$ from Equation (1) in the form

$$\frac{\Delta p}{\rho g d} = (\rho_a/\rho)\, \beta^2 F^2 \,(d/h_1)^2 \left(\frac{B h_1}{J A_a}\right)^2 \qquad (4)$$

This enables the functional relationship (1) to be expressed more conveniently as

$$\beta = fn_2 \left(F , \frac{d}{h_1} , \emptyset , \Theta , \frac{JA_a}{Bh_1} , \frac{\rho_a}{\rho} , \frac{v'}{V} \right) \qquad (5)$$

The advantage of this formulation is that the air demand ratio now depends only on the external flow conditions and on the geometry of the aerator and its air supply system. In the present study, the air supply arrangement was such that

$$A_a = 0.956 \ Bh_1 \qquad (6)$$

The extent of possible scale effects can therefore be investigated by comparing curves of the air demand ratio β against the Froude number F for tests with different sizes of geometrically-similar ramp and equal values of the flow depth ratio d/h_1, the head loss parameter J and the turbulence factor v'/V.

Results obtained from a main group of 226 tests carried out with the six different ramp aerators are listed in Table 1. The tests within this group that were intended specifically to investigate scale effects are identified separately in Table 2. Data from a second group of 96 tests, in which the turbulence intensity was varied to study its effect on the cavity length and the air demand, are listed in Table 3. The turbulence level was increased by means of adding mesh to the floor of the flume upstream of the aerator. The results from all these tests are analysed in the next section.

5 ANALYSIS OF TEST DATA

The air demand data from the main tests (Table 1) are plotted in dimensional form in Figures 3 to 20. Each Figure shows how the air demand per unit width of channel (q_a) varied with flow velocity (V) and water depth (d) for a given aerator and setting of the air valve in the supply system. The same data are plotted in dimensionless form in Figures 21 to 38 using the air demand ratio (β) and the Froude number (F).

Although the data show a fair degree of scatter, several conclusions can be drawn from a study of the dimensional plots in Figures 3 to 20.

(1) An approximately linear relationship exists between q_a and V for a given aerator, valve setting and water depth d.

(2) The effect of water depth on air demand does not follow a consistent pattern (compare for example Figures 13 and 17), but overall it is relatively small.

(3) Extrapolating the linear relationship between q_a and V indicates that for each aerator there is a "minimum" velocity V_o below which air entrainment does not occur.

(4) The value of V_o increases as the height (h_1) of the aerator increases but shows little dependence on the ramp angle ϕ.

These observation suggest that

$$q_a = b\ (V-V_o) \qquad (7)$$

where overall the coefficient b is effectively independent of the flow depth d. The minimum velocity V_o may not exactly be the velocity at which air entrainment begins but is the value obtained by extrapolating the test data back to $q_a = 0$, assuming a linear relationship between q_a and V. Two factors can be expected to influence V_o. Firstly, the velocity of the water needs to be high enough for disturbances along the surface of the lower nappe to grow and entrain air into the body of the flow. Secondly, the curvature of the flow produced by the aerator has to be sufficient to overcome the hydrostatic pressure and produce a pressure in the air cavity that is below atmospheric ; unless this happens, air will not be drawn through the supply system and the cavity will fill with water. Inspection of the plots in Figures 3 to 20 indicates that V_o varies from approximately 2m/s for the two 8mm high ramps to approximately 3 m/s for the two 16mm high ramps. This suggests a "Froudian-type" relationship between V_o and h_1, ie

$$\frac{V_o}{\sqrt{(gh_1)}} = F_* \approx 7.5 \qquad (8)$$

The value of 7.5 is only approximate due to the variable amount of scatter in the plots, but is an adequate basis for investigating the dependence of the coefficient b on other factors. Values of b were calculated from

$$b = \frac{q_a}{[V - F_*(gh_1)^{1/2}]} \qquad (9)$$

using $F_* = 7.5$, and it was found that, for equivalent conditions, the 16mm high aerators had values approximately double those of the 8mm high aerators. Keeping h_1 constant but varying the ramp angle ϕ

from 4.6° to 9.1° did not appear to have any consistent effect on the value of b. This suggested that Equation (9) could be written in non-dimensional form as

$$c = \frac{q_a}{[V - F_*(gh_1)^{½}]\ h_1} \qquad (10)$$

Values of c for all the main tests are listed in Table 1. Having identified a "model" equation that describes the main trends of the data, it is necessary to investigate whether there are any consistent variations in the values of the coefficient c. The functional relationship in Equation (5) suggests that the head-loss parameter (JA_a/Bh_1) of the air supply system could be significant. Figure 39 therefore shows how c varies with this parameter for all the conditions studied in the main tests (Table 1). Each point in the Figure represents the best fit value of c for all the velocities and water depths tested with a given aerator and air valve setting. It can be seen that there is a reasonably well-defined trend with c increasing as (JA_a/Bh_1) increases. Variations in the height of the aerator and the angle of the ramp do not appear to produce any consistent variation in the value of c. The introduction of an additional parameter from Equation (5) such as F, d/h_1, or ϕ is therefore likely to worsen rather than improve the correlation for c. The data in Figure 39 indicate a straight-line relationship through the origin at low values of (JA_a/Bh_1) but with a tendency towards a convex shape at higher values of c.

The parameter J has a maximum theoretical value of $\sqrt{2}$ (see Appendix A), so the corresponding maximum value of (JA_a/Bh_1) in the present tests was about 1.35 (see Eqn (6)). For a given rate of air flow, the pressure Δp in the cavity becomes smaller as J increases, and below a certain limit it can be expected to have no influence on the entrainment process or on the behaviour of the flow passing over the cavity. Thus, it is likely that c will become asymptotic to a maximum value when (JA_a/Bh_1) becomes large. At the opposite end of the range, a small value of J will give rise to a large pressure difference that will reduce the length of the air cavity and also the amount of air entrainment. At the limit, it can be expected that $c \to 0$ as $J \to 0$.

The data plotted in Figure 39 therefore demonstrate the expected type of variation in c, and a suitable form of equation with the required properties is

$$c = \epsilon_1 \; [1 - \exp \; (- \epsilon_2 \; M)] \tag{11}$$

where ϵ_1 and ϵ_2 are coefficients to be determined from the data, and

$$M = \frac{JA_a}{Bh_1} \tag{12}$$

As will be discussed shortly, it was decided to omit from the final correlation all the tests in which the flow velocity was less than 5.2m/s. The best-fit coefficients were therefore determined using the reduced set of data in Table 1 plus the tests in Table 3 that are marked A ; the additional tests were carried out with the same turbulence level as those in Table 1. The values of the coefficients obtained from 180 separate tests were

$$\epsilon_1 = 1.3 \tag{13a}$$

$$\epsilon_2 = 2.5 \tag{13b}$$

The resulting equation for estimating the air demands measured in the present study is thus

$$q_a = 1.3\ [1 - \exp(-2.5M)]\ [V - F_*(gh_1)^{\frac{1}{2}}]\ h_1 \tag{14}$$

This can be expressed in terms of the non-dimensional air demand ratio $\beta = q_a/q$ as

$$\beta = 1.3\ [1 - \exp(-2.5M)]\ [1 - \frac{F_*}{F}\ (\frac{h_1}{d})^{\frac{1}{2}}\ ](\frac{h_1}{d}) \tag{15}$$

where $F_* = 7.5$. This results shows that β initially increases as the Froude number F of the flow increases but tends towards a constant value when F becomes large. This behaviour is consistent with the plots of the experimental data shown in Figures 21 to 38. Typically the value of β becomes constant when the Froude number exceeds about F = 8 to 12, depending upon the particular tests conditions. This suggests that the efficiency of the air entrainment process does not change at high flow velocities.

The angle ϕ of the aerator ramp does not appear in Equation (15) because, as explained above, no consistent differences in the air demand were identified between aerators of the same height with angles of either $\phi = 4.6°$ or $9.1°$. This finding is perhaps unexpected but is supported by the form of the following equation due to Bruschin (1985)

$$\beta = 0.0334\ F\ (w/d)^{1/2} \qquad (16)$$

in which w is the overall height of the ramp and/or offset, but in which the ramp angle does not appear.

It should be stressed that Equation (15) applies to the spillway slope of $\theta = 15.5°$ used in the present study, and that the numerical coefficient 1.3 (and perhaps other terms in the equation) can be expected to vary for other spillway slopes. The degree of agreement between the measured values of the air demand ratio (β_m) and the estimated values (β_e) given by Equation (15) is illustrated by Figure 40 which shows points for all the tests used in the analysis. The average value of the ratio β_e/β_m is 1.022 with a standard deviation of 21.7%.

The possible effect of model scale on the amount of air entrainment can be studied using the data in Table 2 which are a sub-set of those in Table 1. The values of β versus F are plotted in Figures 41 and 42 for the aerators with ramp angles of $\phi = 9.1°$ and 4.6° respectively. In each case the 1:1 model refers to an 8mm high aerator, the 1.5:1 model to a 12mm high aerator and the 2:1 model to a 16mm high aerator.

At first glance the plots in Figures 41 and 42 appear to suggest that significant scale effects exist between geometrically-similar aerators of different size. However, although the tests were carried out with similar values of F, d/h_1 and A_a/Bh_1, it was not possible to obtain equal values of the head-loss parameter J at the three different scales (because the air supply system was not be scaled to match). Therefore, it is first necessary to take account of the effect on air demand produced by these variations in J. This can be done by comparing the measured air demands (β_m) with the predicted values (β_e) given by

Equation (15), which is the best-fit to all the data. If a scale effect exists, there should be a systematic difference between the data for one size of aerator and another.

Values of the ratio β_e/β_m are listed in Table 2 and plotted in Figures 43 and 44. It can be seen that the values are generally consistent for all three sizes of aerator. However, in those tests where the flow velocity was less than about 5.2m/s, the values of β_e/β_m are nearly always greater than unity ; this shows that the actual air demands were significantly lower than expected. The conclusion therefore is that the scale of the model is not a factor provided the water velocity is high enough to produce "fully-developed" entrainment. The data obtained in the present study indicate that this limiting velocity is about 5.2m/s. As mentioned previously, tests with $V < 5.2$m/s were omitted from the final data analysis leading to Equation (15) so as to produce a prediction formula free from identified scale effects.

The effect on air demand of varying the degree of turbulence in the flow approaching an aerator can be evaluated from a study of the additional data in Table 3. Comparative tests for similar sets of flow conditions were carried out with three different degrees of turbulence:

(1) normal turbulence as occurring "naturally" in the test rig (equivalent to conditions for main tests in Table 1);

(2) medium turbulence produced by fixing a layer of mesh to the floor of the flume upstream of the aerator;

(3) <u>higher</u> turbulence produced by adding a coarser layer of mesh upstream of the aerator.

These three turbulence levels are labelled A, B and C respectively in Table 3. Values of the turbulence intensity T_i were measured using the velocity probe described in Section 3. T_i is defined here as

$$T_i = \frac{v'}{V_l} \qquad (17)$$

where v' is the root-mean-square velocity and V_l is the local mean velocity, both measured at about the mid-depth of the flow upstream of the aerator. The terms normal, medium and higher turbulence given above are relative and used only to identify the three different test conditions. The measured values of T_i given in Table 3 for normal turbulence (Type A) are generally in the range 3% to 7%, and agree with results from other studies of naturally-occurring turbulent flows (eg pipes and open channels). Values of T_i for the medium turbulence level (Type B) were about 7% to 12%, and for the higher turbulence level (Type C) they were typically in the range 12% to 20%.

The effect of increasing the turbulence of the flow is illustrated by Figures 45 to 48 which show plots of β versus F for aerators 1, 3, 7 and 9 respectively. At Froude numbers below 10, the air demand of the 8mm high aerators (nos 1 and 3) is generally greater with "medium" and "higher" turbulence than it is with "normal" turbulence. However, when $F > 10$, the air demand is less affected by the turbulence level; the same also applies to the 16mm high aerators (nos 7 and 9). In order to identify the influence of turbulence more precisely, it is necessary to take account of the variations in the head-loss factor J that occurred between the tests in Table 3. This can be done by comparing the measured air demand ratios (β_m) with the estimated values (β_e) given by Equation (15). The results are plotted in Figure 49, and it can be seen that the data for the tests with the "normal" Type A turbulence are distributed fairly symmetrically about the β_e/β_m = 1.0 line. The points for the Type B turbulence are generally a little below the line and those for Type C are further below ; this indicates that increasing the turbulence level tends to produce an increase in the amount of air entrainment.

Referring to Equation (15), it might be expected that higher turbulence would increase the overall numerical coefficient (ie the 1.3 factor) and possibly reduce the value of F_* for the inception velocity below 7.5. There are not enough additional data points for the Type B and C conditions to establish whether F_* is in fact reduced, but the best-fit straight line shown in Figure 49 provides an approximate estimate of the effect of turbulence. It is therefore recommended that Equation (15) should be used in the following modified form:

$$\beta = \left(\frac{1.3}{\Omega}\right) [1 - \exp(-2.5M)] \left[1 - \frac{F_*}{F} \left(\frac{h_1}{d}\right)^{1/2}\right] \left(\frac{h_1}{d}\right) \tag{18}$$

where

$$F_* = 7.5 \tag{19}$$

$$\Omega = 1.0 \quad , \text{ for } T_i \le 0.05 \tag{20}$$

$$\Omega = 1.0 - 2.0 (T_i - 0.05) \quad , \text{ for } T_i > 0.05 \tag{21}$$

"Normal" turbulence in the flume corresponded to a turbulence intensity of about 5%.

Although additional turbulence does tend to increase the amount of air entrainment, the effect is not as marked as perhaps might have been expected. An explanation of this finding is provided by the data on cavity lengths given in Table 3 and plotted in Figures 50 to 53. It can be seen in all cases that increasing the turbulence intensity reduces significantly the length of the air cavity. Thus, although greater turbulence increases the amount of entrainment at the air-water interface, it also causes the water jet to break up more quickly, thus reducing the length of the air cavity. These two opposing effects probably explain why the air demand values in Figure 49 show relatively little variation with changes in the turbulence level.

Several studies of aerators have suggested that the rate of air demand is directly related to the flow velocity and the length of the air cavity, ie

$$q_a = k \, V \, L_c \qquad (22)$$

where k is a coefficient which is approximately constant for a given arrangement of aerator. Values of q_a and (VL_c) from Table 3 are plotted in Figure 54. The data for all four aerators follow a similar pattern, and show that k in Equation (22) can only be considered constant at relatively low values of (VL_c). This result throws doubt on some existing design methods which estimate the air demand by using predicted values of cavity length and assumed constant values of k.

6 TESTS OF NEW AERATOR DESIGNS

As explained in Section 2, the purpose of these tests was to identify a suitable design of aerator that would entrain air efficiently where it is needed at the boundaries of a channel but also cause less disturbance to the flow and be easier to install as a remedial measure to existing spillways.

The first stage involved a preliminary assessment of the flow characteristics of different shapes of aerator using small-scale models. In the second stage, the most promising design was built at a larger scale and its performance measured and compared with that of the equivalent ramp aerator. The selected design was also further developed so as to allow the addition of its own air supply duct.

The first design that was studied was a vertical cylinder extending through the full depth of the flow and consisting of a circular tube with a streamlined fairing around its upstream edge. Flow separation on the downstream side of the tube produced a cavity to which air was supplied through an orifice at the base of the pier. The advantage foreseen for this arrangement was that the hollow pier could act as a ventilation shaft supplying air to the orifice at its base. This would avoid the need for separate air ducts which can be costly and difficult to install in existing spillways. Two sizes of pier were tested (diameters of 16mm and 27mm) but it was found that both types caused an unacceptable amount of disturbance to the flow. Water piled up considerably at the leading edge of the pier and also in the wake at the downstream end of the air cavity. Further development of this design was not therefore pursued.

Test were next carried out on small three-dimensional wedge-shaped aerators of the two types shown in Fig 55. Hay & White (1975) had earlier tested aerators consisting of shallow tear-shaped deflectors upstream of semi-circular notches in the spillway surface, but no subsequent studies appear to have been carried out. Since such aerators can be spaced laterally and longitudinally within a spillway, they produce less intense point disturbances than full-width ramps.

Initial tests were made with five aerators of each type equally-spaced in a single line across the flume (see Fig 55 and Plates 3 and 4). Air was provided through circular holes in the flume floor that connected to the existing air supply system. The amount of entrainment was too small to be measured accurately by the Dall tube so the performance of the two designs was compared in terms of the water surface profile and local air concentrations downstream of the aerators. Tests were made at four different flow rates and two different water depths for a fixed flume slope of $\theta = 15.5^{\circ}$ and with the butterfly valve in the air supply system fully open (position 0).

Comparison of the data showed that the bluffer of the two designs (Type 11) was more effective at entraining air but without causing too much disruption of the flow (see Plates 5 and 6). It was therefore decided to test two aerators of this shape at a larger scale (Type 12 in Fig 56); the height of the wedge was 16mm and therefore 2.67 times that tested initially (see Plate 7). Air was supplied by a rectangular slot in the flume floor downstream of each wedge; the width of the slot was made equal to the height of the wedge (i.e 16mm) so as to provide the same relative area of opening as was used in the earlier tests with the ramp aerators (see Section 4). Tests were carried out for similar flow conditions to those used for the smaller wedges (see Plate 8). In this case, however, the air demand was large enough to be measured accurately by the Dall tube. Results of the tests are listed in Table 4.

The selected design of wedge-shaped aerator was further developed by the addition of a small ramp between the wedges (Part B of Aerator Type 13 in Fig 56). The purpose of this ramp was to provide a means of supplying air to the individual aerators without the need for large ducts beneath the spillway. It was also foreseen that the small ramp might improve the efficiency of the main wedge by increasing the amount of entrainment along the sides of the air cavity. Results of the tests with the Type 13 design are given in Table 4.

7 PERFORMANCE OF WEDGE-SHAPED AERATORS

The Type 12 and 13 wedge-shaped aerators are both 16mm high and have a ramp angle of $\phi = 6.8°$; they are therefore intermediate between the Type 7 ($\phi = 9.1°$) and Type 9 ($\phi = 4.6°$) two-dimensional ramps. In Table 4 the values of the air demand ratio β are calculated from

$$\beta = \frac{Q_a}{B_a q} \tag{23}$$

where Q_a is the total rate of air flow to the two wedges and B_a is the combined width of the wedges (= 2x0.080 = 0.160m out of a total flume width of 0.300m). Similarly, the conveyance parameter M of the air supply system is calculated from

$$M = \frac{JA_a}{B_a h_1} \tag{24}$$

where A_a is the combined outlet area of the ducts for the two wedges. Thus β and M are the values per unit width of wedge, and can therefore be compared with the earlier results for the two-dimensional ramps.

The variations of β with F for the Type 12 and 13 wedges are compared with those for the Type 9 ramp in Figure 57 (air valve setting 0). It can be seen that the points for the plain Type 12 wedges are generally close to but above those for the Type 9 ramp. Some improvement in performance is to be expected because the wedges produce three-dimensional cavities with larger surface areas than equivalent two-dimensional ramps. Figure 57 also shows that the addition of the smaller ramp (Part B in Fig 56) considerably increases the amount of air entrained by the wedges.

As explained previously, variations in the J parameter between one test and another need to be taken into account when comparing alternative designs on a quantitative basis. Table 4 therefore gives values of the ratio β_m/β_r, where β_m is the air demand ratio measured with the Type 12 or 13 wedge, and β_r is the estimated value given by the best-fit Equation (15) for a two-dimensional ramp of the same height and subject to the same flow conditions. The values show that overall the Type 12 wedge entrains about 15% more air (per unit width of wedge) than the equivalent two-dimensional ramp. The corresponding average increase for the Type 13 design (with the small connecting ramp) is 52%.

These tests show that the Type 13 design of wedge aerator provides an efficient means of introducing air at the invert of a spillway channel while causing less disturbance to the high-velocity flow. The wedges can be manufactured in steel and added more easily and cheaply to spillways than conventional full-width designs of ramp aerator. The air flow to the wedges can be conveyed through a duct, partly recessed into the floor of the channel and partly projecting above in the form of a small ramp. The tests showed that this ramp increase the amount of air entrained by the wedges by 30% ; small air outlets in the duct would prevent cavitation damage downstream and would further increase the efficiency of the system.

8 CONCLUSIONS : PART A

(1) A systematic laboratory study has been made of the performance of two-dimensional ramp aerators for chute spillways. 322 tests were carried out to investigate the effect of the following factors on the amount of air entrainment : ramp height ; ramp angle ; flow velocity ; flow depth; head-loss characteristics of air supply system ; scale of model ; and level of turbulence in the flow.

(2) No air entrainment occurred below a minimum flow velocity V_o, which was found to be related to the height h_1 of the aerator by:

$$\frac{V_o}{(gh_1)^{1/2}} = F_* = 7.5$$

(3) Above this minimum velocity, the air demand per unit width q_a for a given aerator and water depth increased approximately linearly with the flow velocity V of the water.

(4) Increases in the following factors were found to increase the air demand q_a : height of aerator ; water velocity ; conveyance parameter M (Eqn (12)) of air supply system. Changing the angle of the ramp from 4.6° to 9.1° (while keeping its height constant) and varying the water depth d produced no overall change in the air demand.

(5) The following best-fit formula for predicting the amount of air entrained by ramp aerators was obtained from the test data.

$$\beta = 1.3\ [1 - \exp\ (-2.5M)]\ [1 - (F_*/F)(h_1/d)^{\frac{1}{2}}]\ [h_1/d]$$

where β is the non-dimensional air demand ratio (air discharge/water discharge) and F is the Froude number $(V/(gd)^{\frac{1}{2}})$. This formula is valid for a flume slope of 15.5°, and has been tested over the following range of conditions: $0.1 \leq M \leq 1.4$; $5.7 \leq F \leq 15.1$; $2.9 \leq h_1/d \leq 9.5$.

(6) Comparative tests were carried out with geometrically similar aerators at three different scales to determine whether the air demand varied with the size of the model. No consistent scale effect was found provided the water velocity in the test exceeded about 5.2m/s. Model tests must reproduce the head-loss characteristics of the air supply system correctly.

(7) Tests were made with three levels of turbulence in the water flow to investigate its effect on the air demand. Increasing the turbulence above the "normal" condition for the flume produced

some increase in the amount of air entrainment. If the turbulence intensity T_i (root-mean-square velocity divided by local mean velocity at mid-depth) exceeds 5%, the predicted air demand ratio β obtained from the equation in (5) should therefore be divided by a factor

$$\Omega = 1 - 2.0\ (T_i - 0.05)$$

(8) Measurements also showed that the length of the air cavity was reduced significantly when the turbulence level was increased. This is believed to be the reason why turbulence has only a relatively small effect on the amount of air entrainment.

(9) A new design of three-dimensional wedge aerator was developed which is approximately 50% more efficient at entraining air than an equivalent two-dimensional ramp but causes less disturbance to the flow. The design could be manufactured in steel and would be suitable for installation in both existing and new chute spillways.

(10) Further development testing of the wedge aerator is recommended to optimise the transverse spacing and method of air supply.

PART B - NUMERICAL AERATOR MODEL

9 SCOPE OF MODEL

CASCADE is a computer program developed by Hydraulics Research, Wallingford to assist in the design of aeration systems for preventing cavitation damage in chute spillways. CASCADE is an acronym for CAvitation Suppression on Chute spillways using Aeration DEvices.

The program is designed to:

- link with the computer program SWAN, developed by Binnie and Partners for the analysis of one-dimensional, steady state flow on a chute spillway. CASCADE is called into the main program when cavitation indicators show there to be a risk of cavitation damage.

- allow the operator to design an efficient aerator, to be installed at the point on the spillway where cavitation is predicted to be a problem.

- give the operator the option of specifying the air supply discharge, air duct geometry or ramp geometry.

- optimise the aerator geometry for specific flow criteria by equating the air demand (generated by the water flow over the aerator) with the rate of air supply (drawn from the atmosphere via a system of ducts).

Various researchers have investigated air entrainment at aerators and have described the phenomenon by equations of different form. The model gives the option of applying six alternative equations to determine the air demand. Some of these equations require an estimate of the length of the air cavity produced by the aerator. In the model this calculation is based on the work of Schwartz & Nutt (1963) on projected nappes but as modified by Rutschmann & Hager (1990).

The procedure to determine the distribution across the spillway of the pressure within the air cavity is based on that described by Rutschmann & Volkart (1988).

10 PROGRAM DESCRIPTION

10.1 General

The program SWAN computes one-dimensional flow down a spillway chute through three types of air/water flow regime; these are

(i) non-aerated, non-uniform flow;

(ii) partially and fully aerated non-uniform flow;

(iii) fully aerated uniform flow.

CASCADE is a program designed to model the air entrainment at an aerator and optimise the aerator geometry. It is called into use when SWAN indicates there to be a risk of cavitation damage. The basis of SWAN is described in Ackers & Priestley (1985), but some changes have been made as explained in Section 10.2.

CASCADE computations are carried out in four stages:

(i) Proposed ramp/step geometry and air ducting arrangement are initially selected together with a first estimate of either sub-nappe pressure or air supply discharge.

(ii) Air entrainment is calculated by several methods which have been derived by dimensional analysis combined with prototype and model measurements. Some methods require the length of the air cavity created by the aerator. In the model this length is determined by calculating the trajectory of the projected nappe using the solution obtained by Schwartz & Nutt (1963) but with modifications by Rutschmann & Hager (1990).

(iii) Frictional and point head losses in the air supply system are calculated using standard formulae. From an equation describing the air supply system, the sub-atmospheric pressure at the spillway side walls is calculated.

(iv) Following the mathematical calculation procedure proposed by Rutschmann & Volkart (1988), the variation across the spillway of the pressure in the air cavity is determined. For an assumed rate of air supply (or pressure at the spillway side wall), the quantity of air entrained is summed in steps across the channel. The calculation is repeated until the rate of air supply matches the total rate at which air is entrained. If the final value of air supply discharge or sub-nappe pressure is not suitable, the operator is given the option of changing various parameters and then repeating the procedure.

The four stages are described in detail in Sections 10.3 to 10.6.

10.2 Modifications to SWAN

SWAN has two sets of cavitation criteria based on research by Falvey (1983) and Arndt et al (1979). A recent review of the literature concerning cavitation indices by May (1987) recommended somewhat different criteria to those given by Arndt et al. SWAN was therefore modified to use these alternative cavitation indices, which are described below.

Considering the conditions required to produce cavitation at a particular point in a flow (e.g. at a step or obstruction), the instantaneous static pressure p_1 at the point of interest is found from Bernoulli's equation to be

$$p_1 = p_o - \rho g z - \tfrac{1}{2}\rho V_o^2\ [(1+\delta)^2(1+\epsilon)^2-1] \qquad (25)$$

where ρ is the density of the fluid, g is the acceleration due to gravity and z is the elevation of point 1 above reference point 0. δ is the proportionate change in the time-averaged velocity caused by the irregularity or change in boundary shape. The factor ϵ describes the instantaneous fluctuation in velocity due to turbulence.

If the absolute pressure p_1 falls below a critical value p_c, nuclei already existing in the flow will expand rapidly to form cavities. The cavitation index of the flow is defined by

$$K = \frac{(p_o - p_v)}{\tfrac{1}{2}\rho V_o^2} \qquad (26)$$

where p_v is the vapour pressure of the liquid at the given temperature. Incipient cavitation occurs when the local pressure p_1 drops to the critical pressure p_c. The corresponding value of the cavitation index, defined in terms of the mean flow conditions at the reference position is

$$K_i = \left[\frac{(p_o - p_v)}{\frac{1}{2}\rho V_o^2}\right]_i \qquad (27)$$

The cavitation number K is calculated from equation (26) above and compared with previously determined values of incipient cavitation index K_i for that type of irregularity. Cavitation will occur if
$K < K_i$

Values of K_i have been determined for various types of irregularity and, in general, most of the experimental results for a given type are in reasonable agreement. Abrupt offsets into the flow have been found to have the greatest cavitation potential. The most suitable formula for calculating K_i is that due to Liu (1983).

$$K_i = 1.02\, h_1^{0.326}, \text{ for } h_1 \leq 15\text{mm} \qquad (28)$$

where h_1 is the height of the offset in mm. The cavitation potential of construction faults can be reduced by grinding them to form chamfers. For an into-flow chamfer the slope needed to lower the value of K_i below the cavitation number K of the flow is estimated from the following empirical equations obtained by Novikova & Semenkov (1985).

$$K_i = 2.3, \text{ for } n \leq 1 \qquad (29)$$

$$K_i = 2.3n^{-0.7}, \text{ for } n > 1 \qquad (30)$$

where the slope is n units parallel to the flow to one unit normal to the flow.

At each distance step along the spillway, the modified version of SWAN compares the value of K for the flow with the values of K_i for the maximum permissible sizes of steps and chamfers (which are input with the other data for the spillway). If $K < K_i$, cavitation will occur and may result in damage unless self-aeration has caused the local air concentration to exceed a certain minimum value ; based on literature reviewed by May (1987), this minimum value used in SWAN is 7%. Thus, if SWAN finds that both $K < K_i$ and $C < 7\%$ at the bed, a warning of possible cavitation damage is given and the CASCADE model is brought into use for the design of a suitable aerator.

The criterion of Falvey (1983) which relates the value of K, its duration and the amount of cavitation damage is kept in SWAN as a warning message.

10.3 Input to CASCADE

The model calls for the geometry of a ramp and/or step to be initially specified. This includes proposed ramp height normal to the spillway floor, ramp angle, and/or step height normal to spillway floor. The geometrical parameters of the spillway (width, radius of curvature etc.) are calculated from within SWAN.

The operator has the option of selecting either the air supply discharge or the sub-nappe pressure. A knowledge of one allows calculation of the other.

10.4 Air entrainment

The physical processes involved in air entrainment are not yet properly understood and cannot therefore be described by fundamental theoretical equations. Data from experimental studies have thus been used to establish empirical relationships between dimensionless groupings of the main parameters.

Several alternative groupings are possible, but the set discussed in Section 4 is as follows:

$\beta = q_a/q$ = air demand ratio

$V/(gd)^{1/2}$ = Froude number

$\Delta p/\rho gd$ = sub-pressure term

Vd/ν = Reynolds number

$V/(\sigma/\rho d)^{1/2}$ = Weber number

$\tan\theta$ = tangent of the spillway slope

$\tan\phi$ = tangent of the ramp angle

$d/(h_1+u_1)$ = flow depth relative to overall height of aerator

v'/V = turbulence parameter

Investigations have shown that certain dimensionless numbers may be neglected if they exceed minimum values (400 for Weber and 10^5 for Reynolds according to Kobus (1984)). Turbulence is an influencing factor but, as yet, more research is needed to quantify its effect.

Empirical equations obtained from several of the more recent studies are included in CASCADE. These are described below.

10.4.1 Rutschmann (1988) carried out extensive tests on various shapes of aerator and recommended the equation

$$\beta = 0.0372\ (L_c/d) - 0.266 \tag{31}$$

This equation strictly applies when the pressure Δp in the air cavity is zero. However, it appears to give reasonable estimates of air demand (within about 20%) for cases where $\Delta p > 0$, provided the cavity length takes account of the effect of sub-atmospheric pressure.

Cavity length in the model is calculated using a general solution developed by Schwartz & Nutt (1963) for the form of a two-dimensional nappe of projected liquid. The equations of the x and z co-ordinates of the jet trajectory after time t are

$$\frac{x}{d} = \frac{F^2 \sin\Theta_o}{a \sin\alpha}\left[\cos\alpha - \cos\left(\frac{Vat}{F^2 d} + \alpha\right)\right] \tag{32}$$

$$\frac{z}{d} = -\frac{Vt}{ad} + \frac{F^2 \sin\Theta_o}{a \sin\alpha}\left[\sin\left(\frac{Vat}{F^2 d}\right) - \sin\alpha\right] \tag{33}$$

where

$$\Theta_o = (\Theta - \phi) \tag{34}$$

$$\alpha = \tan^{-1}\left[\frac{a \sin\Theta_o}{a \cos\Theta_o + 1}\right] \tag{35}$$

$$a = \frac{\Delta p}{\rho g d} \tag{36}$$

Experimental measurements indicate that internal pressures within a jet cause its take-off angle ϕ_i to be depressed below the angle ϕ of the ramp. Rutschmann & Hager (1990) estimated the reduction $\Delta\phi = (\phi - \phi_i)$ caused by streamline curvature to be given by

$$\Delta\phi = 2^{3/2}\ (1 - F^{-2})\ /\ (5F^2) \tag{37}$$

According to this result, the reduction in angle is negligible for $F > 6$.

The relative height of the ramp (h_1/d) also affects the take-off angle, and Pan et al (1980) proposed the relation

$$\frac{\phi_i}{\phi} = \left[\tanh \left(\frac{h_1/d}{\Theta} \right) \right]^{1/2} \qquad (38)$$

The reduction in angle needs to be considered if $(h_1/d) < 0.3$.

Both these modifications to the effective ramp angle are included in CASCADE. The length of the jet is determined by calculating the trajectory of the projected nappe after successive time-steps and establishing the point of impact of the jet on the spillway. The distance, measured along the spillway floor, between the point of impact and the take-off point on the ramp is the jet or cavity length.

10.4.2 Pinto et al (1982) obtained prototype measurements of air demand for ramp aerators on the Foz do Areia Dam in Brazil. Analysis showed that the air demand was given by

$$\beta = 0.033\ L_c/d \qquad (39)$$

for air supplied laterally from both sides of a spillway, and by

$$\beta = 0.023\ L_c/d \qquad (40)$$

for air supplied from one side only. Both equations are included as options in the CASCADE program. The cavity length is calculated using the Schwartz & Nutt method (see above).

10.4.3 Pan & Shao (1984) considered an approach to predicting air demand which would not require prior determination of the cavity length. Analysis of prototype and laboratory data led to the following empirical equation for air demand produced by a ramp or slot (not offset) in a channel of constant slope:

$$\beta = -0.0678 + 0.0982\, X_u - 0.0039\, X_u^2, \text{ for } X_u > 1 \qquad (41)$$

where

$$X_u = \frac{(h_1/d)^{1/2} F}{\cos\Theta \cos\phi} \qquad (42)$$

This equation does not take into account the head loss characteristics of the air supply.

10.4.4 Bruschin (1985) analysed the Foz de Areia data together with results from a model of Piedra del Aguilla Dam in Argentina. He used the overall step height w instead of L_c as the characteristic length and produced the formula

$$\beta = 0.0334\, F\, (w/d)^{1/2} \qquad (43)$$

10.4.5 Rutschmann & Hager (1990) used laboratory data to investigate the effect on the air demand ratio of variations in the Euler number, Froude number, spillway slope and aerator geometry. The resultant empirical expression for determining the air demand was

$$\beta/\beta_{max} = \left[\frac{2}{\Pi} \tan^{-1} (3 \times 10^{-3}\, \Delta E) \right]^{0.7} \qquad (44)$$

where

β_{max} = maximum air demand ratio when $\Delta p = 0$

$$= (\tan\phi)^{1.15} \exp[(1.15 \tan\Theta)^2] (F - 5.4)^{\gamma} \qquad (45)$$

$$\Delta E = E - E_{min} \qquad (46)$$

$$E = \text{Euler number} = \rho V^2/\Delta p \qquad (47)$$

$$E_{min} = 435 (\tan\phi)^{1.15} \exp[(1.15 \tan\Theta)^2] + 333(u_1/h)^2 \qquad (48)$$

The value of γ in Equation (45) varied between different sets of data ; three sets gave a value of $\gamma = 1.0$ and one set a value of $\gamma = 0.35$.

10.5 Air supply

An aerator on the floor of a spillway channel can be supplied with air either from outlets at the base of the side walls or from a manifold system installed across the width of the channel. In both cases, there will be some transverse variation of the pressure within the air cavity formed by the aerator. The velocity of air supplied by side outlets decreases towards the centreline of the spillway, and the resulting conversion of kinetic energy to pressure head causes an increase in the static pressure. Thus the value of Δp in the air cavity tends to decrease with distance from the side walls. Rutschmann & Volkart (1988) developed a method for allowing for the transverse variation of Δp in the case of side outlets, and this is included as an option in CASCADE.

If the required value of Δp at the side walls is specified, the air supply discharge is determined from:

$$Q_a = JA_a(\Delta p/\rho_a)^{½} \tag{49}$$

where the conveyance parameter J for the system is calculated from the geometry of the ducts using the method described in Appendix A. If neither Δp nor Q_a is initially known, a typical value of

$$Q_a = 0.25\ Q \tag{50}$$

is assumed for the first iteration. The air jet issuing from a side outlet is assumed to maintain a constant cross-sectional area A_a equal to that of the outlet. The difference in static pressures between two points 1 and 2 across the spillway is calculated from

$$p_1 - p_2 = \frac{\alpha}{2} \cdot \frac{k_z}{k_{zo}} \rho_a \left(\frac{Q_{a1}^2 - Q_{a2}^2}{A_a^2} \right) \tag{51}$$

where Q_{a1} and Q_{a2} are the rates of air flow within the jet (decreasing with distance from the side wall as air is entrained into the water flow). The coefficients k_z and k_{zo} take account of energy losses and their values are given by Rutschmann & Volkart ; α is the energy coefficient which is assumed to have a value of 1.05.

The calculation procedure is carried out iteratively as follows:

(i) The subpressure Δp at the side wall is assumed or calculated from Equation (49) (usually between 0 and 5 kPa).

(ii) Air supply discharge is either estimated or calculated from Equations (49) or (50).

(iii) Air demand discharge per unit width of spillway is calculated from one of the sets of equations in Section 10.4. The spillway width is divided into incremental lengths and the air demand calculated for each increment.

(iv) The pressure at the end of an increment is calculated according to Equation (51).

(v) The calculation is repeated for all subsequent elements until the centreline of the chute is reached (for symmetrical aeration) or until the opposite side wall is reached (for asymmetrical aeration).

(vi) If the sum of the individual air demands up to this point equals the rate at which air is supplied from the side outlet, then the assumed subpressure at the side wall is correct. If not, then the option is given to change the subpressure, air supply discharge or ramp geometry.

10.6 Aerator spacing

The physical process by which entrained air diffuses into the flow downstream of an aerator is very complex and not yet fully understood in detail. The air injected at the bed will tend to move upwards due to the combined effects of buoyancy and turbulent diffusion. At a certain distance along the spillway the local air concentration at the bed will fall below the value needed to prevent possible cavitation damage (assumed in CASCADE to be $C = 7\%$). Another aerator is then required to maintain the air concentration above the minimum figure for a sufficient distance downstream.

A simplified model of the diffusion process is used in SWAN to describe this process. The air added by the first aerator increases the mean air concentration above that already produced by self-aeration at the free surface. It is assumed that a short distance downstream of the aerator, the air becomes distributed through the depth according to the vertical profile that applies for a self-aerated flow with the same mean air concentration. The existing input-output model of self-aeration in SWAN (turbulence at the free surface drawing air downwards into the flow and buoyancy causing air bubbles to be lost upwards) is then used to determine how the mean air concentration changes with distance along the spillway. When the mean air concentration falls below a value which corresponds to $C = 7\%$ at the bed, a warning of possible cavitation damage is given (assuming also that $K < K_i$). CASCADE can then be used to design a second aerator.

11 DATA REQUIREMENTS FOR CASCADE

The user will be prompted to input the ramp and air duct geometries, air properties, estimated head loss in the duct system and initial estimate of either the subnappe pressure or the air supply discharge. An option is given for up to six different equations for the calculation of air demand. After completion of the run, the user is prompted to input an entirely new ramp arrangement or to change the air supply.

All input values have a free format. All input is from the keyboard and output is to a file "CASCADE.RES". Program information statements are held in CASCADE.INC. This file must be present when running the program.

The following is an example set of input data. The symbol < > denotes input from the user. Do not type the < > symbol as part of the input. (***** Note that the flow properties (discharge, velocity, etc) and spillway geometry (X, Y co-ordinates, width, side slope, radius, etc) have already been defined in the main program *****).

TITLE OF RUN (60 characters limit)

The next four lines are necessary if CASCADE has not been attached to SWAN.

WATER DEPTH NORMAL TO SPILLWAY (m) < >

MEAN WATER VELOCITY (m/s) < >

WATER DISCHARGE (cumecs) < >

LENGTH ALONG SPILLWAY FROM CREST TO RAMP POSITION (m)
< >

The next three lines describe the ramp geometry.

RAMP ANGLE (degrees) < >

RAMP HEIGHT NORMAL TO SPILLWAY (m) < >

The next line is optional. If no step is to be included then type < 0 > or < carriage return >

STEP HEIGHT NORMAL TO SPILLWAY (m) < >

The next line is not used if air supply discharge is to be specified. If no estimate of subnappe pressure is to be provided type < 0 > or < carriage return >

INITIAL ESTIMATE OF SUBNAPPE PRESSURE (Pa) < >

The next line allows for a choice of equation to calculate the air demand discharge. The six equations and their application and limitations are described in Section 10.4. Equation identifier should be an integer value between 1 and 6 according to the following list :

1. Rutschmann (10.4.1)
2. Pinto et al - air outlets on either side of channel (10.4.2)
3. Pinto et al- air outlet on one side only of channel (10.4.2)
4. Pan & Shao (10.4.3)
5. Bruschin (10.4.4)
6. Rutschmann & Hager (10.4.5)

AIR DEMAND DISCHARGE EQUATION IDENTIFIER (1-6)
< >

The next line is not used if the subnappe pressure is to be estimated. If this is the case or if the air supply discharge is not known then type < 0 > or < carriage return >

INITIAL ESTIMATE OF AIR SUPPLY DISCHARGE (cumecs)
< >

The next seven lines are concerned with the geometry of the air supply duct and the properties and distribution of the air.

AREA OF DUCT OUTLET (m^2)

PERIMETER OF DUCT OUTLET (m)

The next line is the headloss parameter (J) of the air supply system. The method of calculating the value of J for a given layout is described in Appendix A.

HEADLOSS PARAMETER J

The next line gives the option of air supply from either one side of the spillway only or from both sides.

AIR SUPPLY FROM ONE SIDE ONLY, ENTER 1 (integer)
AIR SUPPLY FROM BOTH SIDES, ENTER 2 (integer) < >

AIR TEMPERATURE (deg C) < >

The next line specifies the number of increments of width across the spillway to be used in the calculation of the lateral distribution of pressure. This value must be integer.

NUMBER OF WIDTH INCREMENTS ACROSS SPILLWAY < >

12 CONCLUSIONS : PART B

(1) A numerical model named CASCADE has been developed to assist the hydraulic design of aerators for chute spillways.

(2) CASCADE interfaces with a model called SWAN (produced by Binnie & Partners) that predicts the development of flow down a spillway and the amount of self-aeration at the free surface. SWAN identifies the point along the spillway at which surface irregularities can first begin to cause cavitation damage. CASCADE is then used to design a suitable aeration system : the aerators itself and the air supply ducts.

(3) CASCADE offers the option of five different methods for estimating the air demand, some of which require the calculation of the length of the air cavity formed by the aerator. Head losses in the air supply system and the effect of the pressure drop in the air cavity are taken into account.

(4) The convection and dispersion of the entrained air downstream of the aerator is predicted in the combined SWAN/CASCADE model by a simplified input/output description of the process. This determines the next point along the spillway at which another aerator is needed to prevent cavitation damage. The development of a more detailed convection/diffusion model of the two-phase flow is needed to improve the estimates of the aerator spacing.

(5) The alternative methods of predicting air demand do not give consistent results, and testing of the model against available prototype and model data is recommended to identify the most suitable equations.

13 ACKNOWLEDGEMENTS

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TABLES

TABLE 1 Main test data for ramp aerators

AIR VALVE SETTING	WATER DEPTH d mm	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO BETA	PRESSURE IN CAVITY Δp mm H2O	HEAD LOSS PARAMETER M	PARAMETER C
Aerator 1	h1= 8mm	ø= 9.1°					
4	106.0	6.190	6.071	0.0725	-	-	1.4551
4	46.0	5.027	7.484	0.0465	3.90	0.2414	0.4591
4	46.0	10.122	15.068	0.0969	17.50	0.4763	0.7035
4	46.0	5.027	7.484	0.1375	20.40	0.3118	1.3581
4	46.0	14.266	21.237	0.0652	46.20	0.2798	0.4399
4	46.0	12.635	18.810	0.0930	50.80	0.3346	0.6417
3	106.0	6.190	6.071	0.0785	-	-	1.5744
3	46.0	14.266	21.237	0.0750	45.50	0.3240	0.5053
3	46.0	10.122	15.068	0.0860	16.60	0.4334	0.6234
3	46.0	5.027	7.484	0.0595	2.90	0.3583	0.5874
3	46.0	12.635	18.810	0.0821	48.70	0.3014	0.5661
3	46.0	5.027	7.484	0.1587	20.10	0.3627	1.5682
2	46.0	14.266	21.237	0.0781	43.50	0.3454	0.5268
2	46.0	12.635	18.810	0.0775	46.50	0.2914	0.5348
1	46.0	12.635	18.810	0.0773	38.40	0.3199	0.5334
1	46.0	14.266	21.237	0.0777	39.30	0.3612	0.5237
0	106.0	6.190	6.071	0.0811	-	-	1.6263
0	106.0	6.456	6.331	0.0315	15.20	0.2419	0.6187
0	106.0	4.304	4.220	0.0204	2.40	0.2625	0.5276
0	76.0	8.224	9.524	0.0539	13.60	0.4071	0.6874
0	46.0	12.600	18.810	0.0778	41.60	0.3129	0.5387
0	46.0	5.163	7.685	0.0427	5.20	0.1994	0.4144
0	46.0	5.027	7.484	0.0701	2.80	0.4296	0.6922
0	46.0	14.266	21.237	0.0777	33.20	0.3930	0.5237
0	46.0	5.027	7.484	0.1606	19.80	0.3699	1.5865
0	46.0	8.016	11.933	0.0692	30.90	0.2053	0.5399
0	46.0	5.027	7.483	0.0701	3.00	0.4151	0.6922
0	46.0	7.600	11.326	0.0996	24.80	0.3123	0.7927
0	46.0	5.027	7.483	0.0704	2.80	0.4298	0.6955
0	46.0	7.200	10.719	0.0726	21.20	0.2337	0.5897
0	46.0	4.890	7.281	0.0429	3.80	0.2213	0.4328
0	46.0	6.657	9.910	0.0630	16.80	0.2105	0.5295
0	46.0	4.755	7.079	0.0488	2.90	0.2801	0.5028
0	46.0	12.635	18.810	0.0760	45.20	0.2900	0.5244
0	46.0	4.619	6.876	0.0401	2.40	0.2456	0.4228
0	46.0	7.744	11.528	0.0687	29.50	0.2016	0.5405
0	46.0	4.480	6.670	0.0604	2.00	0.3937	0.6547
0	46.0	6.929	10.315	0.0662	18.90	0.2169	0.5460
0	46.0	4.347	6.470	0.0455	1.40	0.3437	0.5062
0	46.0	10.122	15.068	0.0792	29.20	0.3046	0.5749
0	46.0	6.385	9.506	0.0579	13.30	0.2086	0.4966
0	46.0	7.470	11.120	0.0724	28.70	0.2079	0.5797
0	46.0	4.211	6.270	0.0469	1.40	0.3437	0.5388
0	46.0	4.076	6.067	0.0594	1.10	0.4751	0.0000

TABLE 1 (cont) Main test data for ramp aerators

Aerator 3 h1= 8mm ø= 4.57°

AIR VALVE SETTING	WATER DEPTH d mm	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO BETA	PRESSURE IN CAVITY Δp mm H2O	HEAD LOSS PARAMETER M	PARAMETER C
4	106.0	6.190	6.071	0.0508	-	-	1.0189
4	46.0	12.635	18.810	0.0900	79.10	0.2602	0.6206
4	46.0	10.122	15.068	0.0887	51.70	0.2544	0.6439
4	46.0	14.266	21.237	0.0884	96.20	0.2627	0.5963
4	46.0	5.910	8.798	0.0490	5.00	0.2651	0.4375
4	46.0	7.608	11.326	0.0964	18.70	0.3455	0.7656
3	46.0	7.608	11.326	0.1128	17.00	0.4242	0.8964
3	46.0	14.266	21.237	0.1005	100.70	0.2919	0.6780
3	46.0	10.122	15.068	0.0983	50.50	0.2854	0.7137
3	46.0	5.910	8.798	0.0995	4.50	0.5671	0.8878
3	46.0	12.635	18.810	0.1070	75.80	0.3160	0.7377
2	46.0	7.608	11.326	0.1191	17.40	0.4427	0.9466
2	46.0	10.122	15.068	0.1133	45.30	0.3464	0.8219
2	46.0	14.260	21.237	0.1052	101.70	0.3037	0.7095
2	46.0	12.600	18.810	0.1083	76.40	0.3186	0.7492
1	46.0	12.600	18.810	0.1091	75.40	0.3232	0.7553
1	46.0	14.266	21.230	0.1203	92.20	0.3649	0.8112
1	46.0	10.122	15.068	0.1091	46.10	0.3308	0.7917
0	106.0	6.190	6.071	0.0547	-	-	1.0969
0	106.0	6.450	6.330	0.0169	5.00	0.2264	0.3326
0	76.0	8.224	9.524	0.0518	34.50	0.2460	0.6612
0	76.0	7.400	8.570	0.0581	19.90	0.3271	0.7715
0	46.0	7.200	10.719	0.0706	14.70	0.2730	0.5736
0	46.0	5.027	7.483	0.0395	0.60	0.5250	0.3905
0	46.0	8.016	11.930	0.0675	27.40	0.2125	0.5260
0	46.0	5.027	7.483	0.0395	1.40	0.3437	0.3905
0	46.0	7.608	11.326	0.1187	17.30	0.4426	0.9436
0	46.0	14.266	21.237	0.1214	89.50	0.3738	0.8186
0	46.0	10.122	15.068	0.1126	45.3	0.3445	0.8172
0	46.0	6.657	9.910	0.0556	10.70	0.2325	0.4672
0	46.0	12.600	18.810	0.1110	75.40	0.3287	0.7680
0	46.0	7.472	11.124	0.0742	18.19	0.2675	0.5935
0	46.0	6.385	9.506	0.0537	8.00	0.2491	0.4602
0	46.0	6.929	10.315	0.0647	12.30	0.2625	0.5336
0	46.0	7.744	11.528	0.0699	21.70	0.2388	0.5514

TABLE 1 (cont) Main test data for ramp aerators

AIR VALVE SETTING	WATER DEPTH d mm	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO BETA	PRESSURE IN CAVITY Δp mm H2O	HEAD LOSS PARAMETER M	PARAMETER C
Aerator 4 h1= 12mm ø= 9.1°							
0	72.0	9.418	11.207	0.0481	60.60	0.1244	0.3968
0	72.0	9.071	10.794	0.0461	52.80	0.1231	0.3859
0	72.0	8.767	10.432	0.0463	47.50	0.1260	0.3931
0	72.0	8.464	10.071	0.0434	43.20	0.1195	0.3742
0	72.0	8.116	9.657	0.0479	39.30	0.1325	0.4208
0	72.0	7.813	9.296	0.0487	35.30	0.1370	0.4361
0	72.0	7.509	8.934	0.0467	24.00	0.1528	0.4262
0	72.0	6.076	7.230	0.0566	12.60	0.2075	0.5890
0	72.0	5.903	7.024	0.0557	11.40	0.2088	0.5929
0	72.0	5.729	6.817	0.0553	10.00	0.2144	0.6020
0	72.0	5.599	6.662	0.0536	8.20	0.2247	0.5955
0	72.0	5.425	6.456	0.0509	6.30	0.2354	0.5806
0	72.0	5.252	6.249	0.0391	4.40	0.2094	0.4605
0	72.0	5.122	6.094	0.0339	3.60	0.1956	0.4088
Aerator 6 h1= 12mm ø= 4.57°							
0	72.0	9.418	11.207	0.0493	70.20	0.1184	0.4067
0	72.0	9.071	10.794	0.0485	55.60	0.1260	0.4058
0	72.0	8.767	10.432	0.0458	47.90	0.1239	0.3887
0	72.0	8.464	10.071	0.0469	43.50	0.1284	0.4042
0	72.0	8.116	9.657	0.0460	36.90	0.1311	0.4040
0	72.0	7.813	9.296	0.0464	31.00	0.1388	0.4148
0	72.0	7.509	8.934	0.0486	25.00	0.1557	0.4437

TABLE 1 (cont) Main test data for ramp aerators

AIR VALVE SETTING	WATER DEPTH d mm	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO BETA	PRESSURE IN CAVITY Δp mm H2O	HEAD LOSS PARAMETER M	PARAMETER C
Aerator 7 h1= 16mm ø= 9.1°							
4	106.0	6.190	6.071	0.0841	-	-	1.0719
4	76.0	8.430	9.762	0.0897	25.90	0.2527	0.6577
4	76.0	7.480	8.667	0.1084	17.80	0.3269	0.8545
4	76.0	6.540	7.572	0.1007	11.50	0.3304	0.8764
4	46.0	4.212	6.270	0.0554	2.30	0.1572	0.5405
4	46.0	10.122	15.068	0.0906	10.10	0.2949	0.3686
4	46.0	7.608	11.326	0.0969	17.70	0.1792	0.4570
4	46.0	5.027	7.484	0.0504	10.60	0.0796	0.3546
3	106.0	6.190	6.071	0.0882	-	-	1.1241
3	76.0	8.430	9.762	0.0920	25.80	0.2598	0.6748
3	76.0	7.480	8.667	0.1088	18.20	0.3247	0.8579
3	76.0	6.540	7.572	0.1152	11.10	0.3848	1.0025
3	46.0	5.027	7.484	0.0804	11.20	0.1234	0.5650
3	46.0	7.608	11.326	0.1175	17.50	0.2187	0.5546
3	46.0	10.122	15.068	0.0950	16.60	0.2413	0.3866
2	76.0	7.480	8.667	0.1093	17.80	0.3295	0.8614
2	76.0	8.430	9.762	0.0920	25.60	0.2608	0.6748
2	46.0	6.726	10.011	0.1284	15.60	0.2237	0.6613
2	46.0	7.608	11.326	0.1253	16.40	0.2408	0.5909
2	46.0	10.122	15.068	0.0968	18.70	0.2315	0.3938
1	46.0	10.122	15.068	0.0997	17.90	0.2439	0.4058
0	106.0	6.190	6.071	0.0870	-	-	1.1082
0	106.0	5.160	5.059	0.0657	-	-	1.0263
0	106.0	5.395	5.291	0.0377	3.50	0.2520	0.5562
0	106.0	6.456	6.331	0.0451	26.00	0.1322	0.5536
0	96.0	6.445	6.642	0.0355	21.30	0.1057	0.3948
0	96.0	6.087	6.273	0.0407	15.30	0.1351	0.4772
0	96.0	6.641	6.843	0.0362	25.40	0.1019	0.3934
0	96.0	5.534	5.702	0.0390	6.50	0.1806	0.5056
0	96.0	6.250	6.440	0.0374	18.10	0.1171	0.4272
0	96.0	5.892	6.071	0.0417	12.60	0.1475	0.5042
0	96.0	5.697	5.870	0.0410	10.10	0.1568	0.5142
0	96.0	6.999	7.212	0.0344	30.70	0.0927	0.3585
0	96.0	6.803	7.011	0.0350	26.30	0.0991	0.3730
0	76.0	8.223	9.524	0.0553	34.20	0.1311	0.4113
0	76.0	7.480	8.667	0.1098	18.20	0.3276	0.8660
0	76.0	8.430	9.762	0.0942	25.70	0.2665	0.6906
0	76.0	4.111	4.762	0.0586	3.10	0.2310	1.0042
0	76.0	6.578	7.619	0.0663	14.10	0.1962	0.5746
0	76.0	6.540	7.572	0.1184	10.00	0.4167	1.0307
0	76.0	4.687	5.428	0.0541	0.10	1.3642	0.7016
0	46.0	10.122	15.068	0.1009	18.90	0.2401	0.4106
0	46.0	7.608	11.326	0.1269	16.30	0.2448	0.5990
0	46.0	5.910	8.790	0.1309	12.20	0.2263	0.7567
0	46.0	4.211	6.270	0.0868	2.00	0.2641	0.8474

TABLE 1 (cont) Main test data for ramp aerators

Aerator 9 h1= 16 mm ø= 4.57°

AIR VALVE SETTING	WATER DEPTH d mm	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO BETA	PRESSURE IN CAVITY Δp mm H2O	HEAD LOSS PARAMETER M	PARAMETER C
4	106.0	6.190	6.071	0.0591	-	-	0.7536
4	76.0	8.430	9.762	0.0809	33.20	0.2019	0.5936
4	76.0	8.430	9.762	0.0955	33.20	0.2374	0.7008
4	76.0	4.690	5.429	0.0476	0.30	0.6923	0.6164
4	76.0	7.480	8.667	0.0914	17.90	0.2739	0.7206
4	76.0	5.630	6.524	0.0534	2.30	0.3384	0.5379
4	76.0	5.630	6.524	0.0859	4.40	0.3924	0.8649
4	76.0	6.540	7.572	0.0838	7.80	0.3348	0.7289
4	76.0	7.480	8.667	0.1070	17.50	0.3265	0.8438
4	46.0	8.424	12.540	0.1571	19.60	0.3034	0.6978
4	46.0	5.030	7.484	0.0640	0.20	0.7193	0.4493
4	46.0	3.396	5.056	0.2107	2.50	0.4592	4.8424
4	46.0	5.910	8.798	0.1309	5.40	0.3333	0.7572
4	46.0	9.239	13.754	0.1562	24.00	0.2990	0.6620
4	46.0	6.730	10.012	0.1402	6.80	0.3616	0.7210
4	46.0	1.698	2.528	0.3632	3.00	0.3643	-1.3929
4	46.0	10.120	15.068	0.1623	22.60	0.3456	0.6609
4	46.0	13.450	20.024	0.1674	46.40	0.3304	0.6177
4	46.0	9.239	13.754	0.1422	23.30	0.2763	0.6027
4	46.0	9.239	13.754	0.1761	24.90	0.3326	0.7461
4	46.0	12.640	18.810	0.1705	40.40	0.3390	0.6405
4	46.0	7.610	11.327	0.1976	11.40	0.4527	0.9320
3	106.0	6.190	6.071	0.0564	-	-	0.7187
3	76.0	8.430	9.762	0.0978	32.20	0.2466	0.7173
3	76.0	6.540	7.572	0.0920	6.70	0.3968	0.8009
3	76.0	7.480	8.667	0.1091	16.80	0.3397	0.8600
3	76.0	5.630	6.524	0.0976	3.80	0.4797	0.9822
3	76.0	8.430	9.762	0.0847	31.70	0.2164	0.6213
3	76.0	5.630	6.524	0.0565	1.10	0.5179	0.5687
3	76.0	7.480	8.667	0.0979	17.30	0.2985	0.7714
3	46.0	6.730	10.012	0.1723	5.40	0.4988	0.8866
3	46.0	8.424	12.540	0.1159	17.60	0.2373	0.5146
3	46.0	5.910	8.798	0.1509	3.20	0.4990	0.8724
3	46.0	13.450	20.024	0.2116	43.40	0.4320	0.7809
3	46.0	10.120	15.068	0.2017	21.40	0.4413	0.8211
3	46.0	8.424	12.540	0.1786	16.00	0.3817	0.7933
3	46.0	7.610	11.327	0.1950	10.60	0.4631	0.9196
3	46.0	8.424	12.540	0.1184	17.10	0.3896	0.8370
3	46.0	5.030	7.484	0.1012	0.20	1.1372	0.7101
3	46.0	2.514	3.740	0.2514	1.10	0.6171	-3.9668
3	46.0	12.640	18.810	0.2125	40.00	0.4247	0.7986
3	46.0	1.698	2.528	0.4159	2.60	0.4482	-1.5951
3	46.0	3.397	5.057	0.2647	4.20	0.4452	6.0761
3	46.0	9.239	13.753	0.1925	23.20	0.3747	0.8157

TABLE 1 (cont) Main test data for ramp aerators

AIR VALVE SETTING	WATER DEPTH d mm	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO BETA	PRESSURE IN CAVITY Δp mm H2O	HEAD LOSS PARAMETER M	PARAMETER C
Aerator 9 h1= 16 mm ϕ= 4.57							
2	106.0	6.190	6.071	0.0635	-	-	0.8086
2	76.0	6.540	7.572	0.0952	5.00	0.4747	0.8286
2	76.0	7.480	8.667	0.1006	17.20	0.3078	0.7932
2	76.0	8.430	9.762	0.0972	32.10	0.2454	0.7126
2	76.0	8.430	9.762	0.0979	32.00	0.2488	0.7178
2	46.0	5.910	8.798	0.0929	2.90	0.3298	0.5369
2	46.0	7.608	11.326	0.1316	13.20	0.2819	0.6208
0	106.0	6.190	6.071	0.0617	-	-	0.7868
0	106.0	6.456	6.331	0.0387	5.20	0.2542	0.4749
0	76.0	5.630	6.524	0.1080	3.90	0.5237	1.0870
0	76.0	6.578	7.619	0.0644	5.50	0.3050	0.5581
0	76.0	7.480	8.667	0.1004	16.60	0.3127	0.7920
0	76.0	5.633	6.524	0.0534	1.20	0.4687	0.5370
0	76.0	8.223	9.524	0.0531	34.50	0.1254	0.3946
0	76.0	4.690	5.429	0.0541	0.10	1.3643	0.7004
0	76.0	8.430	9.762	0.1011	31.00	0.2612	0.7418
0	76.0	6.540	7.572	0.0957	5.60	0.4507	0.8324
0	76.0	5.630	6.524	0.0534	1.20	0.4685	0.5379
0	76.0	8.430	9.762	0.0968	31.30	0.2476	0.7098
0	46.0	13.450	20.024	0.2453	41.20	0.5138	0.9051
0	46.0	6.730	10.012	0.1616	23.60	0.2273	0.8316
0	46.0	12.640	18.810	0.2441	38.60	0.4966	0.9173
0	46.0	5.910	8.798	0.1667	2.40	0.6365	0.9639
0	46.0	9.239	13.753	0.2132	21.60	0.4300	0.9036
0	46.0	7.610	11.327	0.2049	10.00	0.5013	0.9665
0	46.0	7.609	11.327	0.1185	10.50	0.2824	0.5590
0	46.0	3.397	5.056	0.1911	3.50	0.3543	4.3871
0	46.0	8.424	12.540	0.2097	15.70	0.4522	0.9316
0	46.0	10.120	15.068	0.2289	20.80	0.5080	0.9319
0	46.0	7.609	11.327	0.1792	9.80	0.4412	0.8452
0	46.0	2.514	3.742	0.2090	0.70	0.6436	-3.2987
0	46.0	1.698	2.528	0.4477	2.20	0.5244	-1.7164

TABLE 2 Test data on scale effects for ramp aerator

RAMP ANGLE deg	SCALE RATIO	RELATIVE FLOW DEPTH d/h1	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO βm	HEAD LOSS PARAMETER M	βe/βm
9.10	2:1	6.000	6.999	7.212	0.0344	0.0927	0.7500
9.10	2:1	6.000	5.892	6.071	0.0417	0.1475	0.7941
9.10	2:1	6.000	6.803	7.011	0.0350	0.0991	0.7648
9.10	2:1	6.000	5.697	5.870	0.0410	0.1568	0.8197
9.10	2:1	6.000	6.087	6.273	0.0407	0.1351	0.7810
9.10	2:1	6.000	6.250	6.440	0.0374	0.1171	0.7722
9.10	2:1	6.000	5.534	5.702	0.0390	0.1806	0.9349
9.10	2:1	6.000	6.641	6.843	0.0362	0.1019	0.7430
9.10	2:1	6.000	6.445	6.642	0.0355	0.1057	0.7645
9.10	1.5:1	6.000	8.464	10.071	0.0434	0.1195	0.8968
9.10	1.5:1	6.000	8.116	9.657	0.0479	0.1325	0.8712
9.10	1.5:1	6.000	9.071	10.794	0.0461	0.1231	0.8921
9.10	1.5:1	6.000	7.509	8.934	0.0467	0.1528	0.9684
9.10	1.5:1	6.000	5.252	6.249	0.0391	0.2094	1.1508
9.10	1.5:1	6.000	8.767	10.432	0.0463	0.1260	0.8937
9.10	1.5:1	6.000	5.903	7.024	0.0557	0.2088	0.8917
9.10	1.5:1	6.000	9.418	11.207	0.0481	0.1244	0.8759
9.10	1.5:1	6.000	7.813	9.296	0.0487	0.1370	0.8642
9.10	1.5:1	6.000	5.729	6.817	0.0553	0.2144	0.8964
9.10	1.5:1	6.000	5.599	6.662	0.0536	0.2247	0.9381
9.10	1.5:1	6.000	5.425	6.456	0.0509	0.2354	0.9961
9.10	1.5:1	6.000	5.122	6.094	0.0339	0.1956	1.2290
9.10	1.5:1	6.000	6.076	7.230	0.0566	0.2075	0.8934
9.10	1:1	5.750	4.755	7.079	0.0488	0.2801	1.3018
9.10	1:1	5.750	6.657	9.910	0.0630	0.2105	1.0046
9.10	1:1	5.750	4.890	7.281	0.0429	0.2213	1.2769
9.10	1:1	5.750	7.200	10.719	0.0726	0.2337	0.9756
9.10	1:1	5.750	5.027	7.483	0.0704	0.4298	1.2304
9.10	1:1	5.750	7.600	11.326	0.0996	0.3123	0.8902
9.10	1:1	5.750	8.016	11.933	0.0692	0.2053	0.9680
9.10	1:1	5.750	5.027	7.484	0.1606	0.3699	0.4944
9.10	1:1	5.750	12.635	18.810	0.0760	0.2900	1.2785
9.10	1:1	5.750	5.027	7.484	0.0701	0.4296	1.2366
9.10	1:1	5.750	6.385	9.506	0.0579	0.2086	1.0639
9.10	1:1	5.750	4.347	6.470	0.0455	0.3437	1.4809
9.10	1:1	5.750	4.211	6.270	0.0469	0.3437	1.3917
9.10	1:1	5.750	7.470	11.120	0.0724	0.2079	0.9093
9.10	1:1	5.750	4.619	6.876	0.0401	0.2456	1.4111
9.10	1:1	5.750	10.122	15.068	0.0792	0.3046	1.2051
9.10	1:1	5.750	7.744	11.528	0.0687	0.2016	0.9492
9.10	1:1	5.750	6.929	10.315	0.0662	0.2169	0.9963
9.10	1:1	5.750	4.480	6.670	0.0604	0.3937	1.2442
9.10	1:1	5.750	14.266	21.237	0.0777	0.3930	1.5530
9.10	1:1	5.750	4.076	6.067	0.0594	0.4751	1.2816
9.10	1:1	5.750	5.163	7.685	0.0427	0.1994	1.2312
9.10	1:1	5.750	12.600	18.810	0.0778	0.3129	1.3139

TABLE 2 (cont) Test data on scale effects for ramp aerator

RAMP ANGLE deg	SCALE RATIO	RELATIVE FLOW DEPTH d/h1	WATER VELOCITY V m/s	FROUDE NUMBER F	AIR DEMAND RATIO βm	HEAD LOSS PARAMETER M	βe/βm
4.57	1.5:1	6.000	7.509	8.934	0.0486	0.1557	0.9443
4.57	1.5:1	6.000	8.767	10.432	0.0458	0.1239	0.8911
4.57	1.5:1	6.000	9.418	11.207	0.0493	0.1184	0.8188
4.57	1.5:1	6.000	9.071	10.794	0.0485	0.1260	0.8655
4.57	1.5:1	6.000	8.464	10.071	0.0469	0.1284	0.8829
4.57	1.5:1	6.000	8.116	9.657	0.0460	0.1311	0.8991
4.57	1.5:1	6.000	7.813	9.296	0.0464	0.1388	0.9189
4.57	1:1	5.750	5.027	7.483	0.0395	0.5250	2.4330
4.57	1:1	5.750	7.200	10.719	0.0706	0.2730	1.1213
4.57	1:1	5.750	5.027	7.483	0.0395	0.3437	1.9193
4.57	1:1	5.750	7.472	11.124	0.0742	0.2675	1.0677
4.57	1:1	5.750	6.657	9.910	0.0556	0.2325	1.2259
4.57	1:1	5.750	6.929	10.315	0.0647	0.2625	1.1719
4.57	1:1	5.750	6.385	9.506	0.0537	0.2491	1.3093
4.57	1:1	5.750	7.744	11.528	0.0699	0.2388	1.0596
4.57	1:1	5.750	12.600	18.810	0.1110	0.3287	0.9517
4.57	1:1	5.750	7.608	11.326	0.1187	0.4426	0.9227
4.57	1:1	5.750	10.122	15.068	0.1126	0.3445	0.9187
4.57	1:1	5.750	14.266	21.237	0.1214	0.3738	0.9644
4.57	1:1	5.750	8.016	11.930	0.0675	0.2125	1.0182

TABLE 3 Additional test data (turbulence and cavity lengths) for ramp aerators

WATER DEPTH d mm	WATER VELOCITY V m/s	AIR DEMAND RATIO βm	HEAD LOSS PARAMETER M	TURBULENCE TYPE	TURBULENCE INTENSITY Ti %	CAVITY LENGTH Lc m	βe/βm
Aerator 1 h1= 8 mm ø= 9.1°							
76.0	5.756	0.0614	0.381	A	5.92	0.470	0.869
46.0	12.635	0.1117	0.405	A	3.09	0.900	1.074
76.0	4.934	0.0355	0.36	A	5.88	0.320	1.313
46.0	7.609	0.2027	1.03	A	3.17	0.755	0.746
76.0	5.756	0.0641	0.39	A	6.05	0.360	0.844
76.0	8.223	0.0894	0.42	A	5.78	0.620	0.741
46.0	12.635	0.1091	0.4	A	3.53	0.880	1.092
46.0	10.122	0.1225	0.48	A	7.13	0.850	1.022
76.0	8.223	0.088	0.42	A	7.64	0.650	0.753
46.0	7.608	0.1507	0.63	A	3.01	0.750	0.861
76.0	4.934	0.0809	0.37	B	20.47	0.225	0.586
46.0	7.608	0.1608	0.613	B	27.60	0.370	0.798
46.0	10.122	0.1311	0.5	B	21.91	0.380	0.975
76.0	5.757	0.0947	0.624	C	7.50	0.258	0.725
46.0	12.636	0.1023	0.72	C	21.65	0.102	1.538
76.0	4.934	0.0714	0.328	C	14.09	0.190	0.616
76.0	4.934	0.0714	0.32	C	14.09	0.190	0.606
76.0	5.757	0.1061	0.6	C	16.59	0.000	0.636
76.0	8.224	0.1089	0.33	C	17.58	0.202	0.526
76.0	8.224	0.1009	0.31	C	20.88	0.222	0.545

TABLE 3 (cont) Additional test data (turbulence and cavity lengths) for ramp aerators

WATER DEPTH d mm	WATER VELOCITY V m/s	AIR DEMAND RATIO βm	HEAD LOSS PARAMETER M	TURBULENCE TYPE	TURBULENCE INTENSITY Ti %	CAVITY LENGTH Lc m	$\beta e/\beta m$
Aerator 3	h1= 8 mm	ϕ= 4.6°					
76.0	6.578	0.06	0.451	A	4.82	0.270	1.050
46.0	12.635	0.129	0.396	A	8.91	0.520	0.918
76.0	8.223	0.0889	0.79	A	8.51	0.380	0.987
76.0	5.756	0.0445	0.447	A	4.53	0.250	1.314
46.0	7.608	0.1483	0.477	A	3.43	0.480	0.769
46.0	10.122	0.1277	0.331	A	3.49	0.510	0.790
76.0	6.578	0.0673	0.531	A	5.00	0.300	1.017
46.0	12.635	0.1385	0.439	A	4.83	0.590	0.907
76.0	8.224	0.0877	0.797	A	5.98	0.340	1.003
46.0	10.122	0.1301	0.357	A	3.53	0.520	0.813
46.0	7.608	0.1509	0.479	A	4.14	0.470	0.757
76.0	6.579	0.0788	0.535	B	17.02	0.144	0.872
46.0	12.636	0.1431	0.388	B	11.87	0.270	0.818
46.0	7.609	0.156	0.432	B	7.49	0.220	0.693
76.0	5.757	0.0767	0.495	B	17.89	0.114	0.804
46.0	12.635	0.1556	0.409	B	12.03	0.250	0.776
46.0	10.122	0.14	0.344	B	8.30	0.260	0.738
76.0	6.578	0.0879	0.626	B	15.35	0.130	0.838
46.0	7.609	0.1555	0.433	B	7.36	0.210	0.696
46.0	10.122	0.1393	0.346	B	8.89	0.230	0.745
46.0	12.636	0.1269	0.437	C	11.29	0.099	0.987
76.0	6.579	0.0873	0.314	C	17.19	0.145	0.580
76.0	8.224	0.097	0.296	C	16.77	0.142	0.549
46.0	7.609	0.1602	0.791	C	4.07	0.100	0.880
76.0	5.757	0.0745	0.228	C	19.54	0.120	0.507
46.0	10.122	0.1355	0.433	C	6.62	0.092	0.874
46.0	12.636	0.1374	0.407	C	8.84	0.093	0.876
76.0	6.579	0.1038	0.366	C	14.58	0.170	0.538
76.0	8.224	0.1001	0.442	C	16.12	0.155	0.681
46.0	7.609	0.1576	2.725	C	4.53	0.121	1.037
46.0	10.122	0.136	0.417	C	5.33	0.094	0.853

TABLE 3 (cont) Additional test data (turbulence and cavity lengths) for ramp aerators

WATER DEPTH d mm	WATER VELOCITY V m/s	AIR DEMAND RATIO βm	HEAD LOSS PARAMETER M	TURBULENCE TYPE	TURBULENCE INTENSITY Ti %	CAVITY LENGTH Lc m	βe/βm
Aerator 7 h1= 16 mm ø= 9.1°							
46.0	7.608	0.1378	0.29	A	3.34	1.060	1.031
46.0	10.122	0.1182	0.33	A	5.25	1.330	1.518
46.0	7.608	0.1571	0.33	A	3.17	1.080	0.985
46.0	6.725	0.1631	0.36	A	3.93	0.930	0.918
46.0	10.122	0.1178	0.33	A	4.43	1.460	1.523
46.0	7.608	0.1558	0.34	B	7.27	0.780	1.013
46.0	10.122	0.126	0.28	B	13.59	0.950	1.276
46.0	10.122	0.1345	0.36	B	8.09	0.960	1.409
76.0	7.401	0.1195	0.2	C	14.67	0.239	0.539
76.0	8.224	0.1115	0.19	C	15.18	0.202	0.593
46.0	10.122	0.1431	1.74	C	12.93	0.270	2.203
46.0	6.726	0.1869	1.34	C	8.99	0.212	1.303
76.0	7.401	0.1215	0.22	C	12.93	0.267	0.570
76.0	8.224	0.1478	0.25	C	11.28	0.162	0.550
76.0	6.579	0.1281	0.23	C	18.54	0.333	0.512
Aerator 9 h1= 16 mm ø= 4.6°							
46.0	6.725	0.1326	0.3	A	6.51	0.550	1.004
46.0	10.122	0.1245	0.19	A	2.91	0.700	0.970
46.0	7.609	0.1569	0.3	A	7.31	0.640	0.927
46.0	12.635	0.1223	0.17	A	4.96	0.750	0.979
46.0	6.725	0.1667	0.29	A	6.88	0.580	0.781
46.0	6.726	0.1728	0.29	B	8.18	0.380	0.753
46.0	10.122	0.1447	0.17	B	7.90	0.420	0.764
46.0	7.609	0.1566	0.22	B	19.45	0.400	0.745
46.0	12.635	0.1789	0.23	B	9.66	0.600	0.845
46.0	6.725	0.1824	0.31	B	9.96	0.360	0.746
76.0	7.401	0.0968	0.2	C	14.28	0.223	0.666
46.0	6.726	0.1638	1.24	C	18.66	0.174	1.472
46.0	7.609	0.0157	0.09	C	17.06	0.280	3.537
76.0	7.401	0.1029	0.22	C	13.81	0.194	0.673
76.0	8.224	0.1010	0.2	C	14.57	0.213	0.681
76.0	6.579	0.1123	0.29	C	16.46	0.232	0.689
46.0	6.726	0.1853	1.78	C	17.14	0.237	1.346
76.0	8.224	0.1012	0.2	C	14.88	0.230	0.680

TABLE 4 Test data for wedge-shaped aerators

Water depth d mm	Water velocity V m/s	Froude number F	Air discharge Qa m3/s	Air demand ratio βm	Pressure in cavity Δp mm H2O	Head loss parameter M	βm/βr eqn(15)
		Aerator 12	h1=16mm	ø=6.8°			
76	8.635	10.00	1.850E-02	0.1762	18.9	0.5931	1.270
76	7.648	8.857	1.658E-02	0.1783	17.3	0.5556	1.419
76	6.127	7.095	8.968E-03	0.1204	3.3	0.6882	1.040
76	4.523	5.238	0	0	2.2	-	-
46	14.27	21.24	2.729E-02	0.2599	90.2	0.3974	1.153
46	12.64	18.81	1.961E-02	0.2109	30.5	0.4884	0.865
46	10.12	15.07	1.994E-02	0.2677	31.3	0.4914	1.185
		Aerator 13	h1=16mm	ø=6.8°			
76	8.635	10.00	2.806E-02	0.2672	41.2	0.6081	1.905
76	7.648	8.857	2.097E-02	0.2255	29.7	0.5353	1.827
76	6.127	7.095	1.226E-02	0.1646	2.4	0.9872	1.276
76	4.523	5.238	0	0	0.4	-	-
46	14.27	21.24	4.418E-02	0.4207	93.1	0.6347	1.477
46	12.64	18.81	3.786E-02	0.4071	70.8	0.6237	1.491
46	10.12	15.07	2.988E-02	0.4010	36.7	0.6835	1.533
46	7.473	11.12	1.639E-02	0.2979	1.8	1.693	1.110

FIGURES

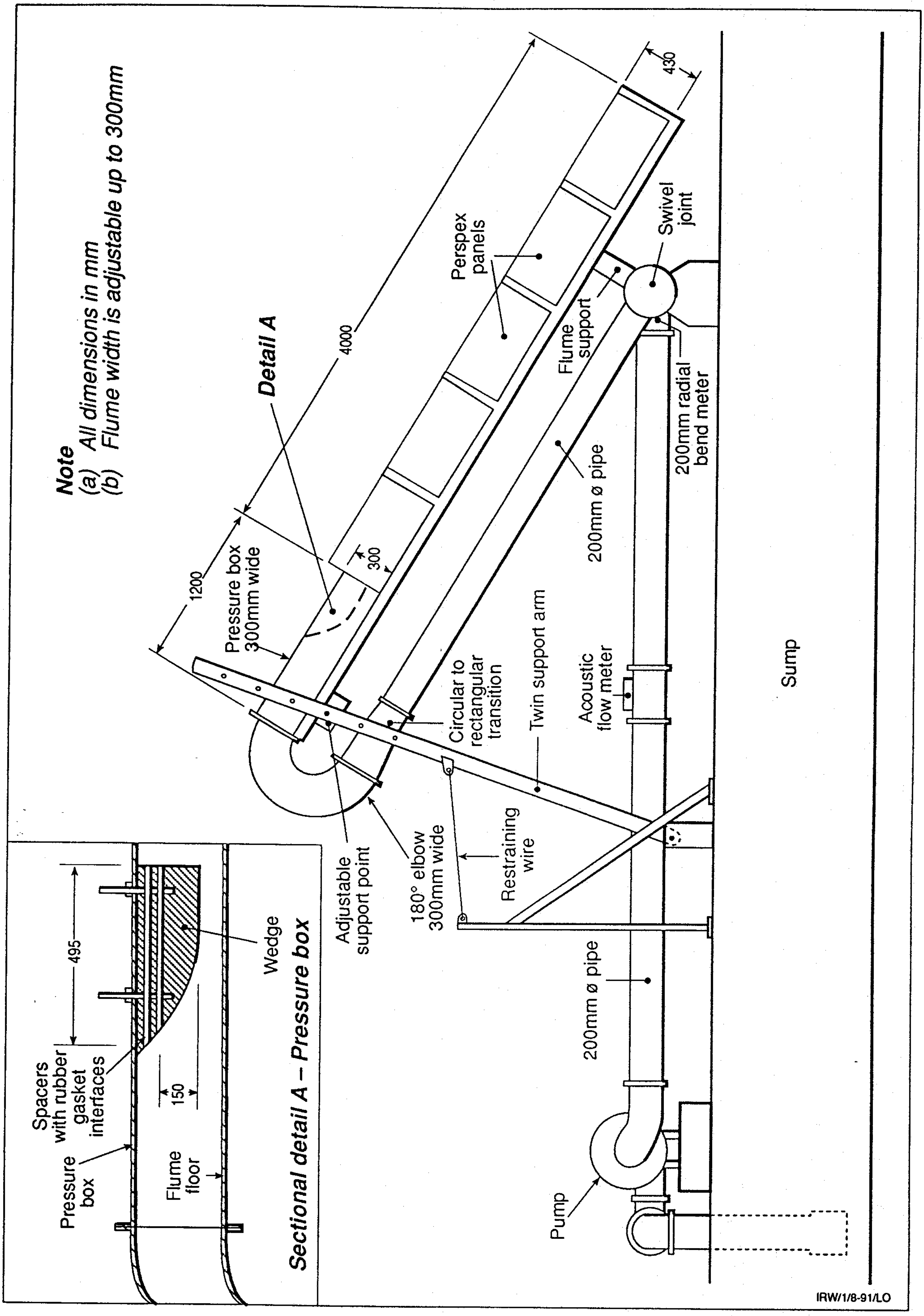
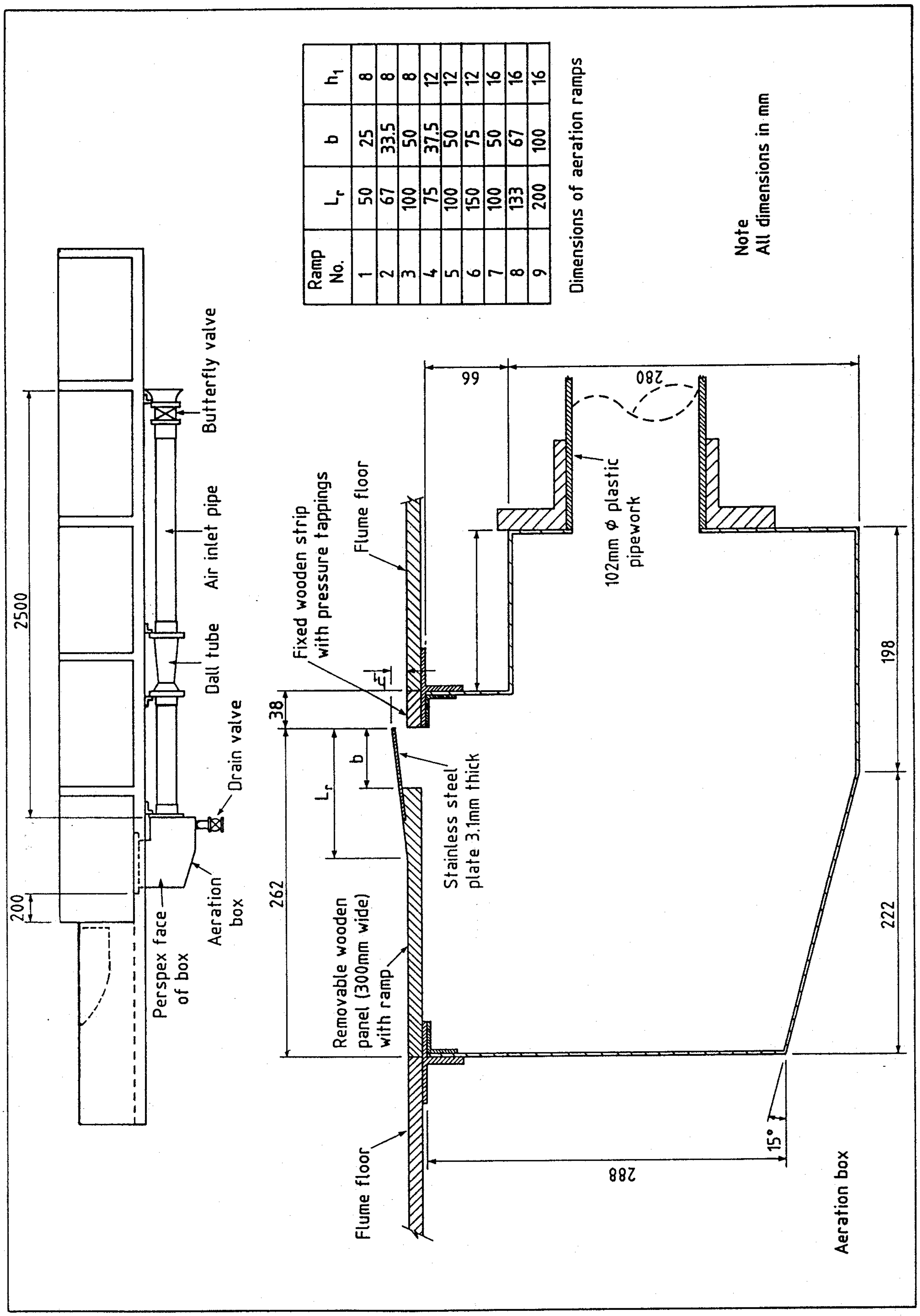


Fig 1   Layout of high velocity flume



Ramp No.	L_r	b	h_1
1	50	25	8
2	67	33.5	8
3	100	50	8
4	75	37.5	12
5	100	50	12
6	150	75	12
7	100	50	16
8	133	67	16
9	200	100	16

Fig 2 Air supply system and aeration ramps

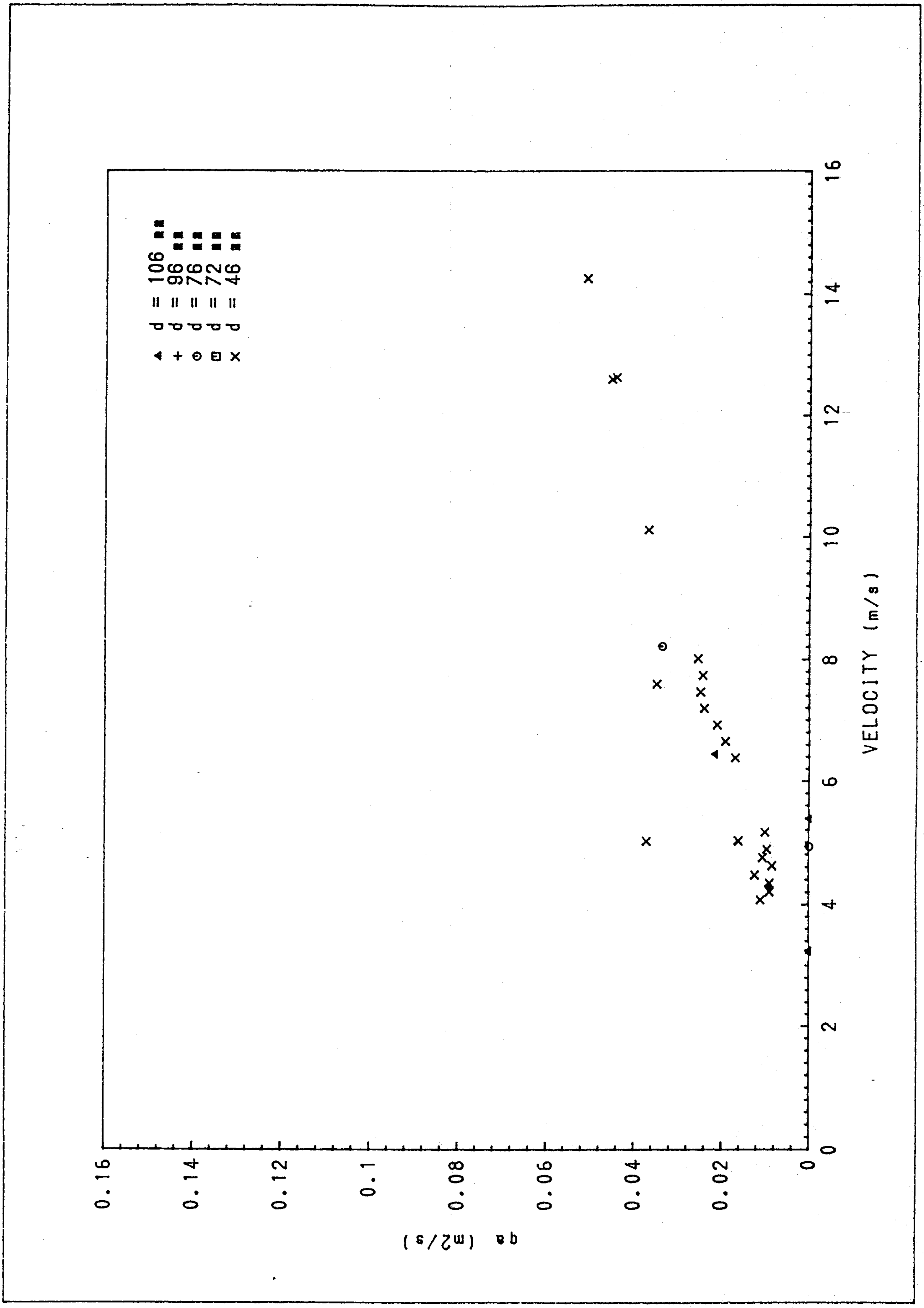


Fig 3 Air demand versus flow velocity : Aerator 1, air valve setting 0

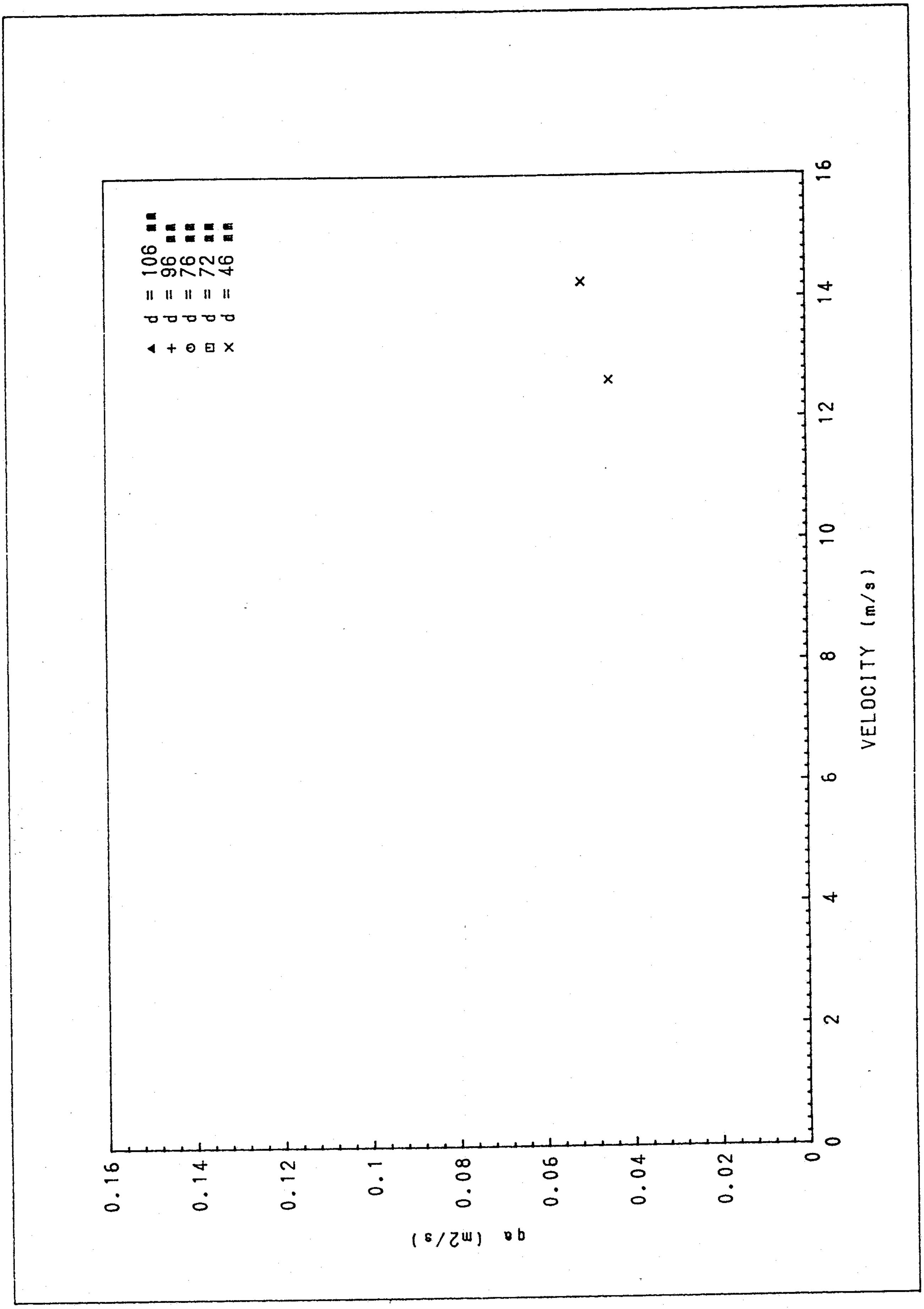


Fig 4 Air demand versus flow velocity : Aerator 1, air valve setting 2

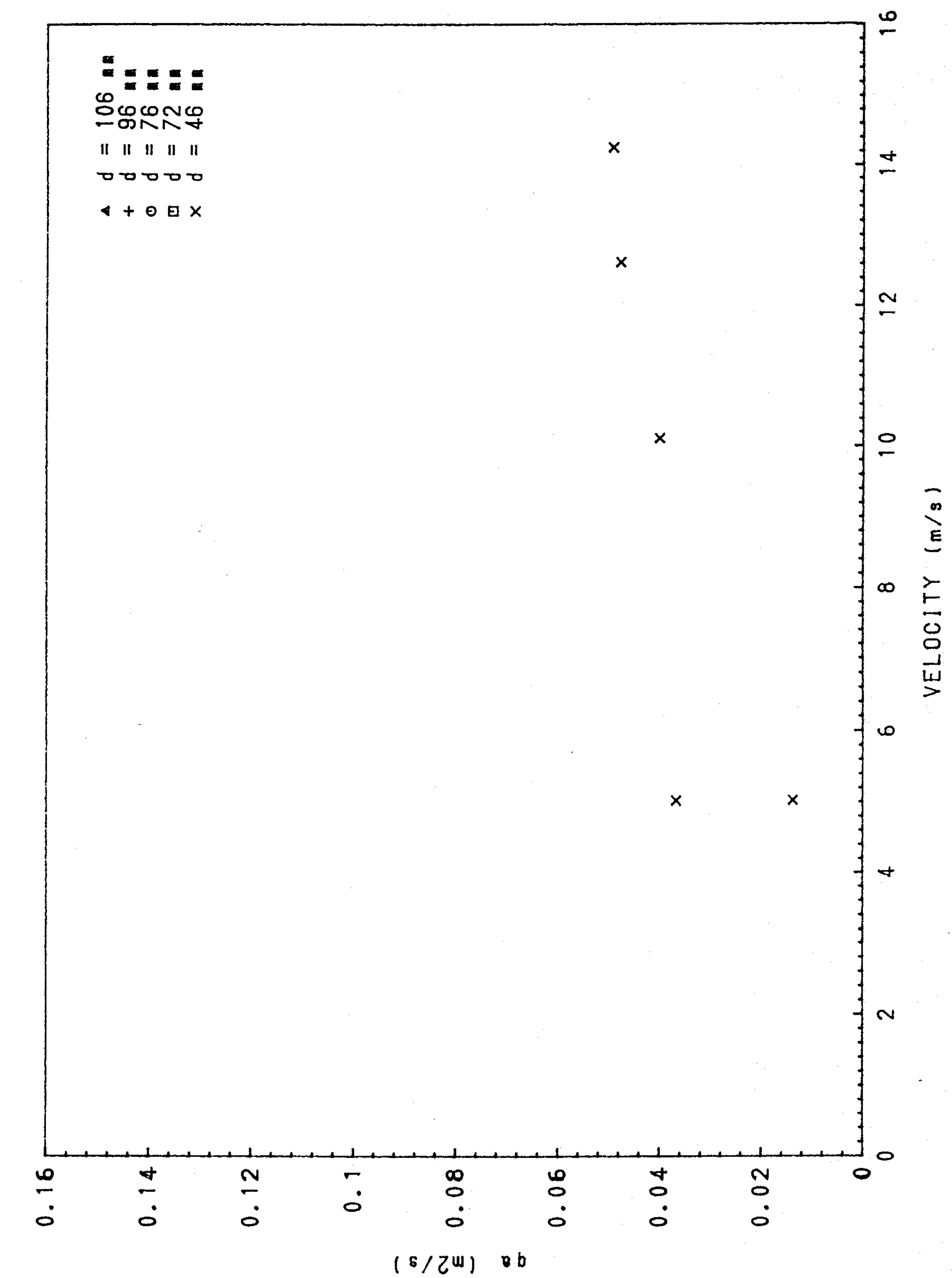


Fig 5 Air demand versus flow velocity : Aerator 1, air valve setting 3

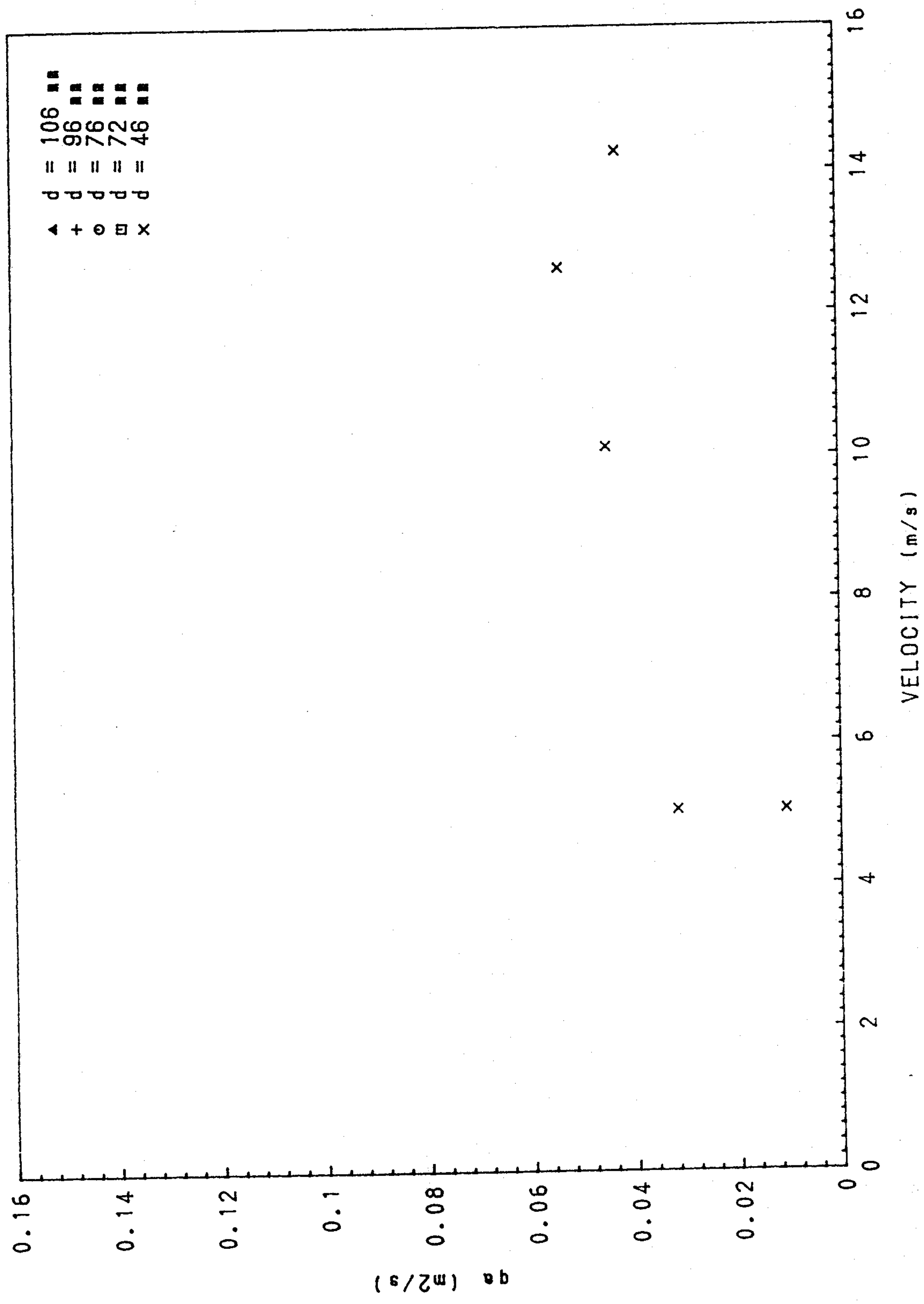


Fig 6 Air demand versus flow velocity : Aerator 1, air valve setting 4

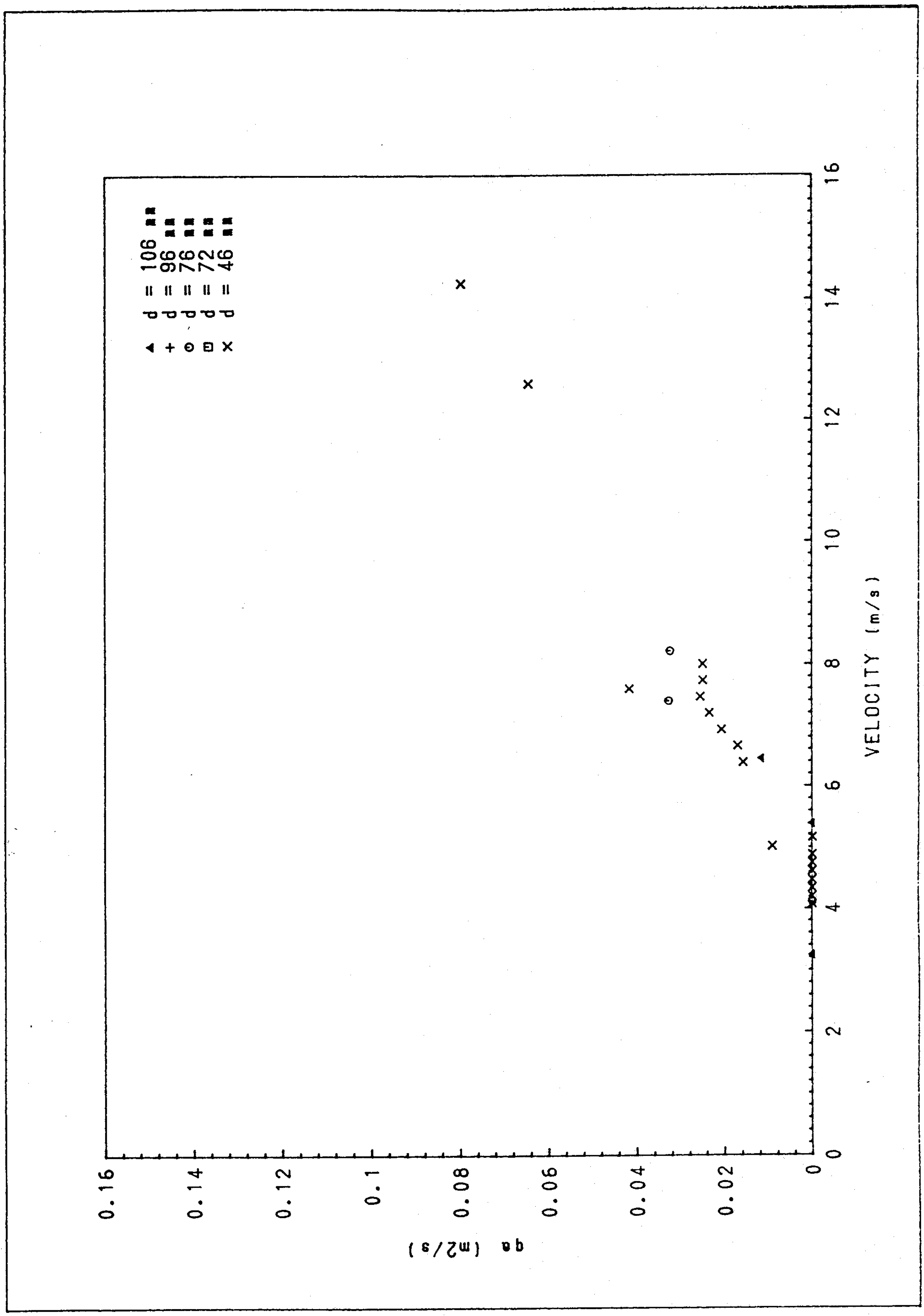


Fig 7 Air demand versus flow velocity : Aerator 3, air valve setting 0

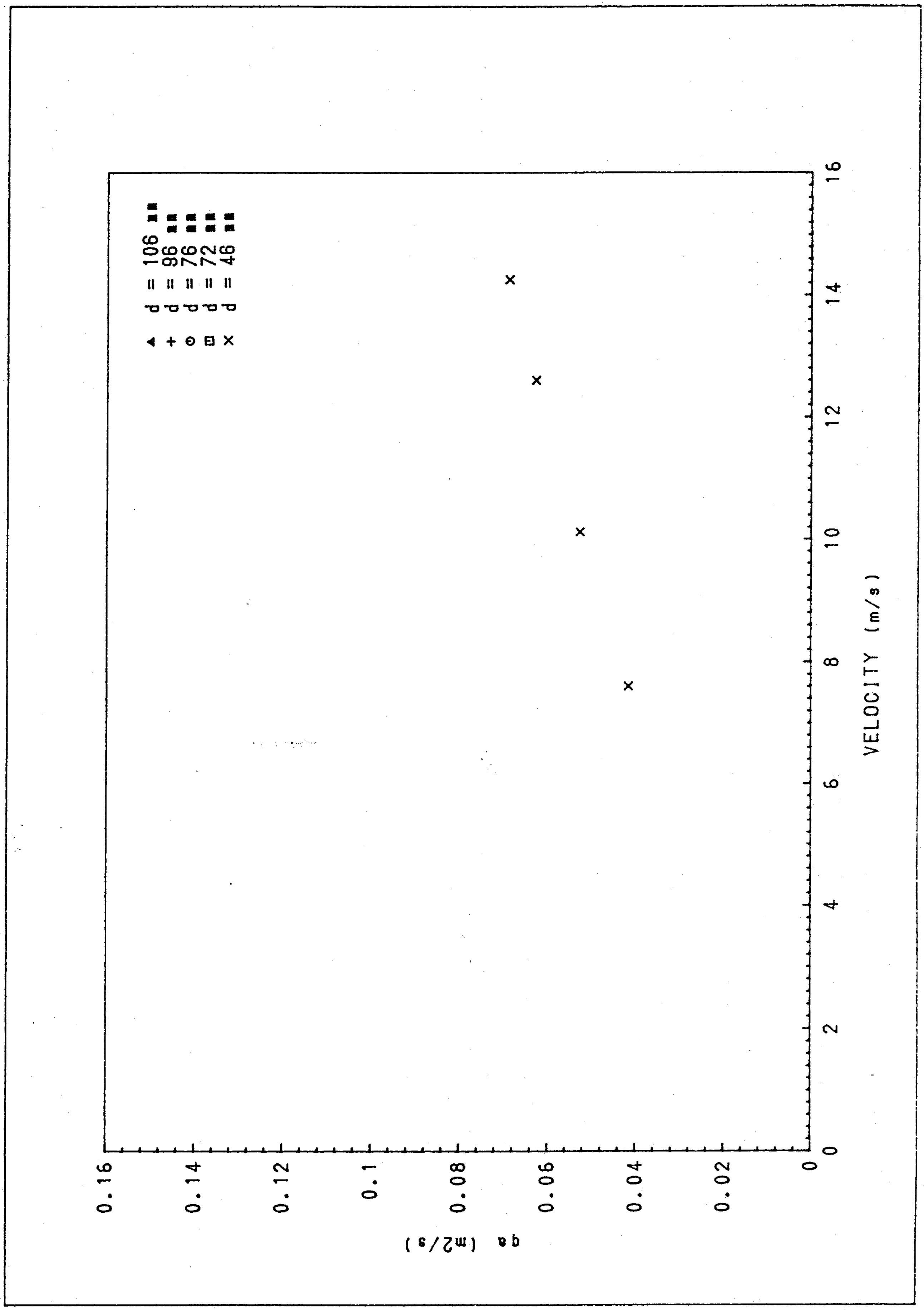


Fig 8 Air demand versus flow velocity : Aerator 3, air valve setting 2

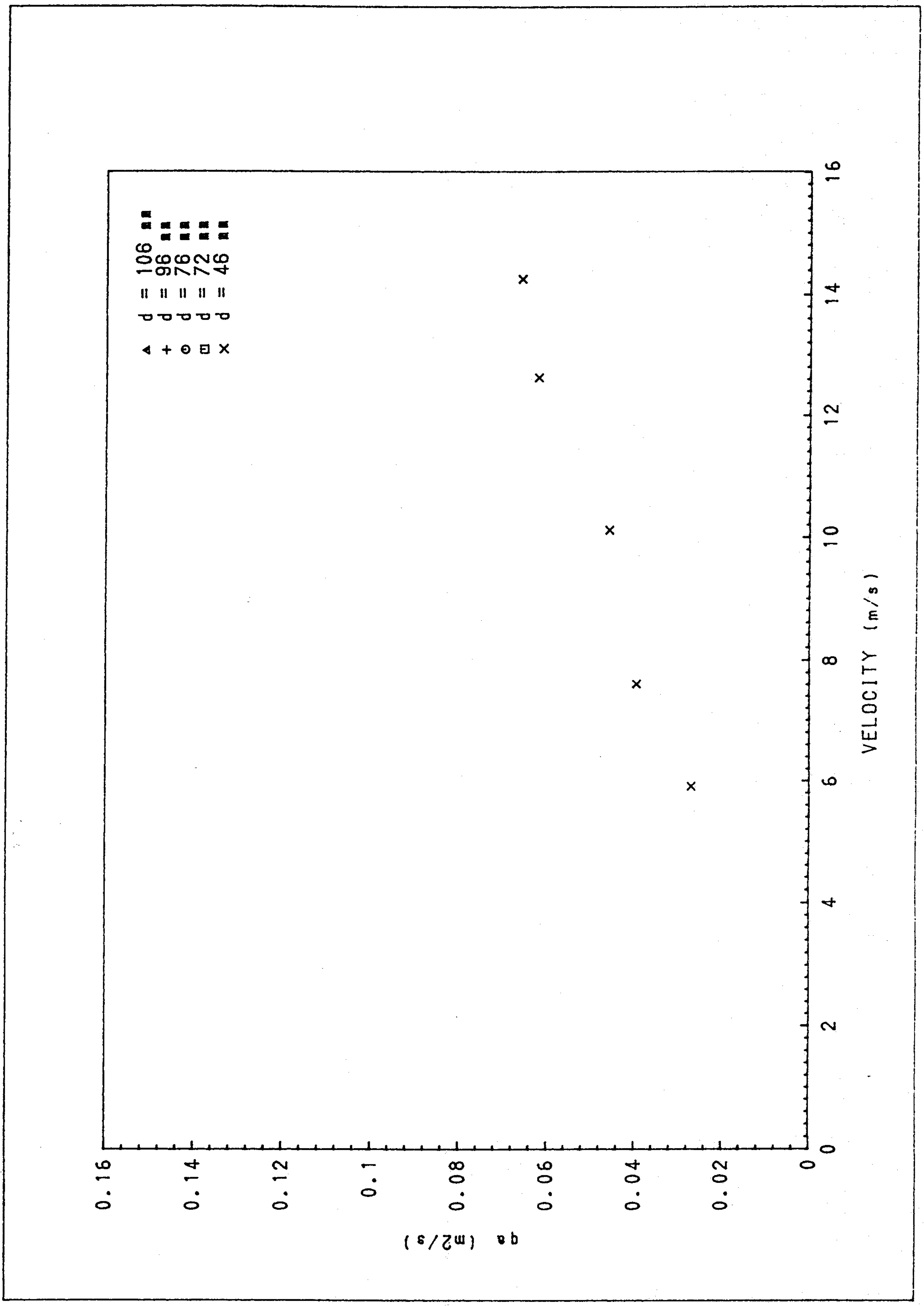


Fig 9 Air demand versus flow velocity : Aerator 3, air valve setting 3

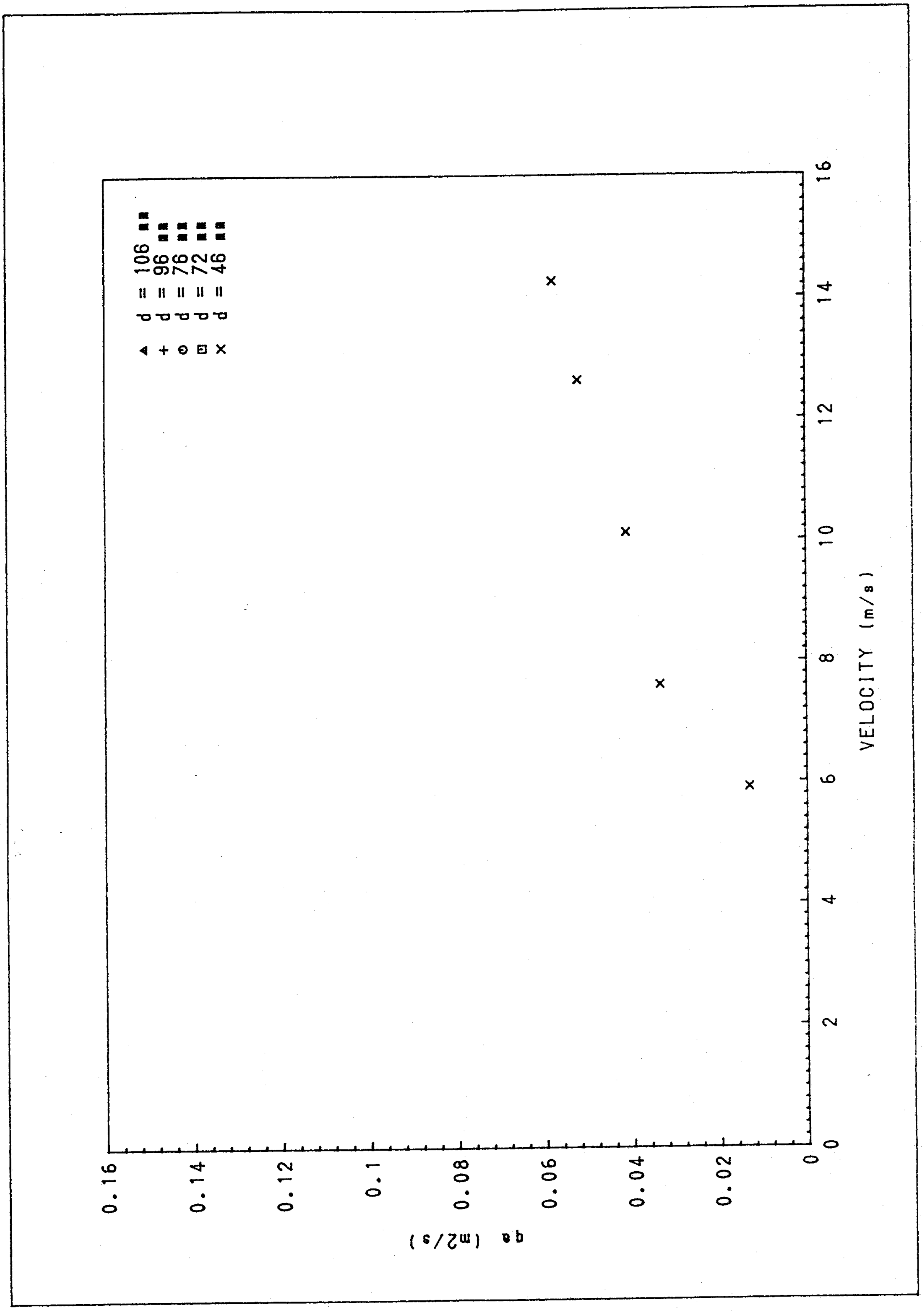


Fig 10 Air demand versus flow velocity : Aerator 3 air valve setting 4

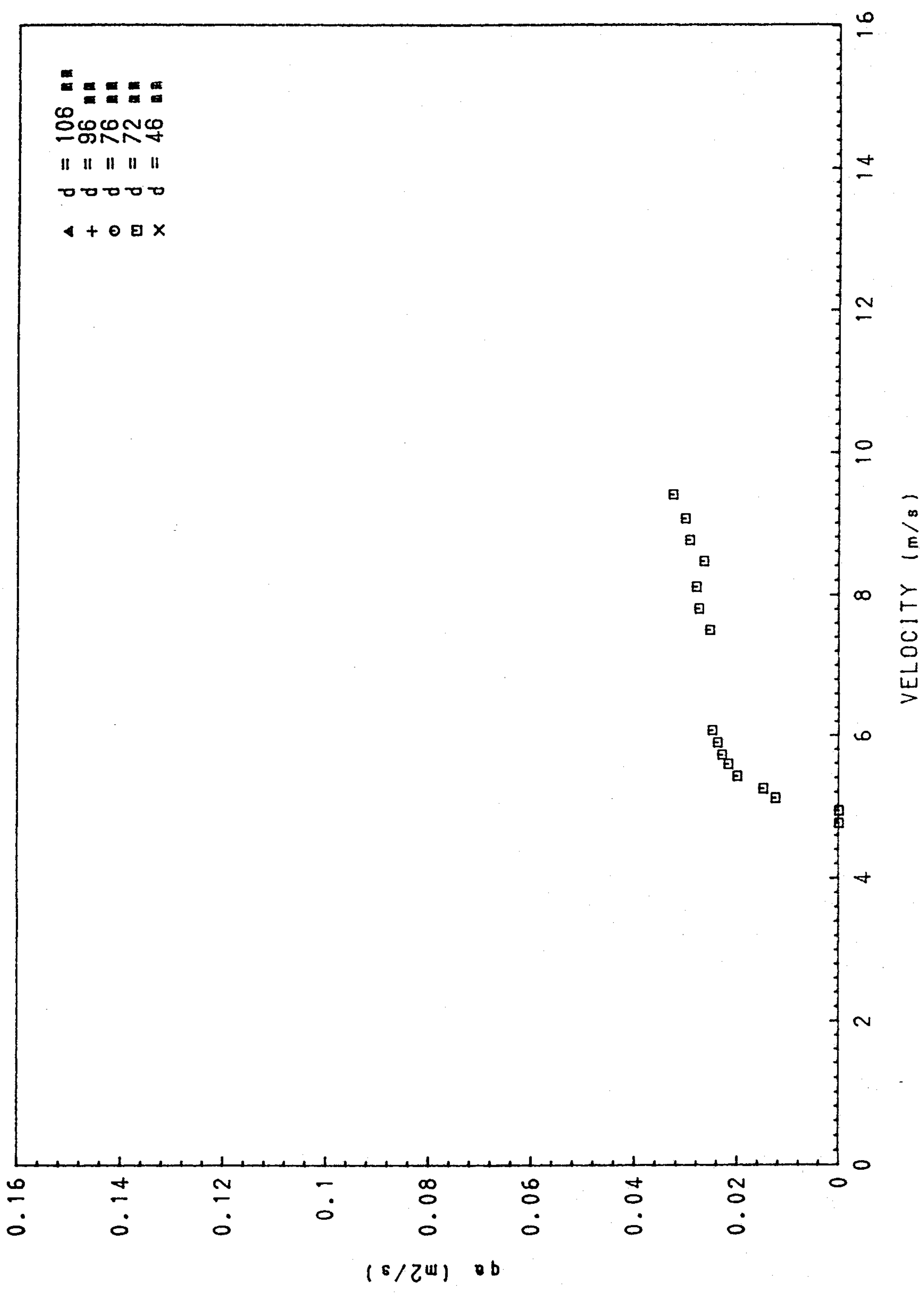


Fig 11 Air demand versus flow velocity : Aerator 4 air valve setting 0

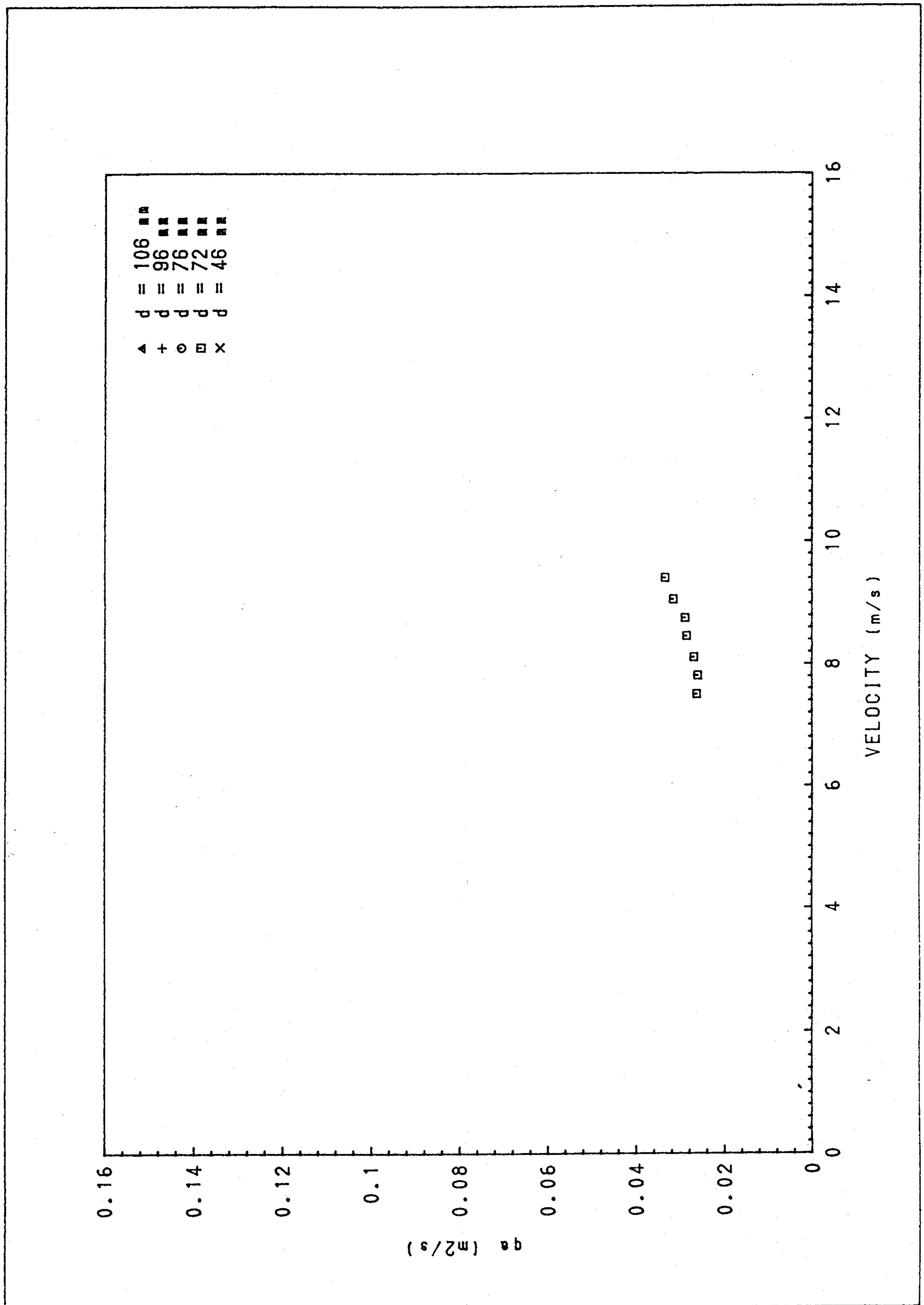


Fig 12   Air demand versus flow velocity : Aerator 6 air valve setting 0

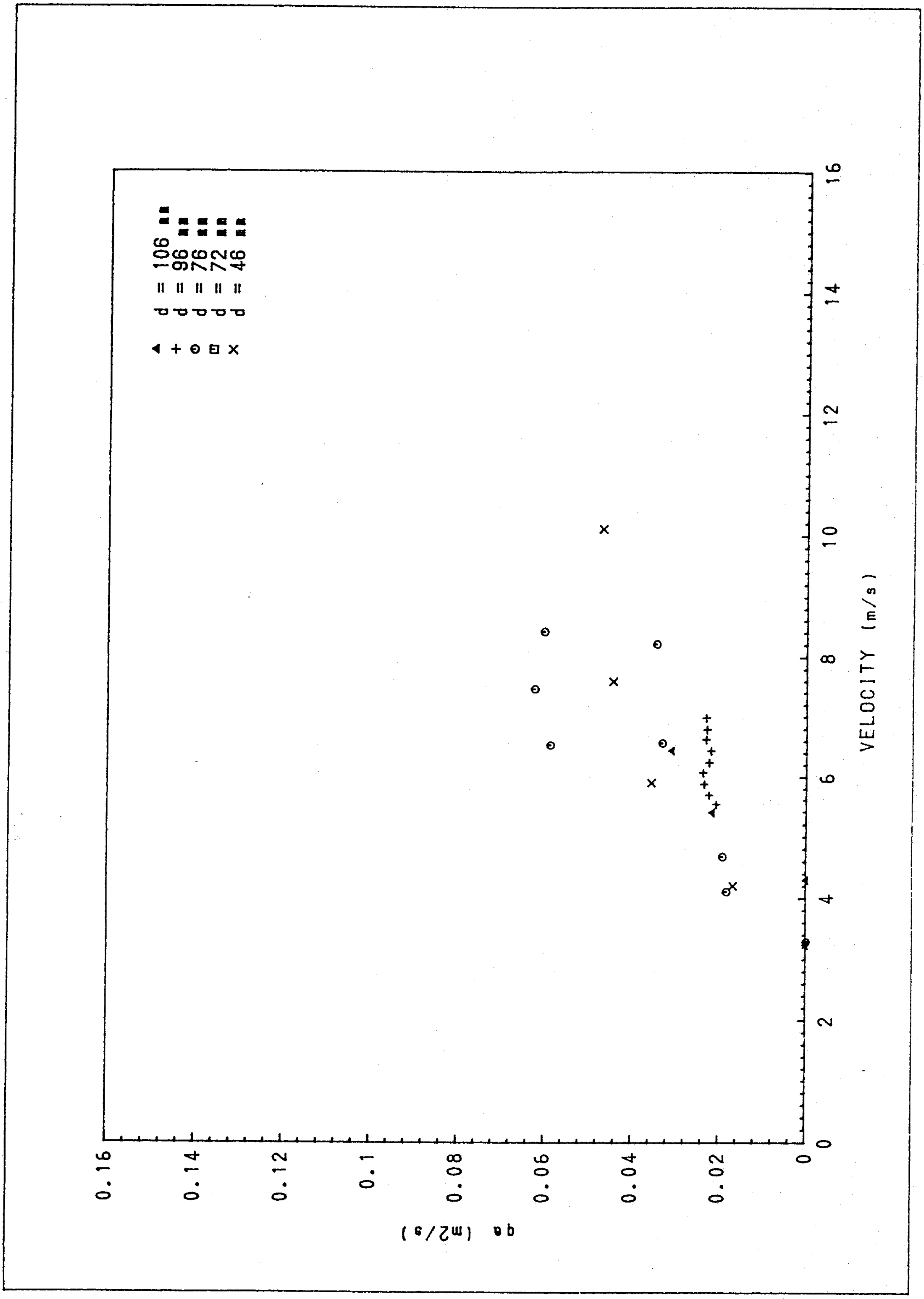


Fig 13 Air demand versus flow velocity : Aerator 7 air valve setting 0

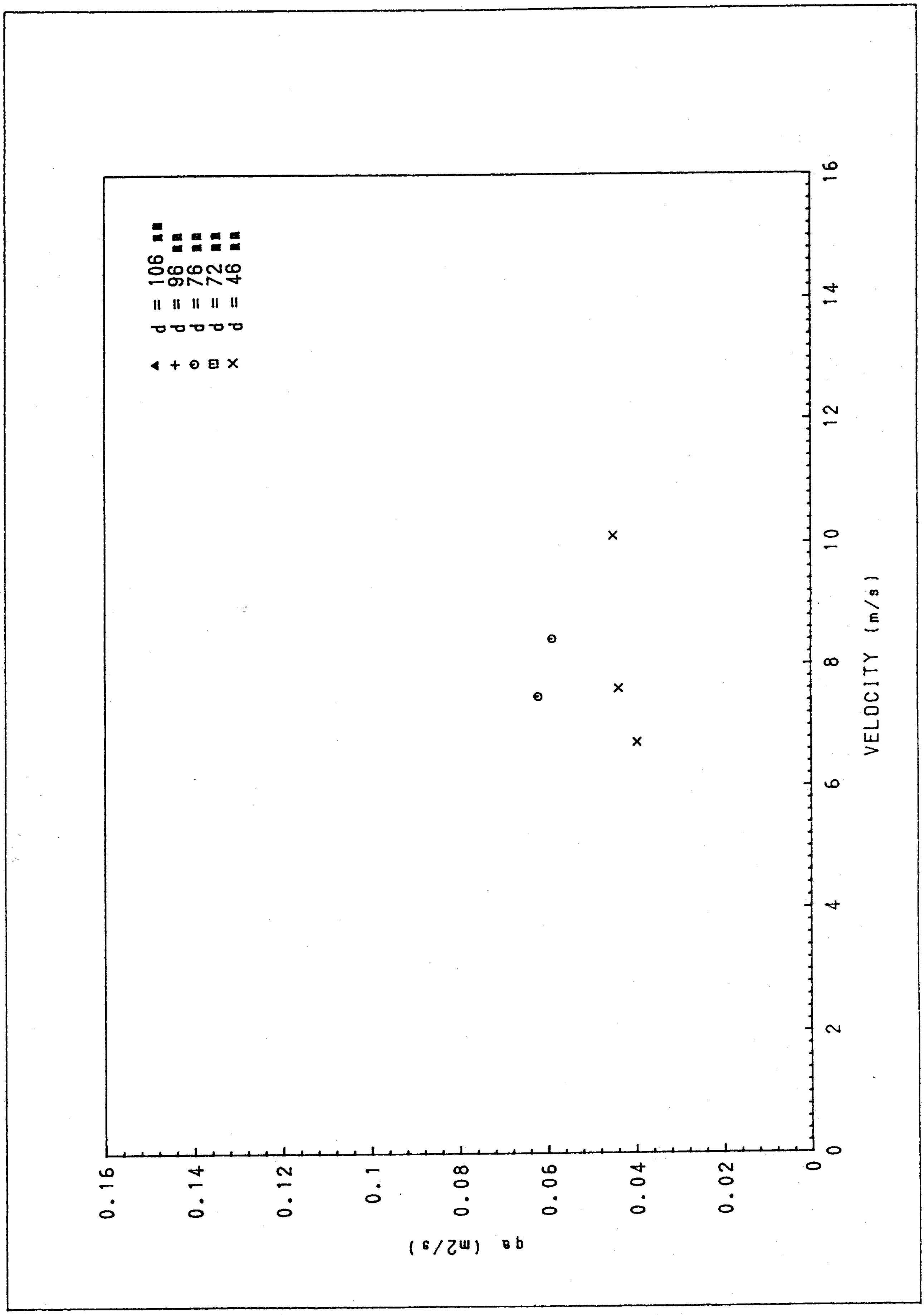


Fig 14   Air demand versus flow velocity : Aerator 7   air valve setting 2

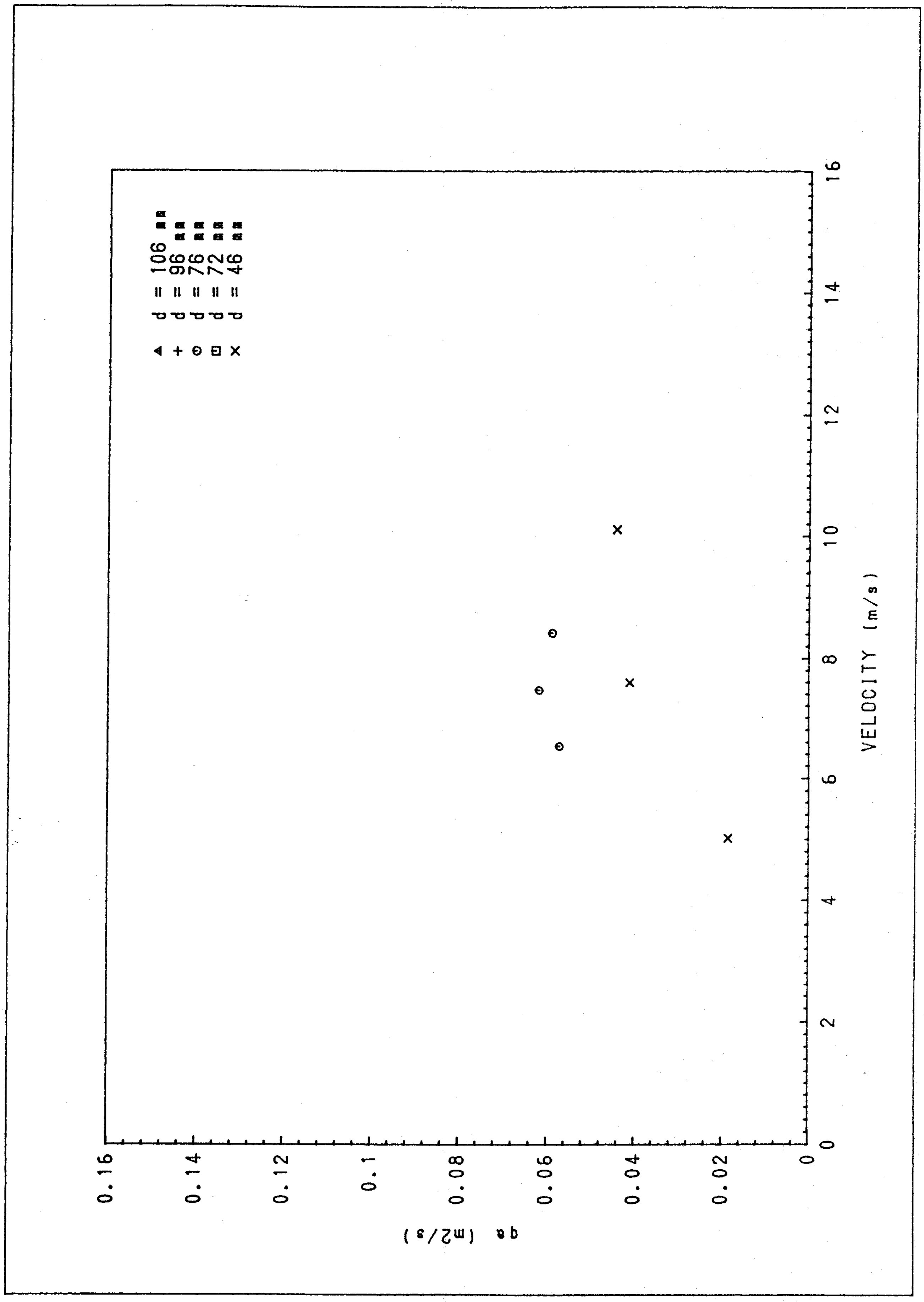


Fig 15 Air demand versus flow velocity : Aerator 7 air valve setting 3

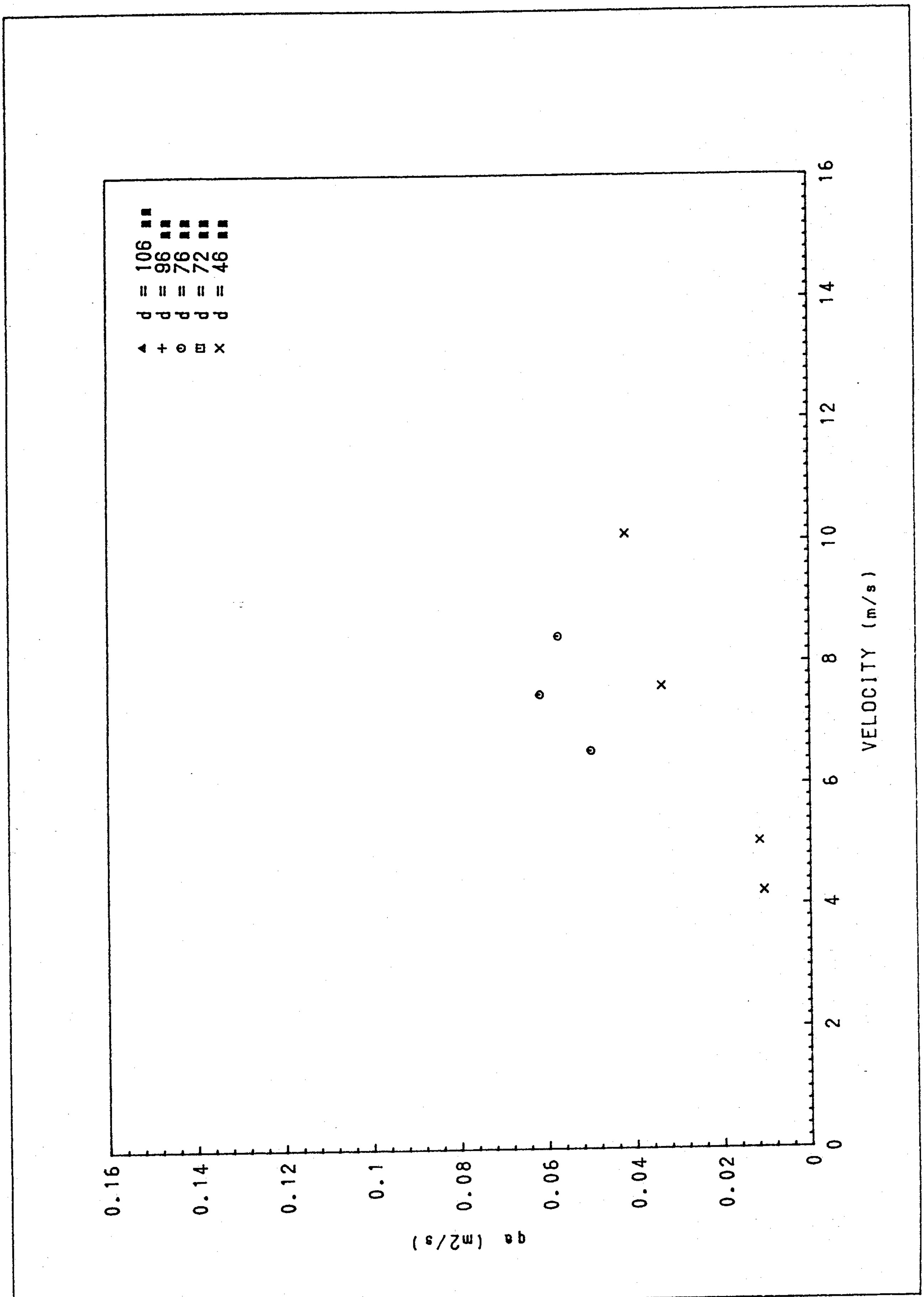


Fig 16 Air demand versus flow velocity : Aerator 7 air valve setting 4

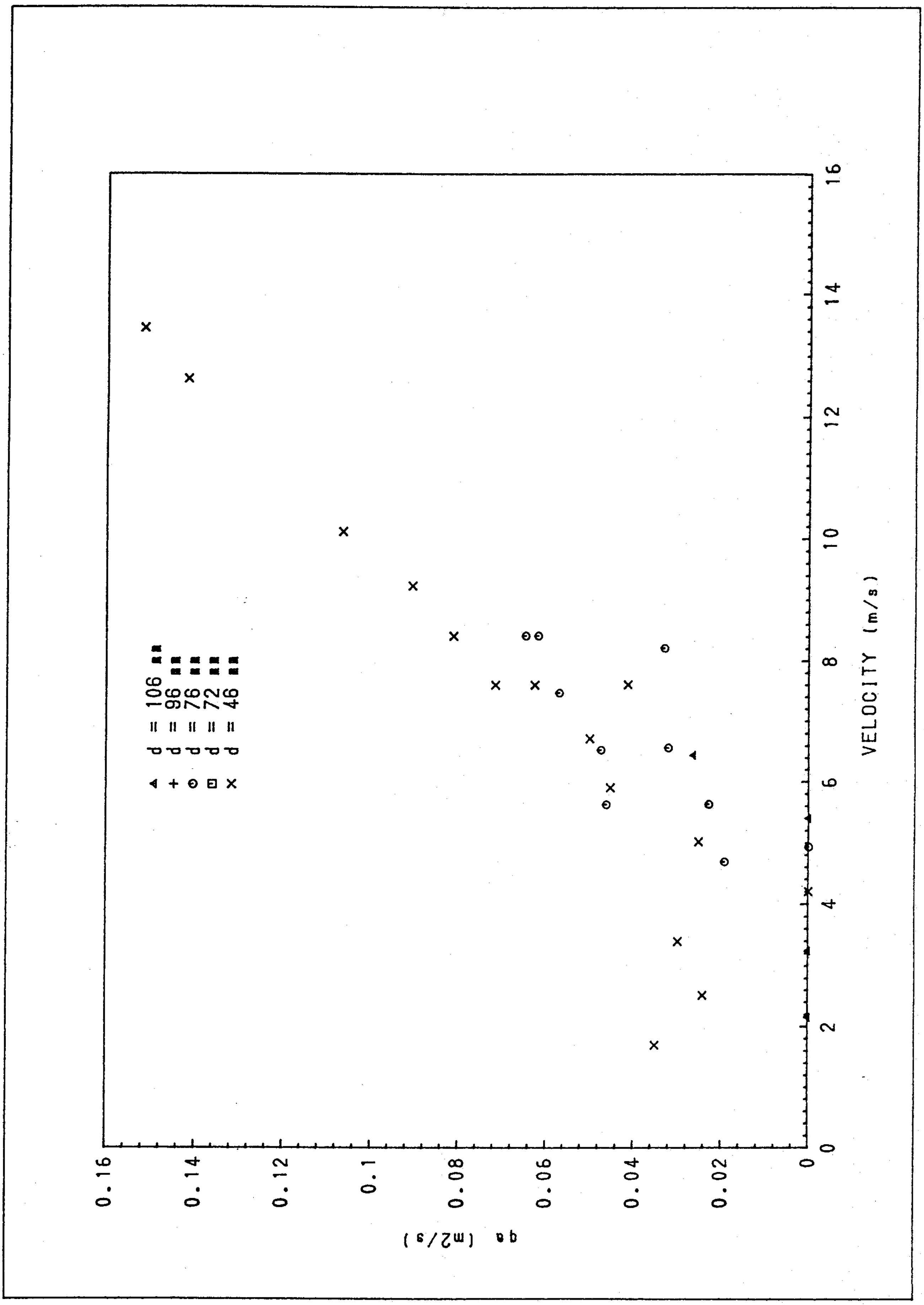


Fig 17 Air demand versus flow velocity : Aerator 9 air valve setting 0

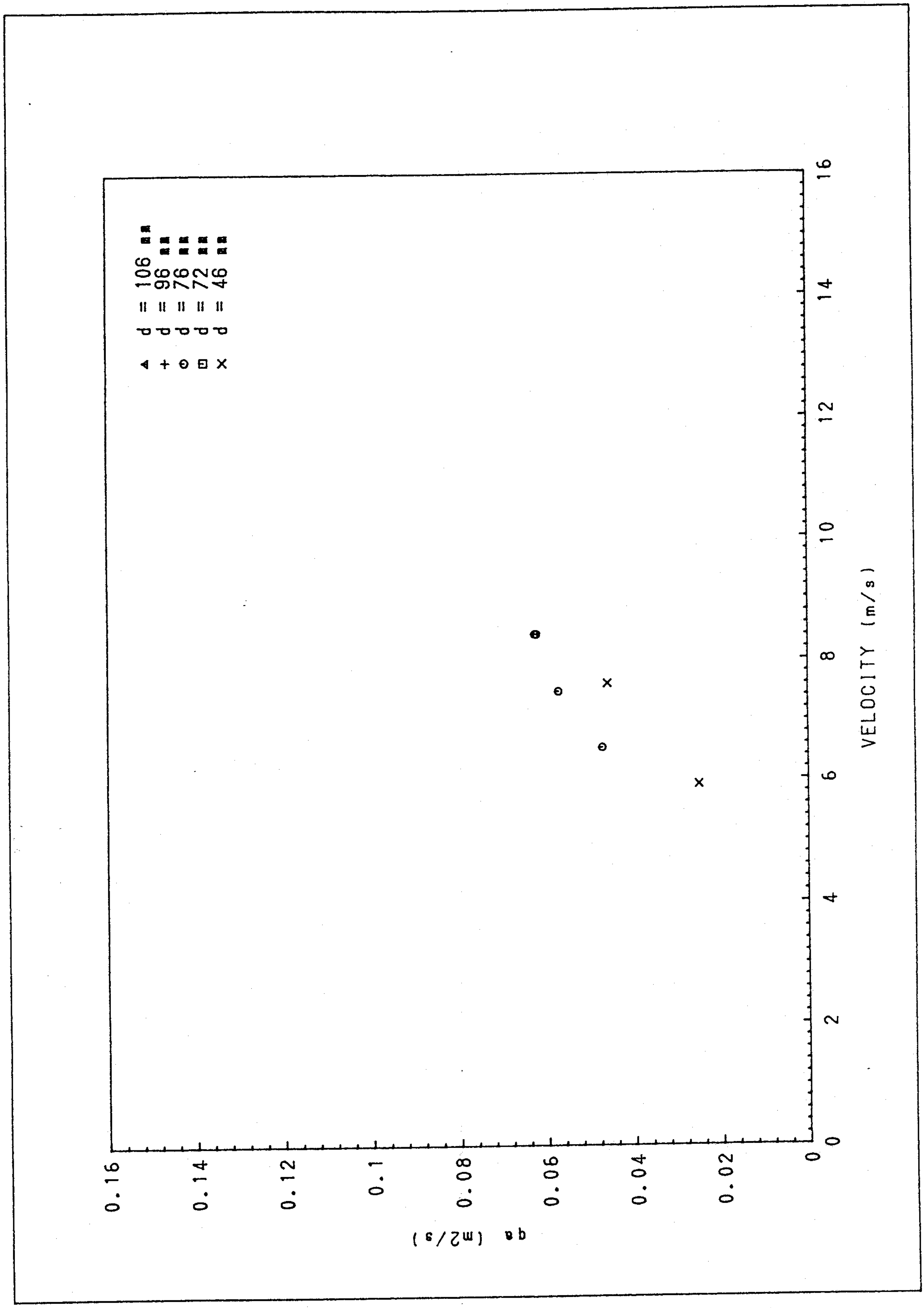


Fig 18 Air demand versus flow velocity : Aerator 9 air valve setting 2

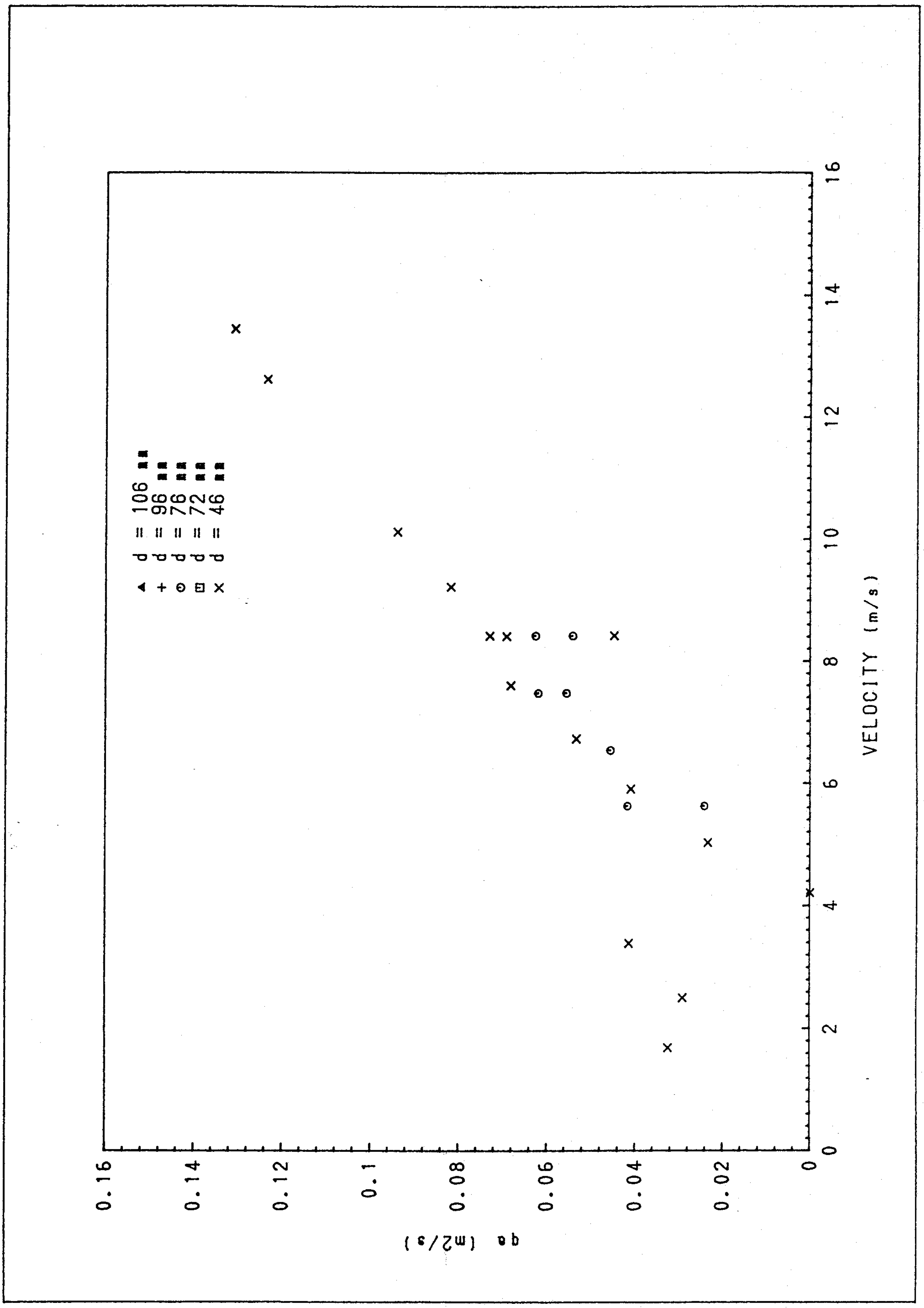


Fig 19 Air demand versus flow velocity : Aerator 9 air valve setting 3

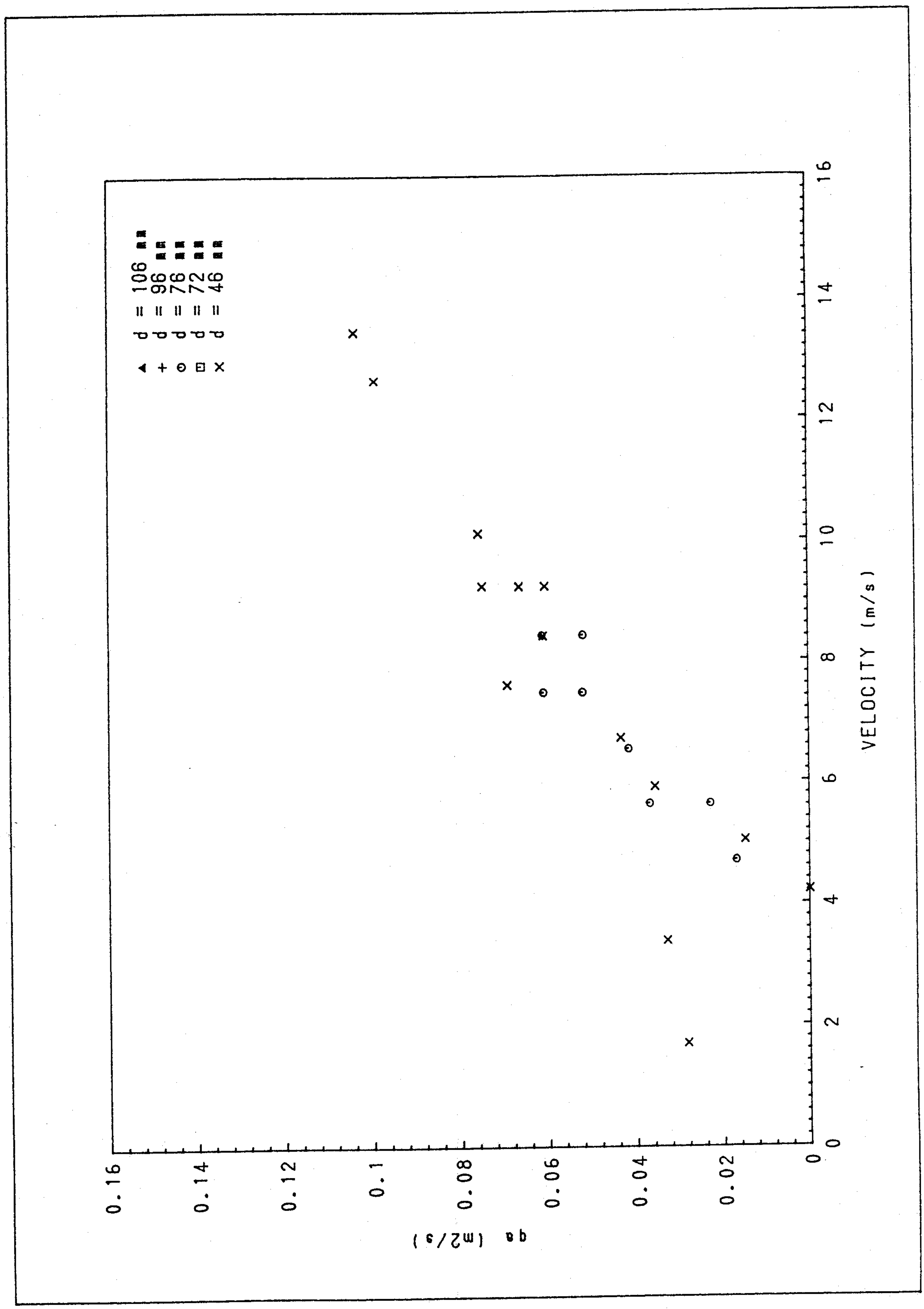


Fig 20 Air demand versus flow velocity : Aerator 9, air valve setting 4

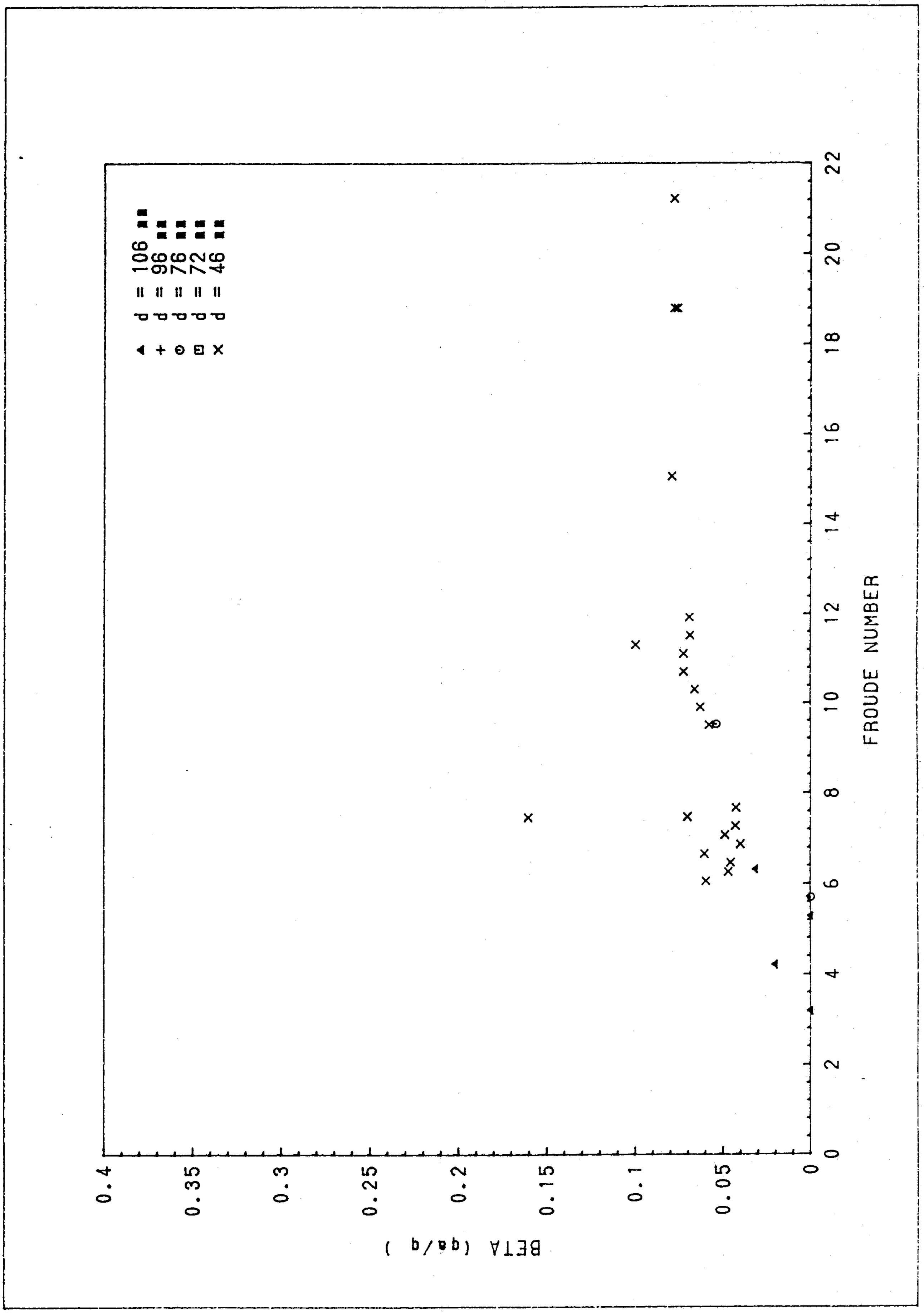


Fig 21 Air demand ratio versus Froude number : Aerator 1, air valve setting 0

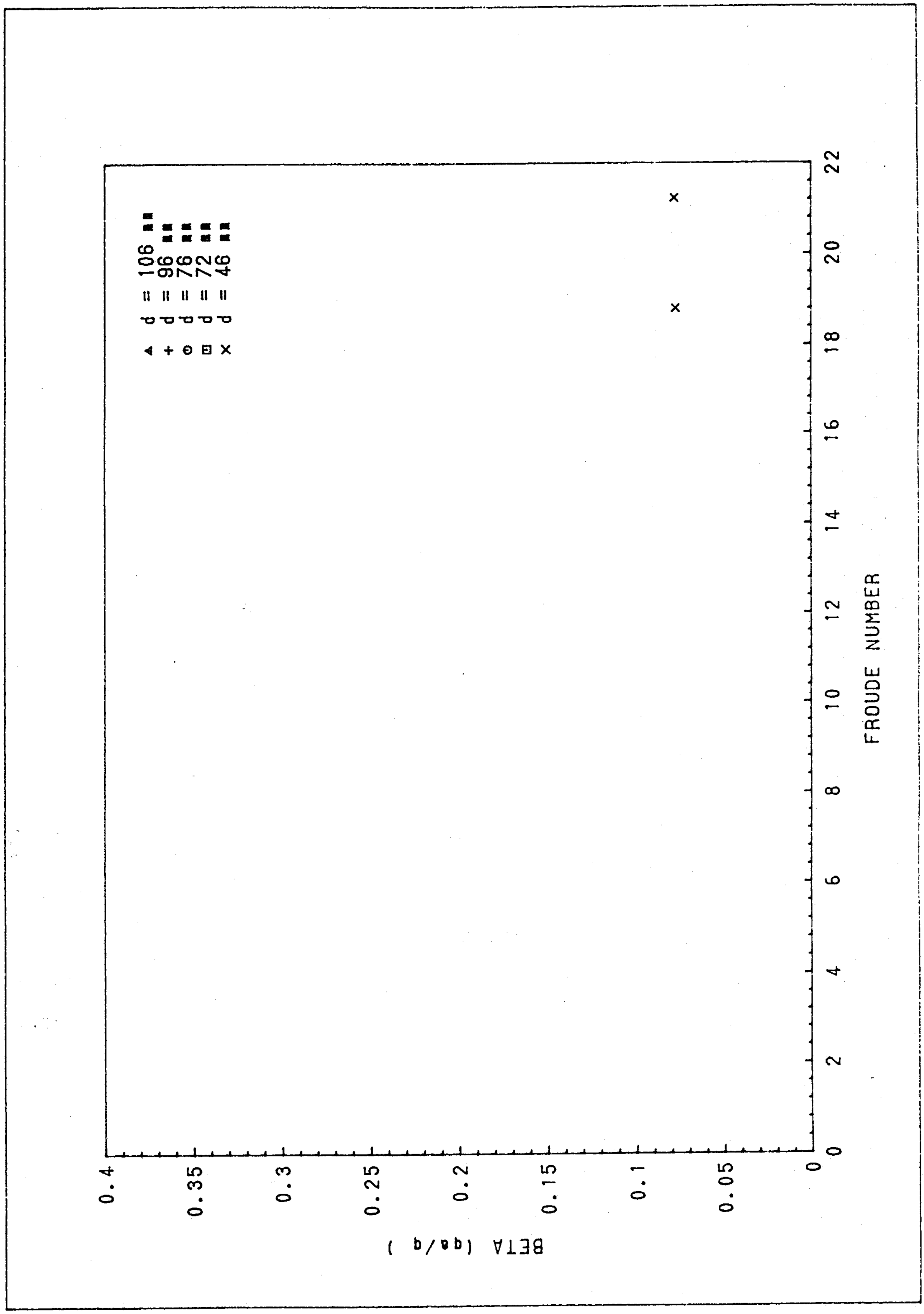


Fig 22 Air demand ratio versus Froude number : Aerator 1, air valve setting 2

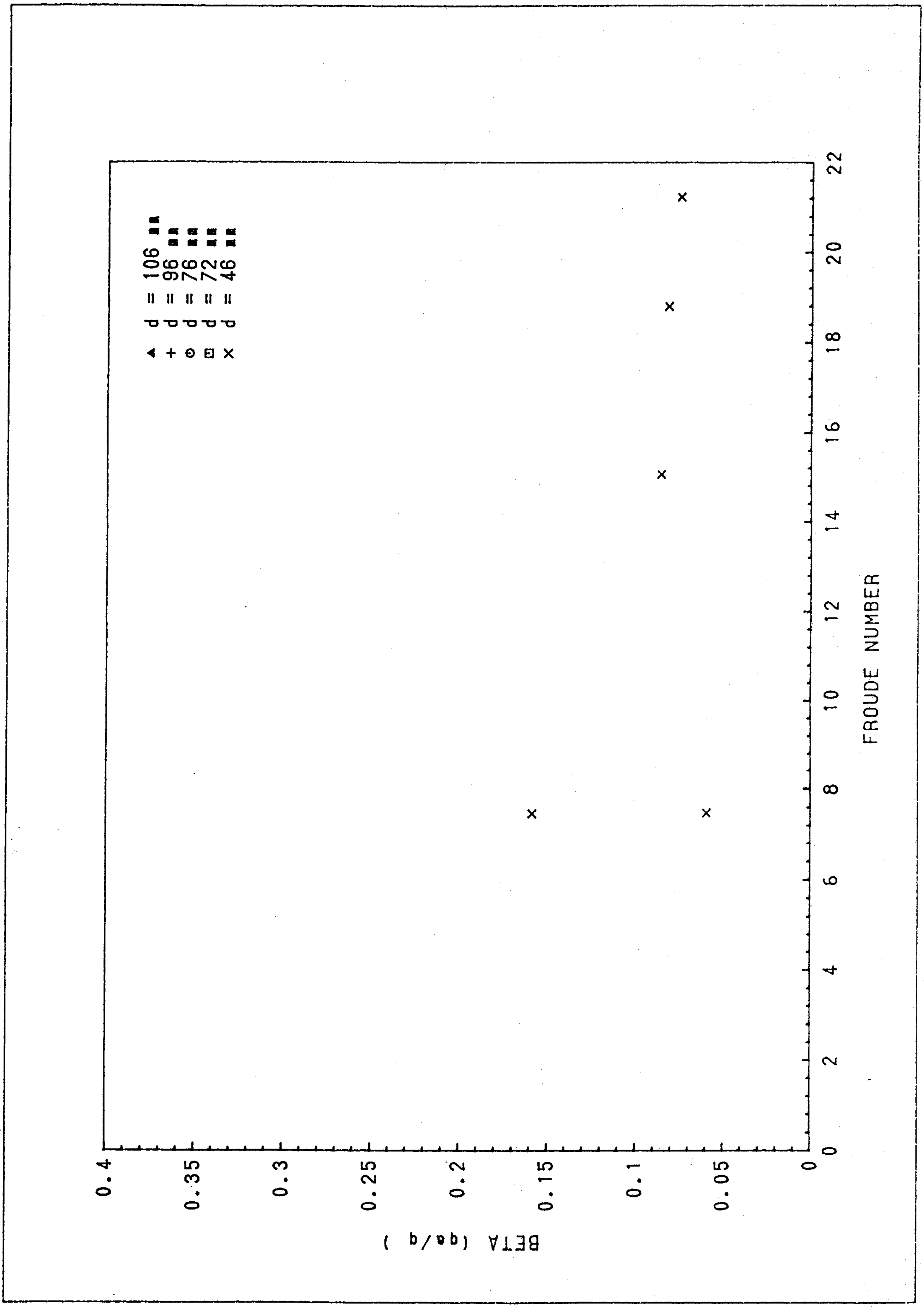


Fig 23 Air demand ratio versus Froude number : Aerator 1, air valve setting 3

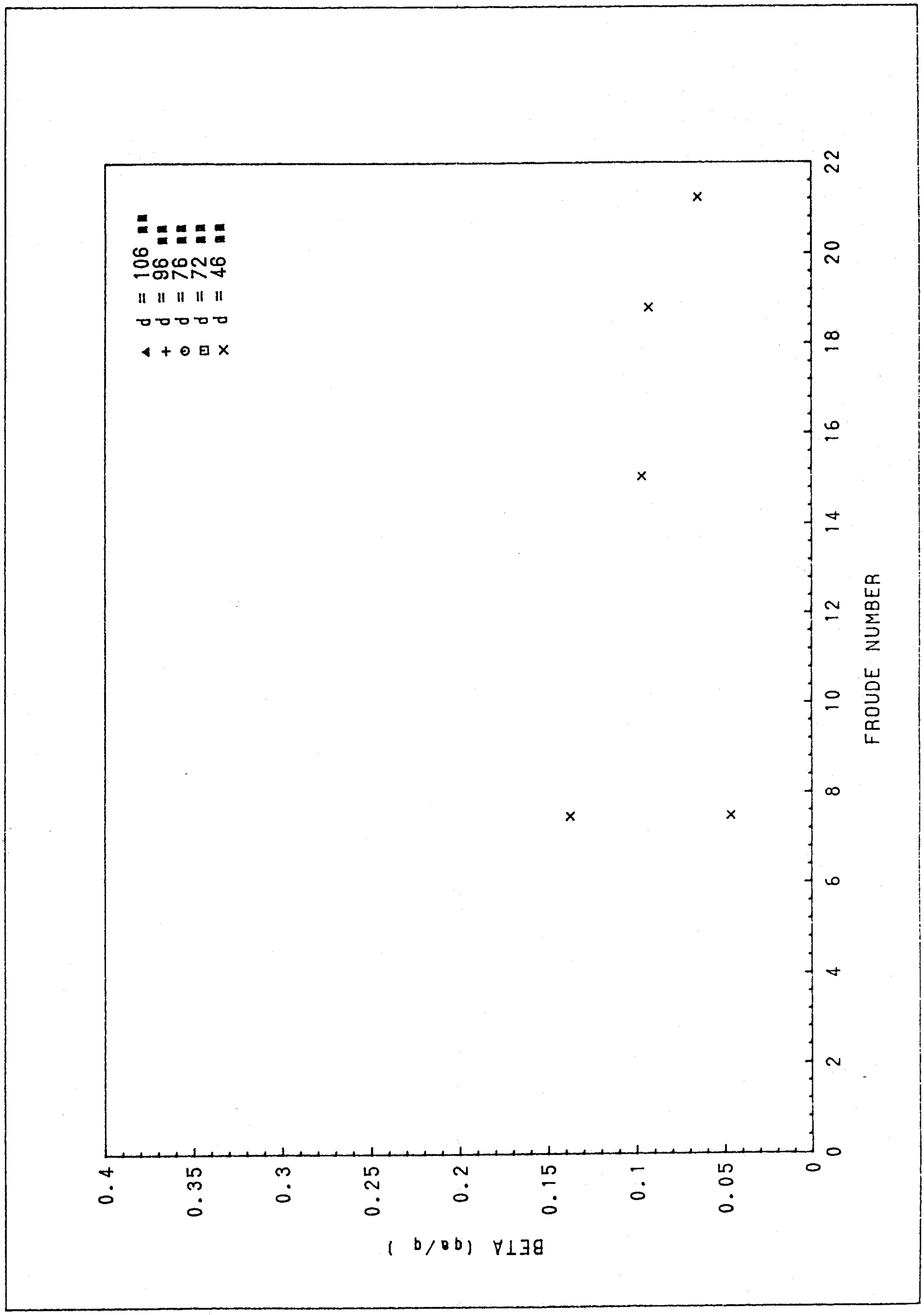


Fig 24 Air demand ratio versus Froude number : Aerator 1, air valve setting 4

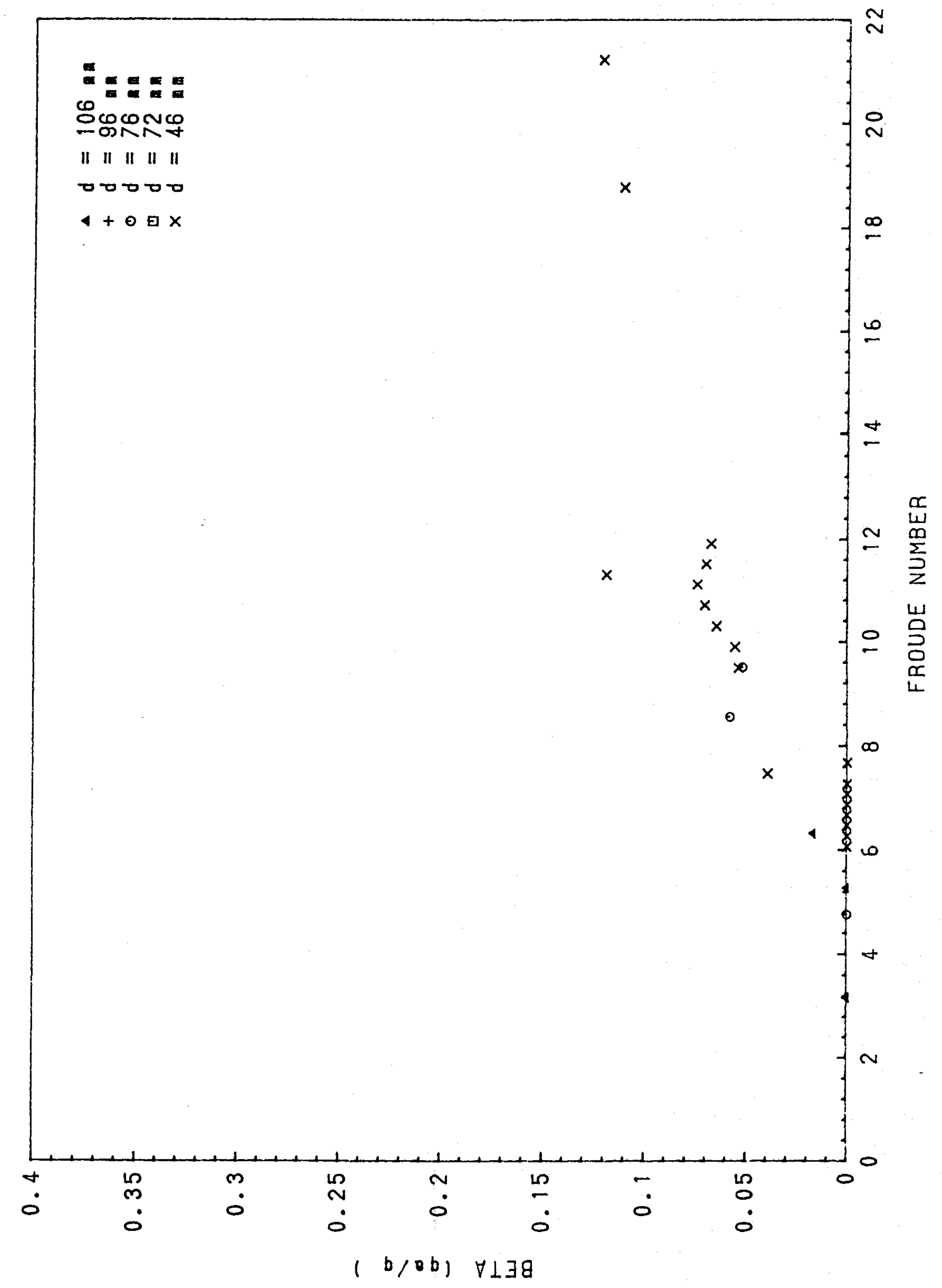


Fig 25 Air demand ratio versus Froude number : Aerator 3, air valve setting 0

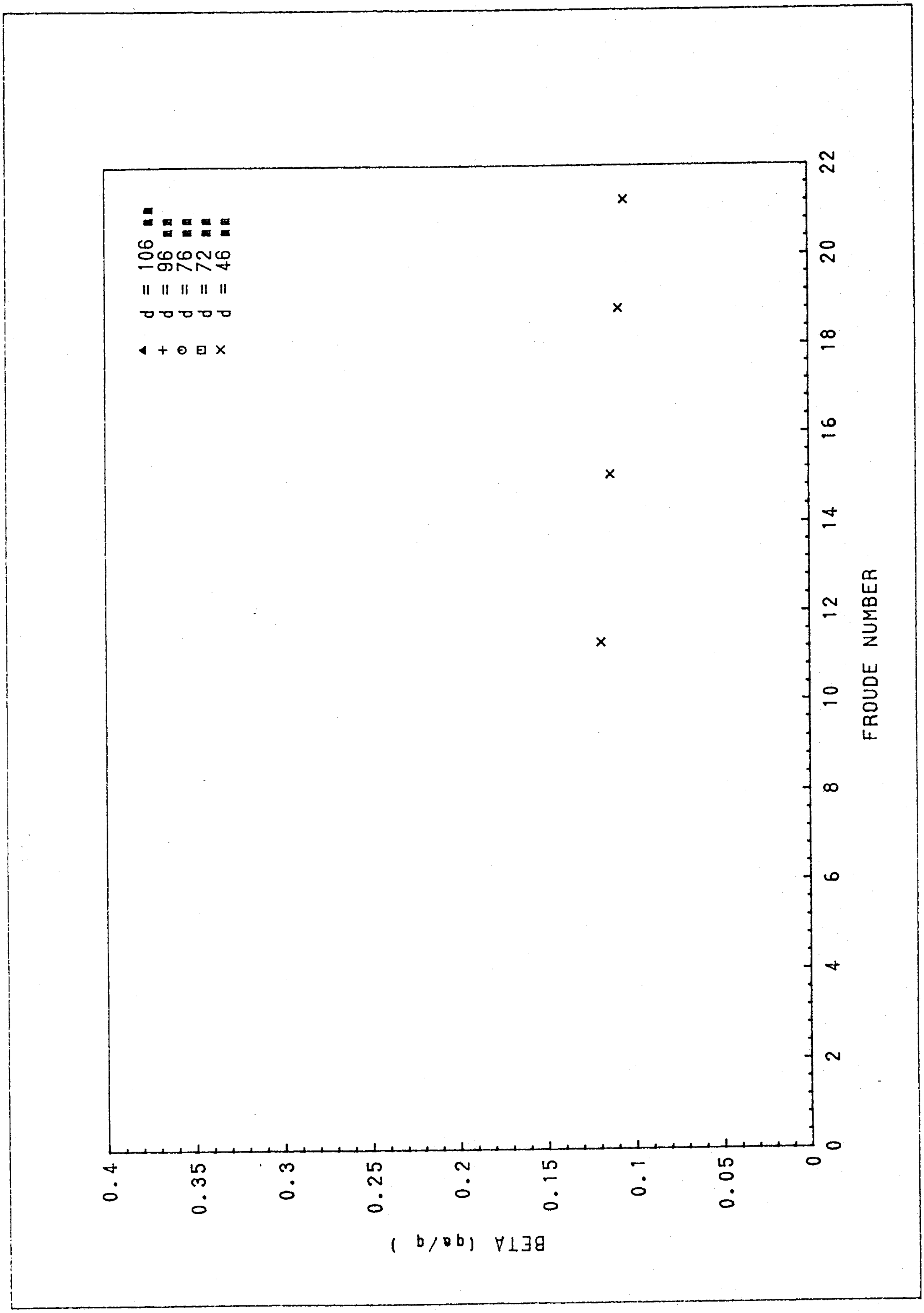


Fig 26 Air demand ratio versus Froude number : Aerator 3, air valve setting 2

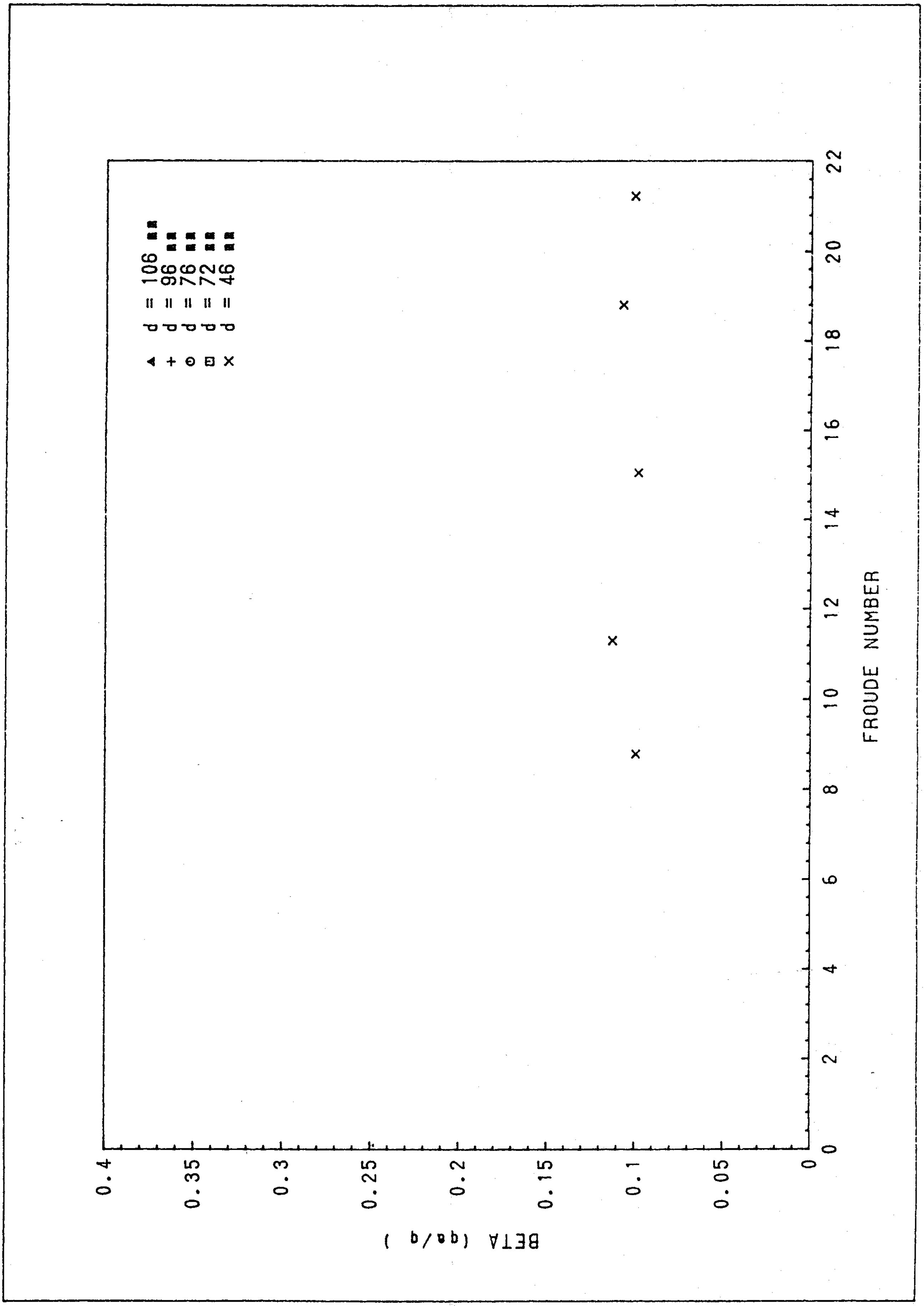


Fig 27 Air demand ratio versus Froude number : Aerator 3, air valve setting 3

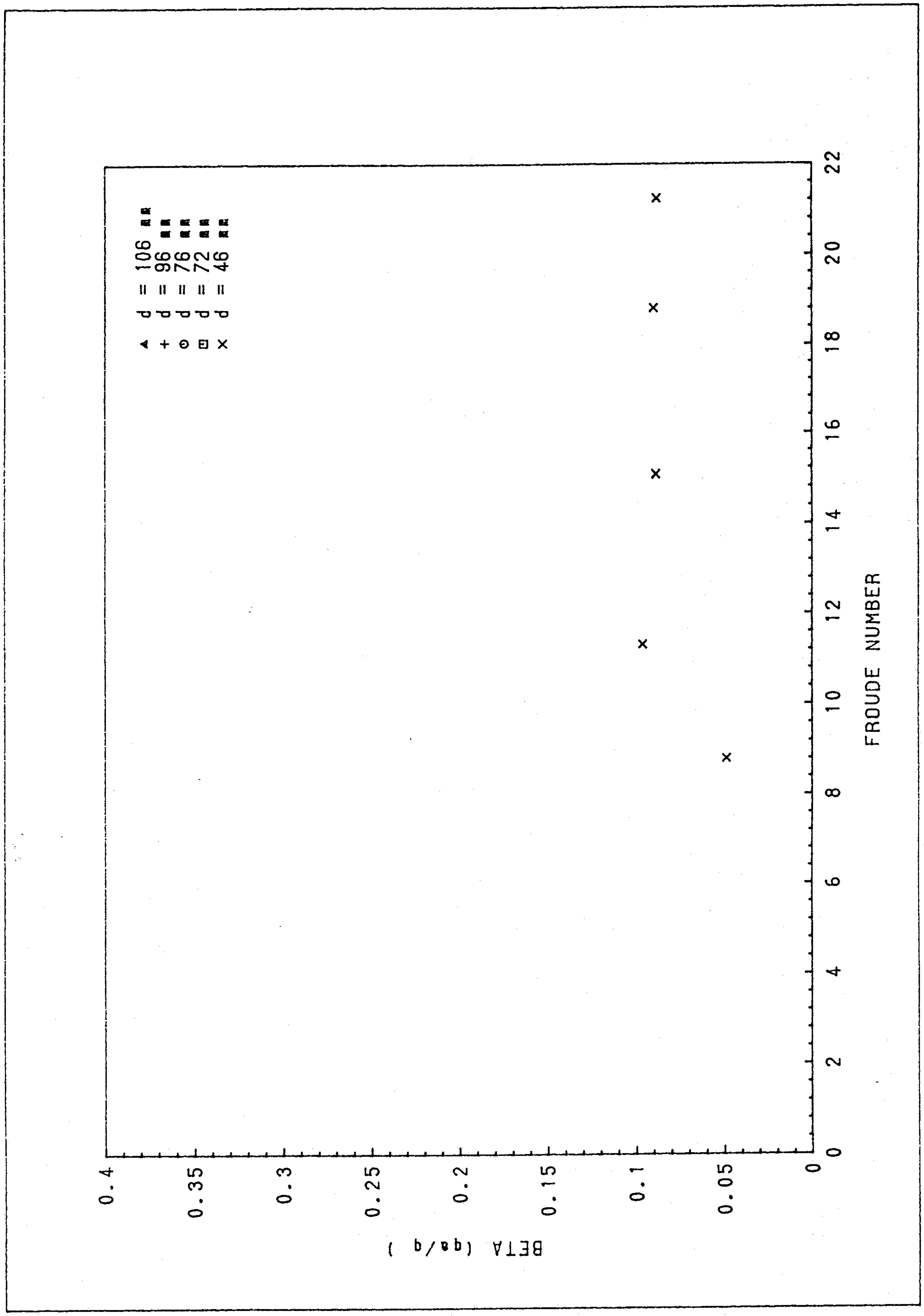


Fig 28 Air demand ratio versus Froude number : Aerator 3, air valve setting 4

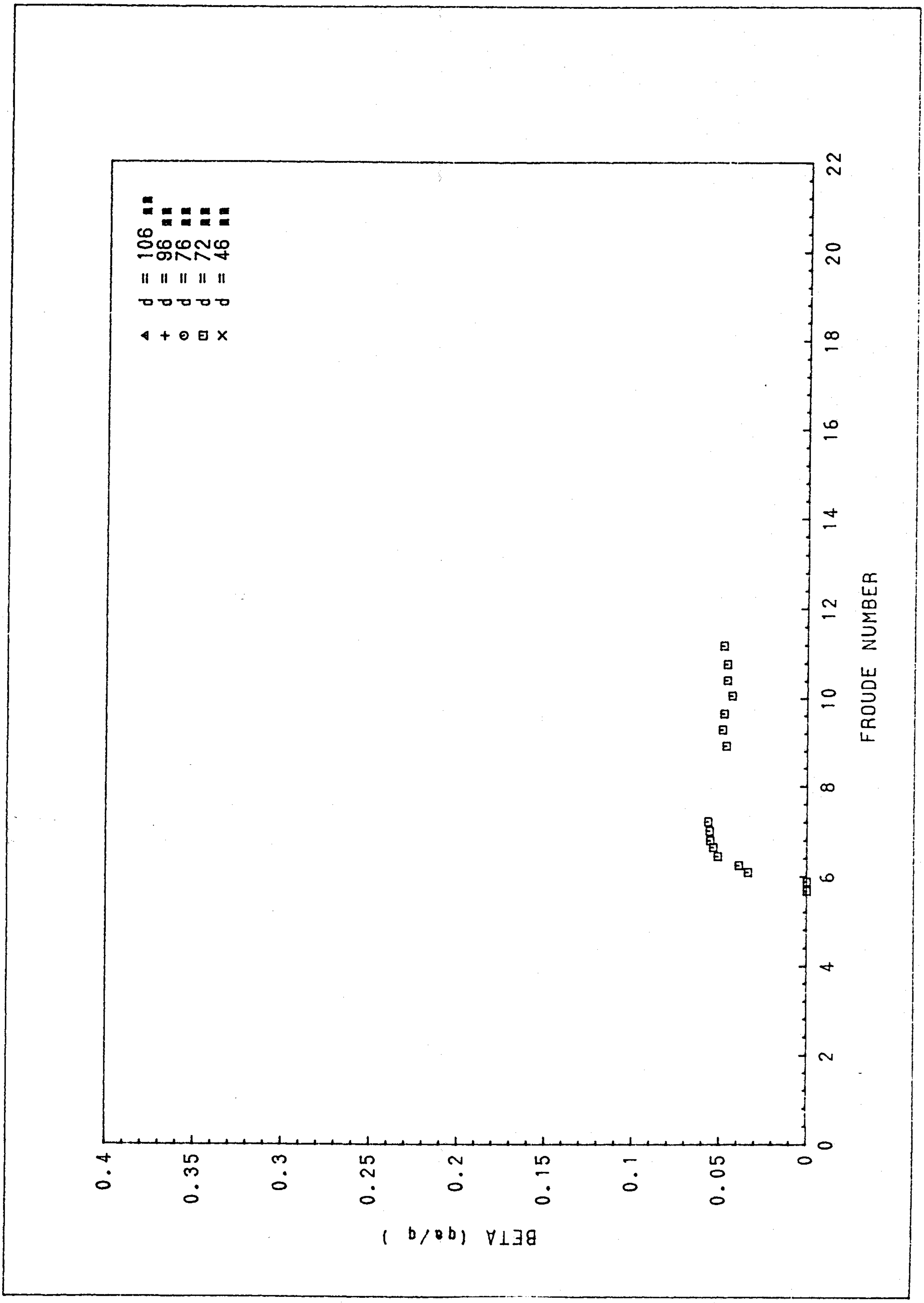


Fig 29 Air demand ratio versus Froude number : Aerator 4, air valve setting 0

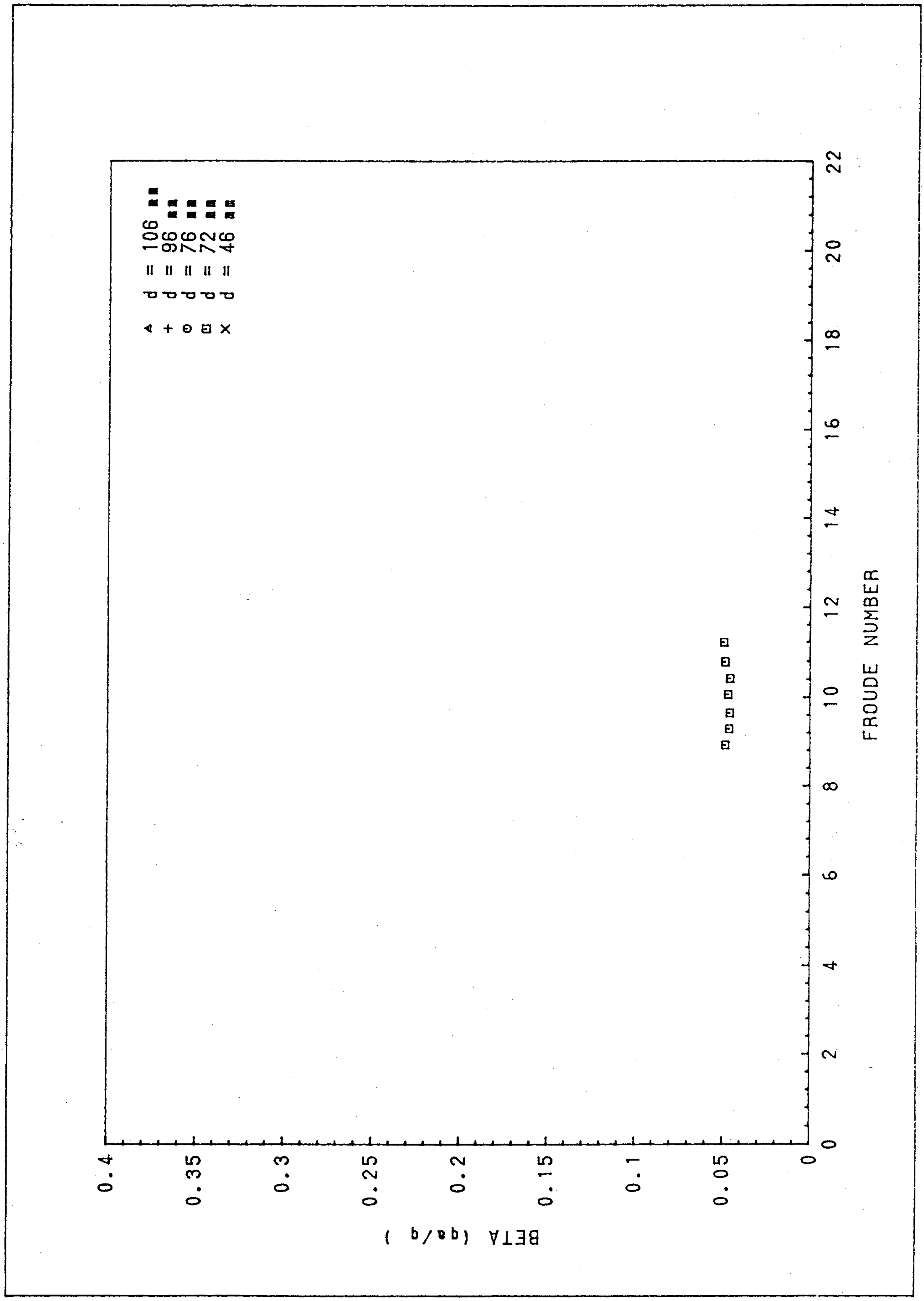


Fig 30 Air demand ratio versus Froude number : Aerator 6, air valve setting 0

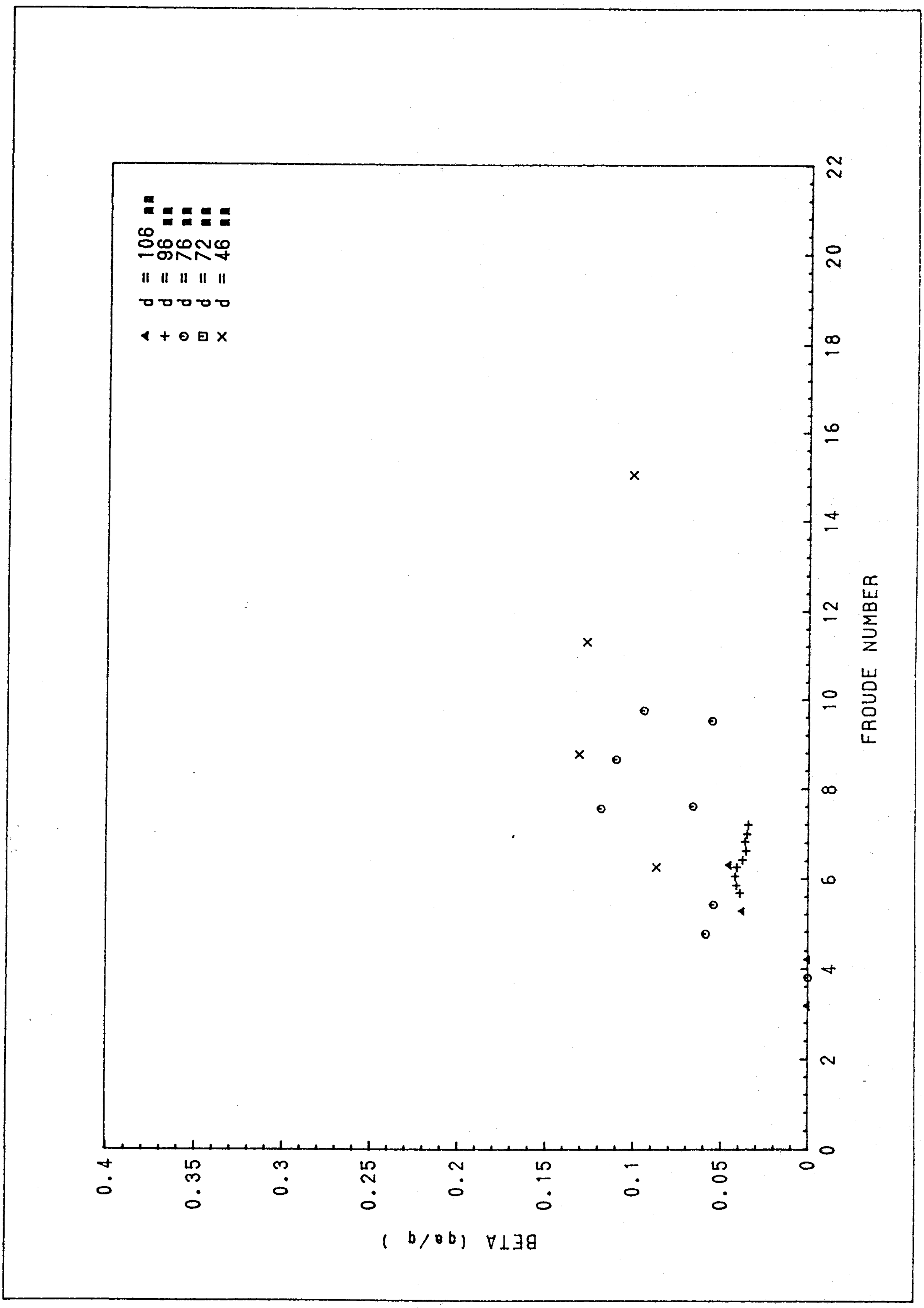


Fig 31 Air demand ratio versus Froude number : Aerator 7, air valve setting 0

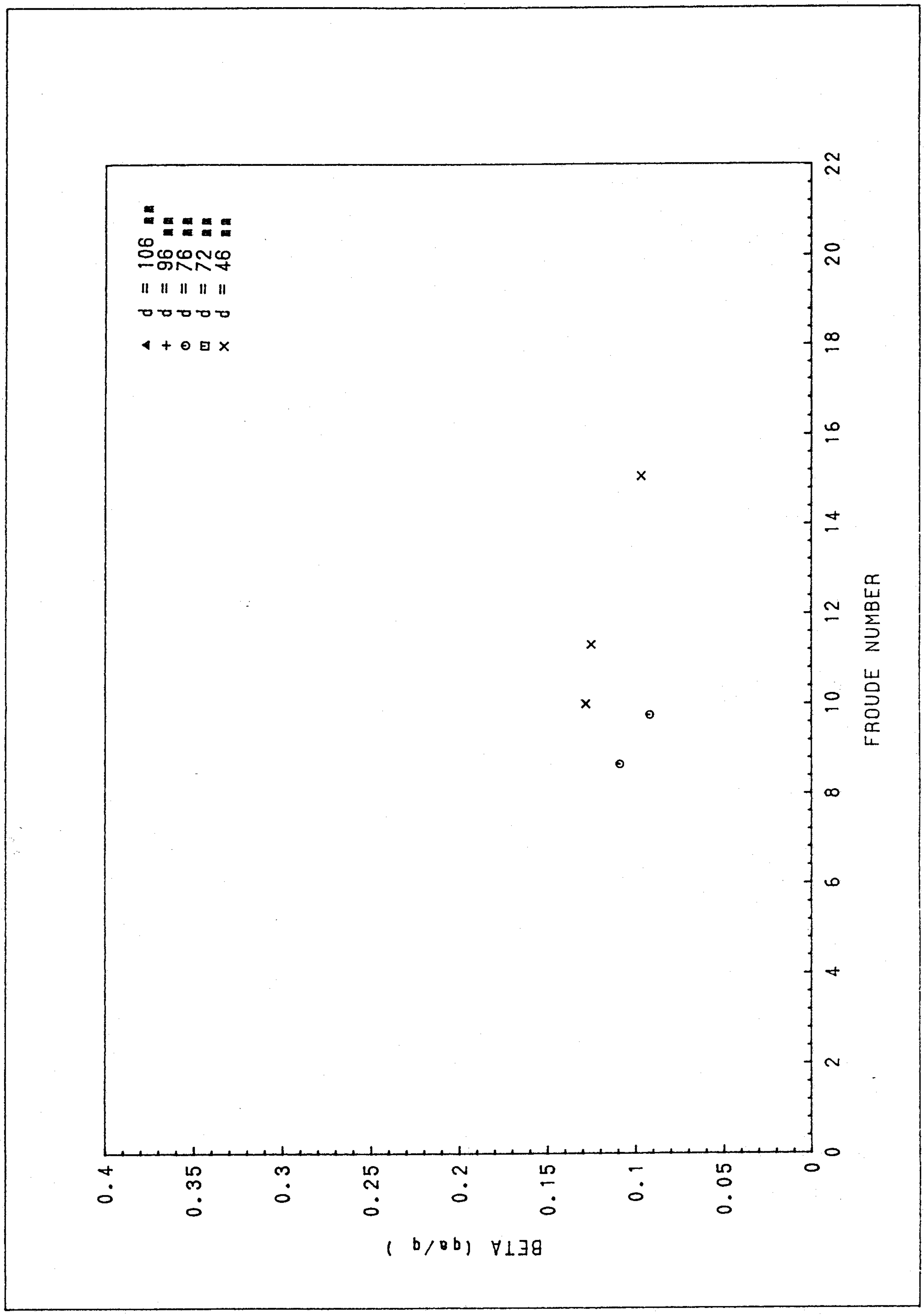


Fig 32 Air demand ratio versus Froude number : Aerator 7, air valve setting 2

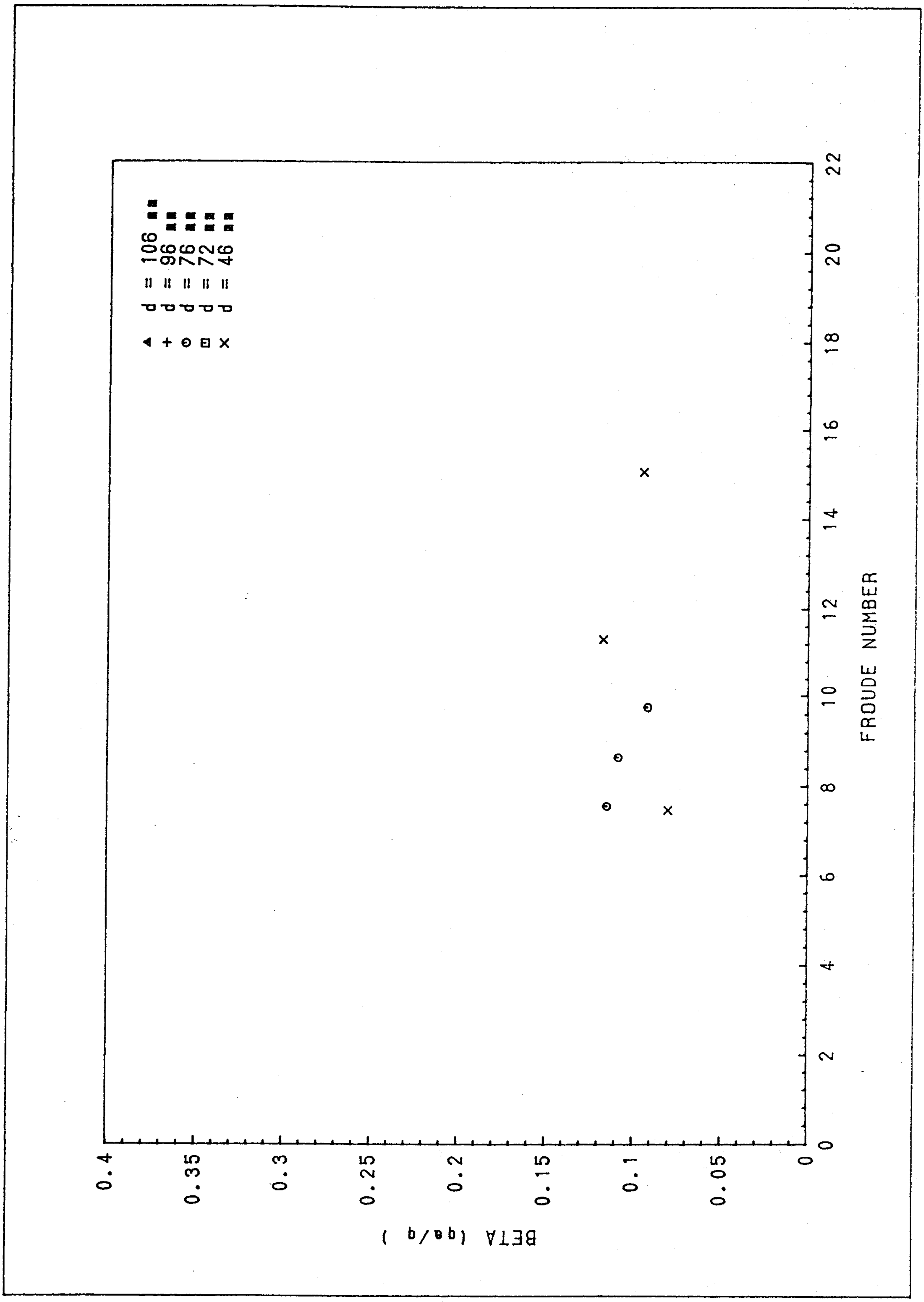


Fig 33 Air demand ratio versus Froude number : Aerator 7, air valve setting 3

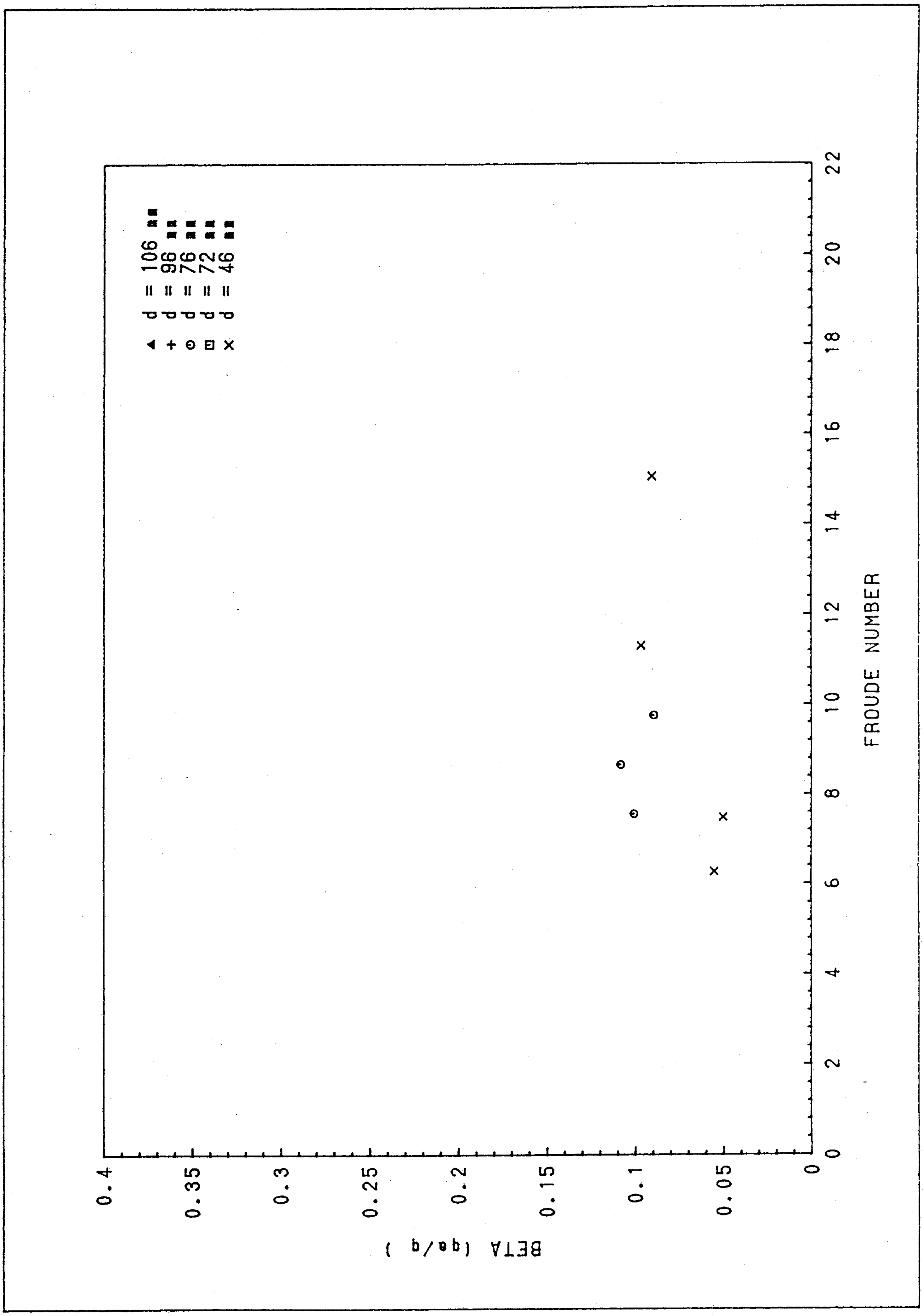


Fig 34 Air demand ratio versus Froude number : Aerator 7, air valve setting 4

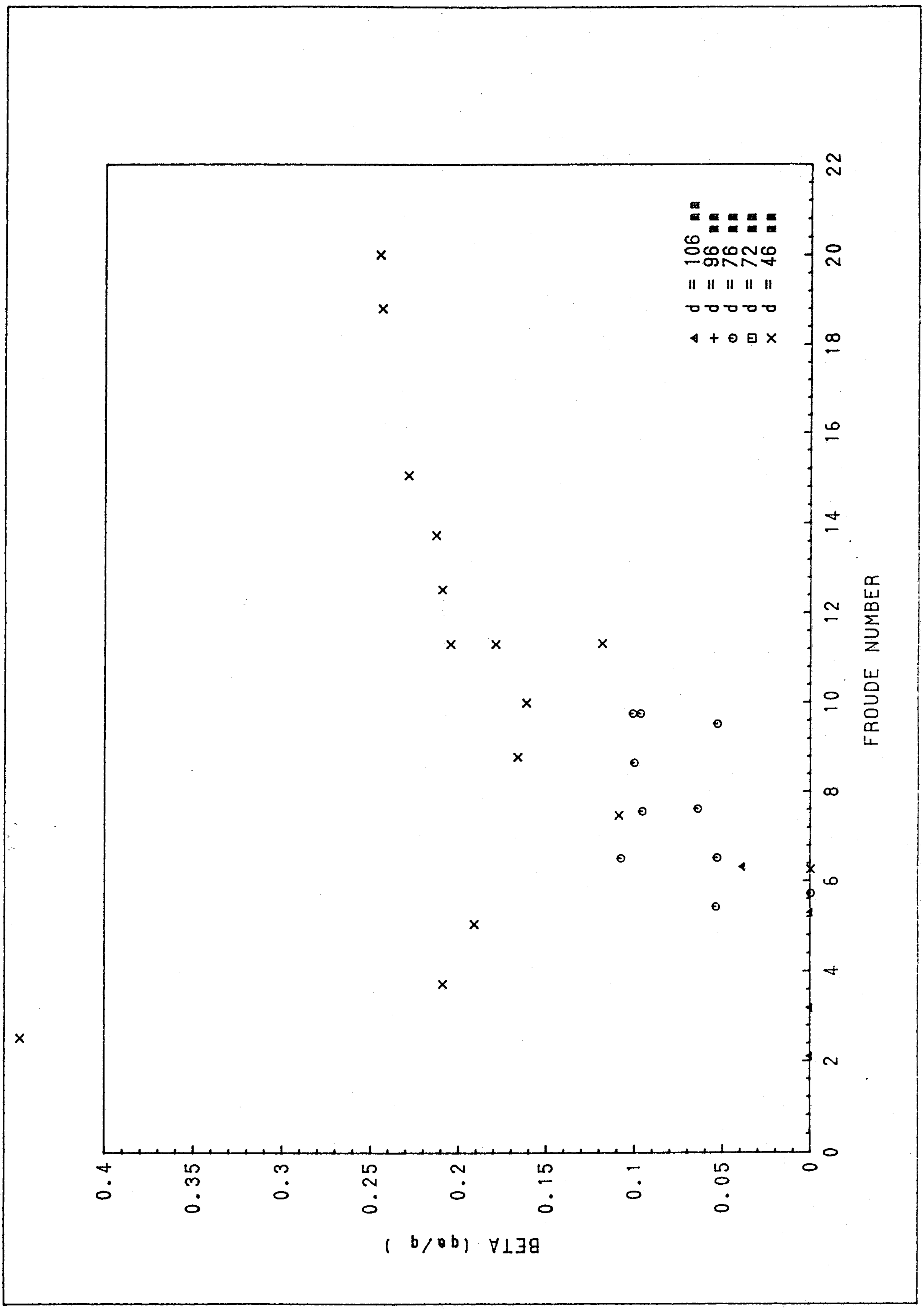


Fig 35 Air demand ratio versus Froude number : Aerator 9, air valve setting 0

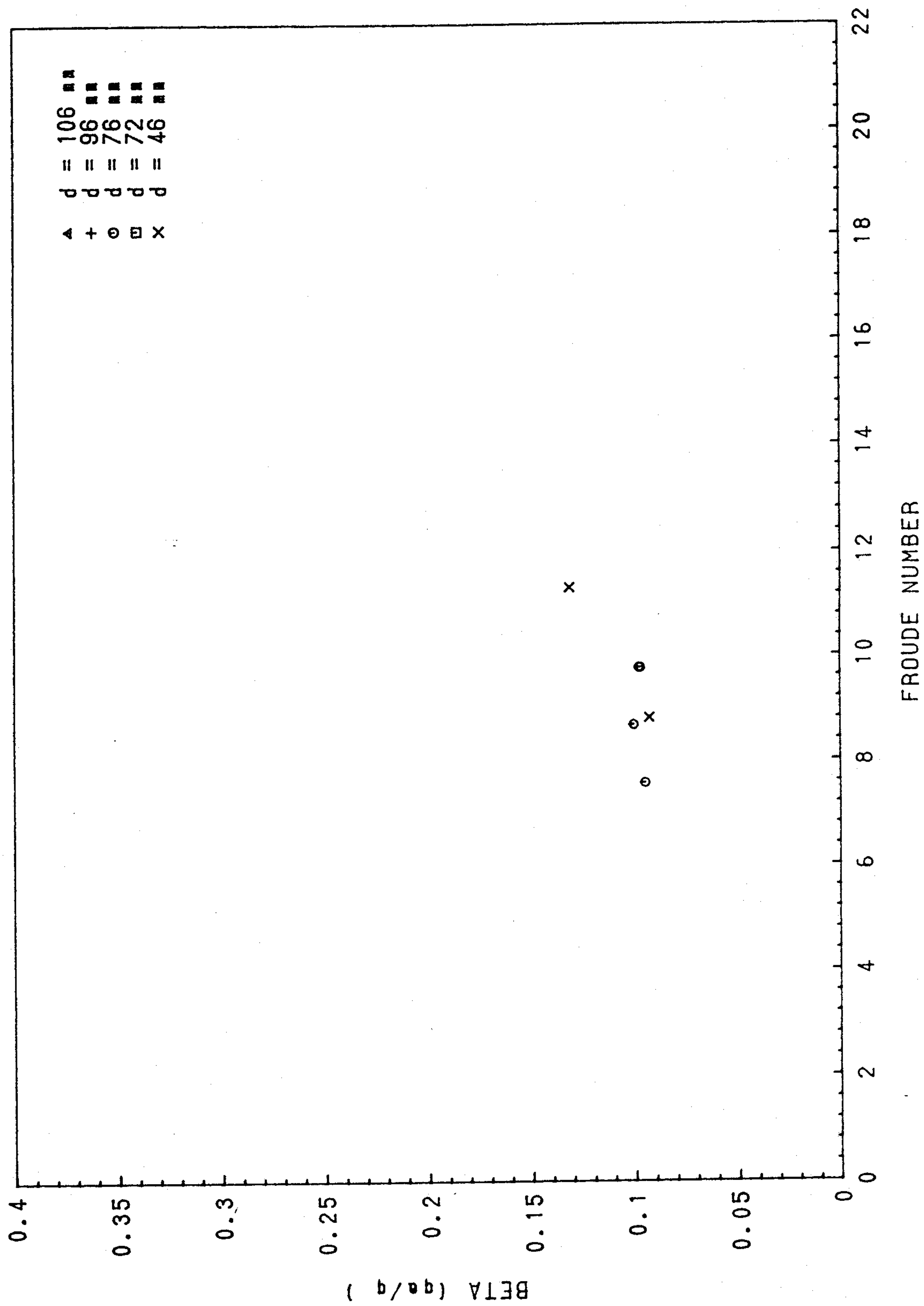


Fig 36   Air demand ratio versus Froude number : Aerator 9, air valve setting 2

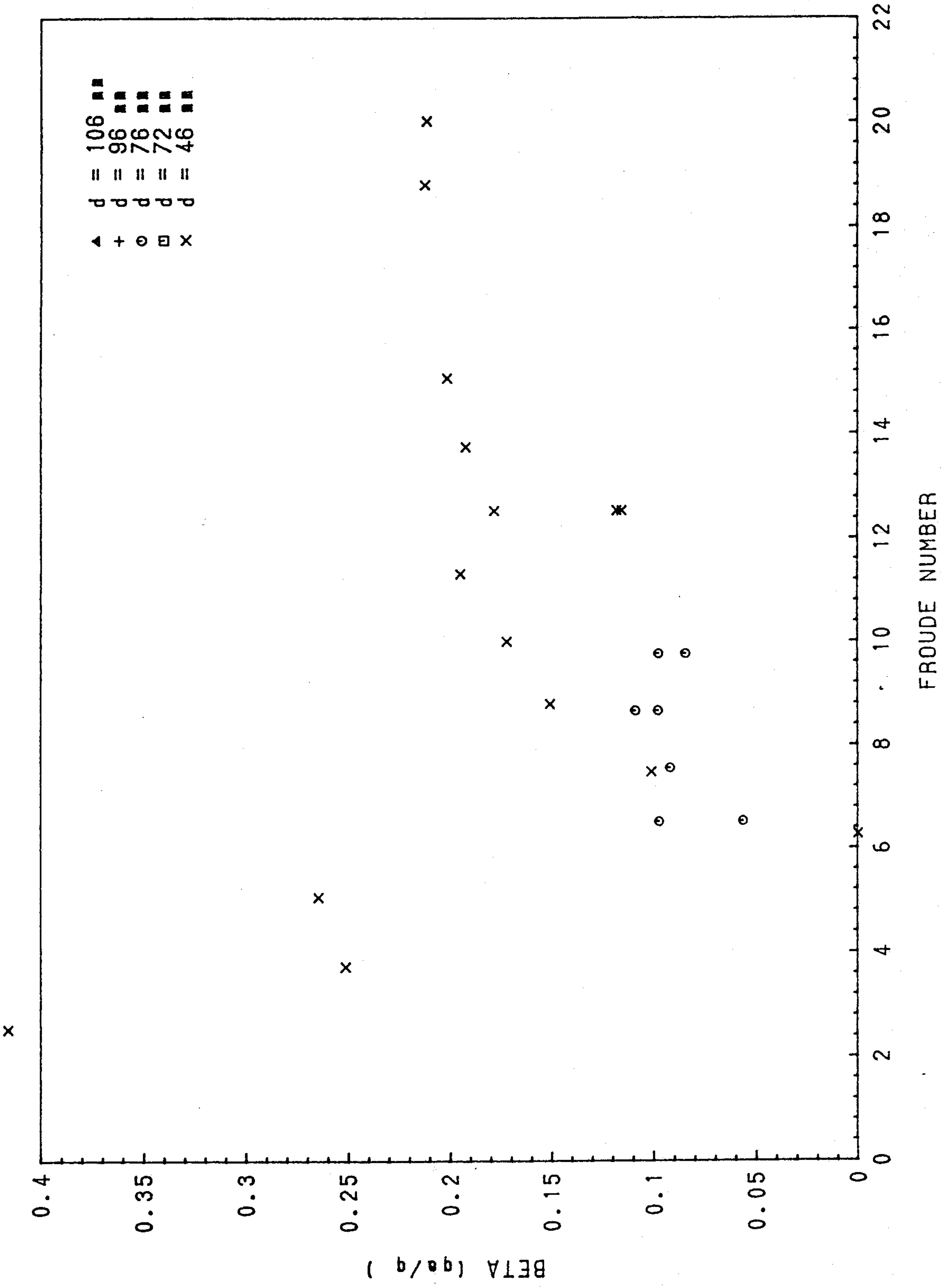


Fig 37   Air demand ratio versus Froude number : Aerator 9, air valve setting 3

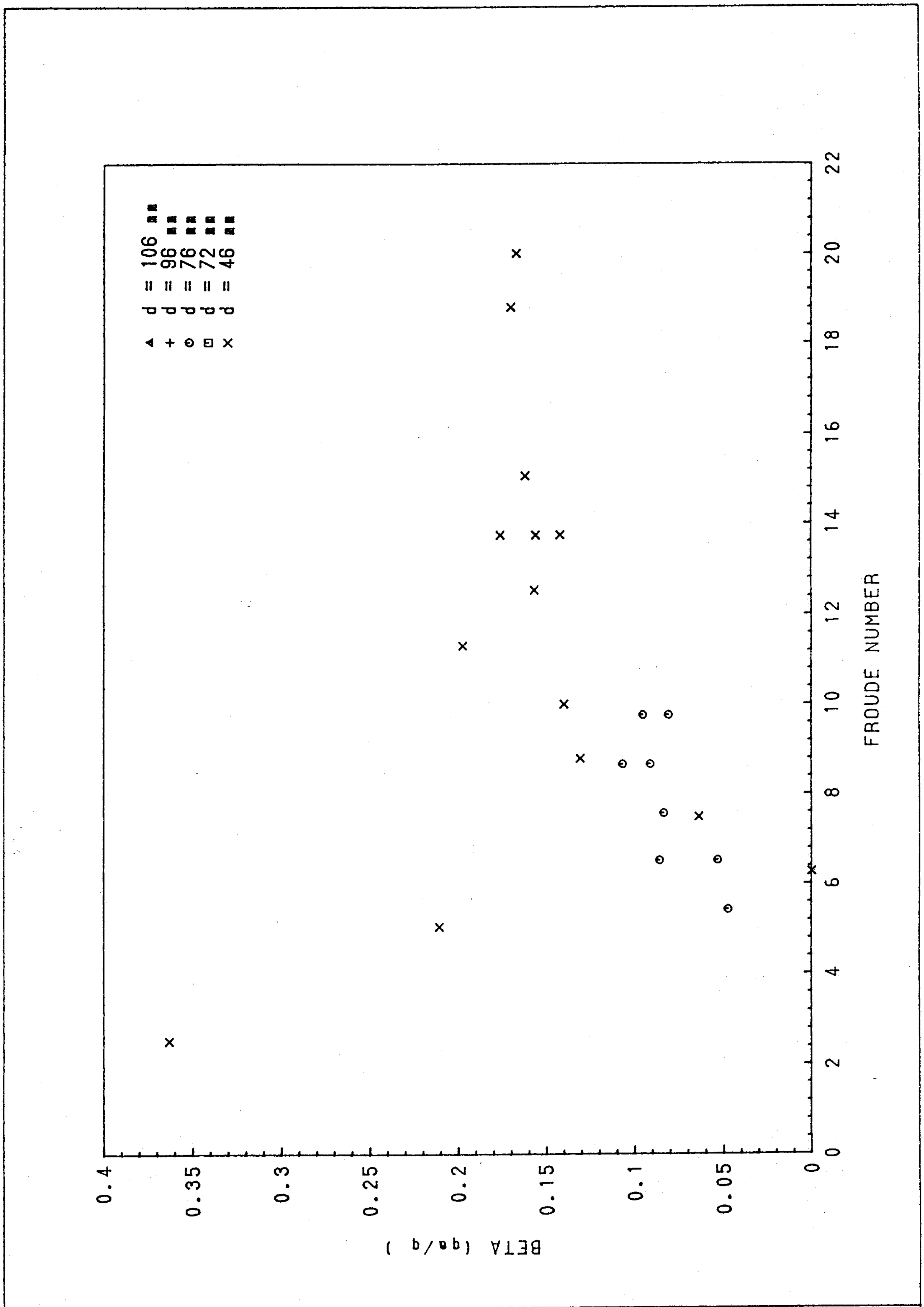


Fig 38 Air demand ratio versus Froude number : Aerator 9, air valve setting 4

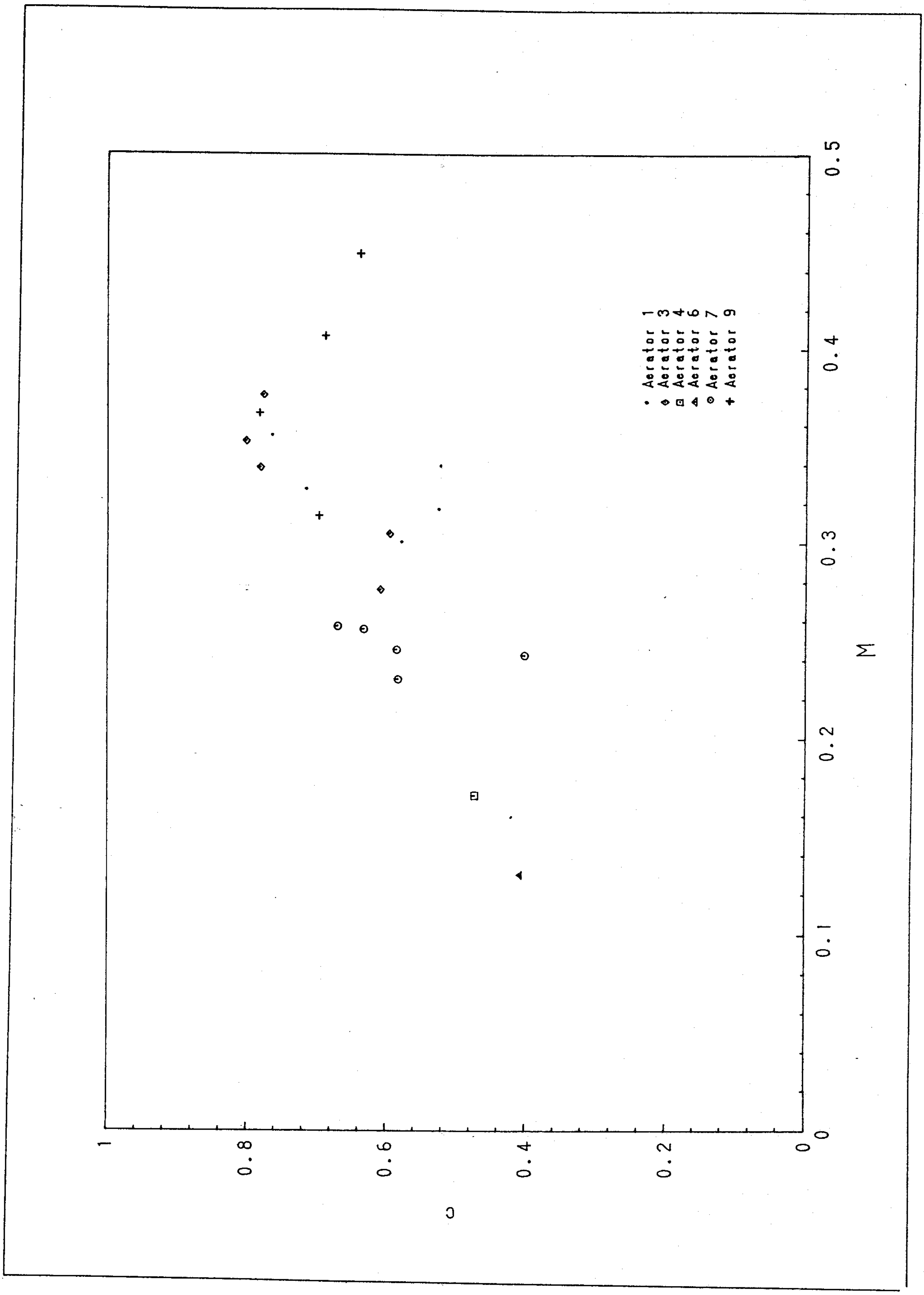


Fig 39 Variation of aeration coefficient c with head-loss parameter M

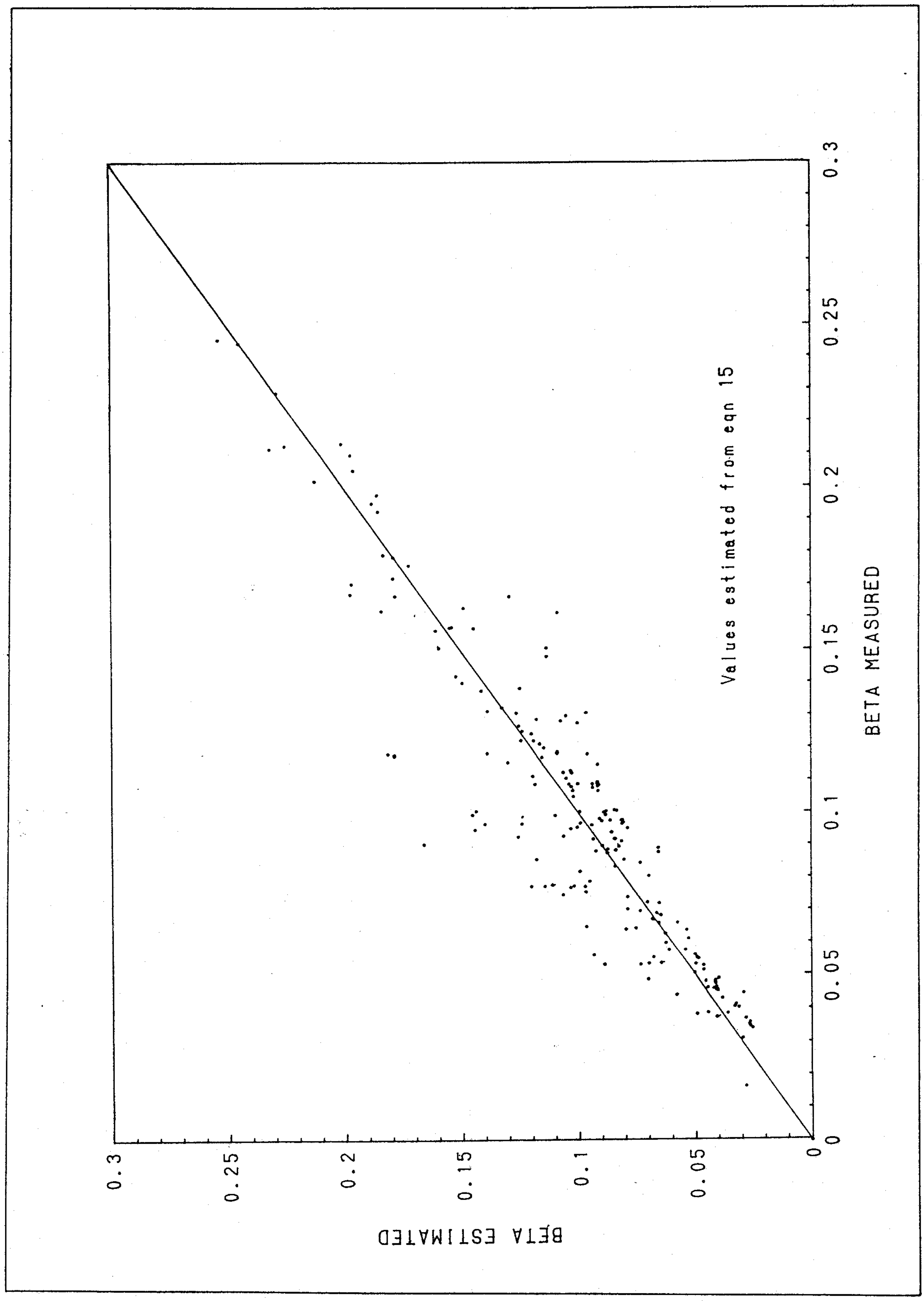


Fig 40 Comparison of measured and estimated values of air demand for main tests

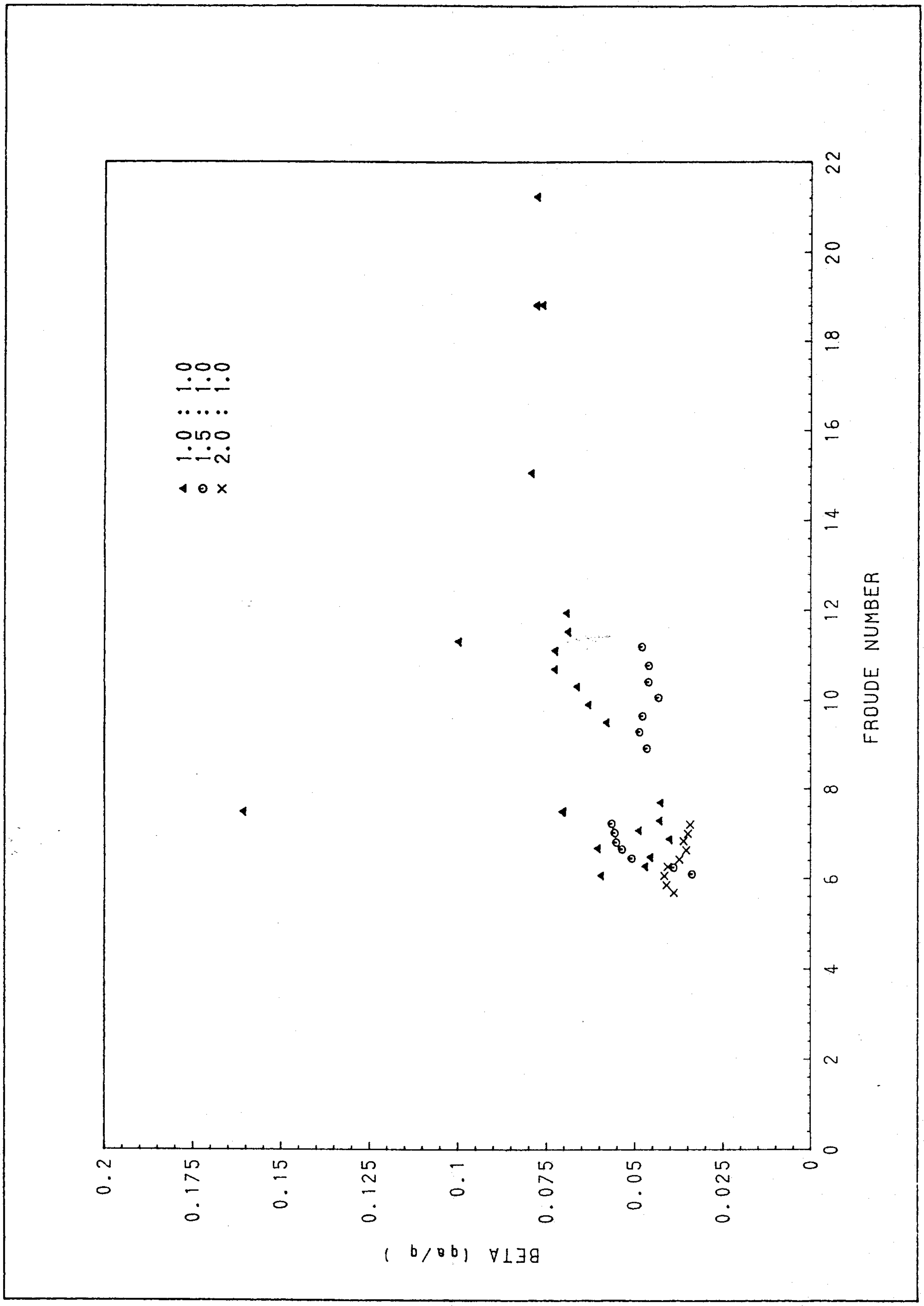


Fig 41   Effect of scale on air demand for ramps with angle of ϕ = 9.1°

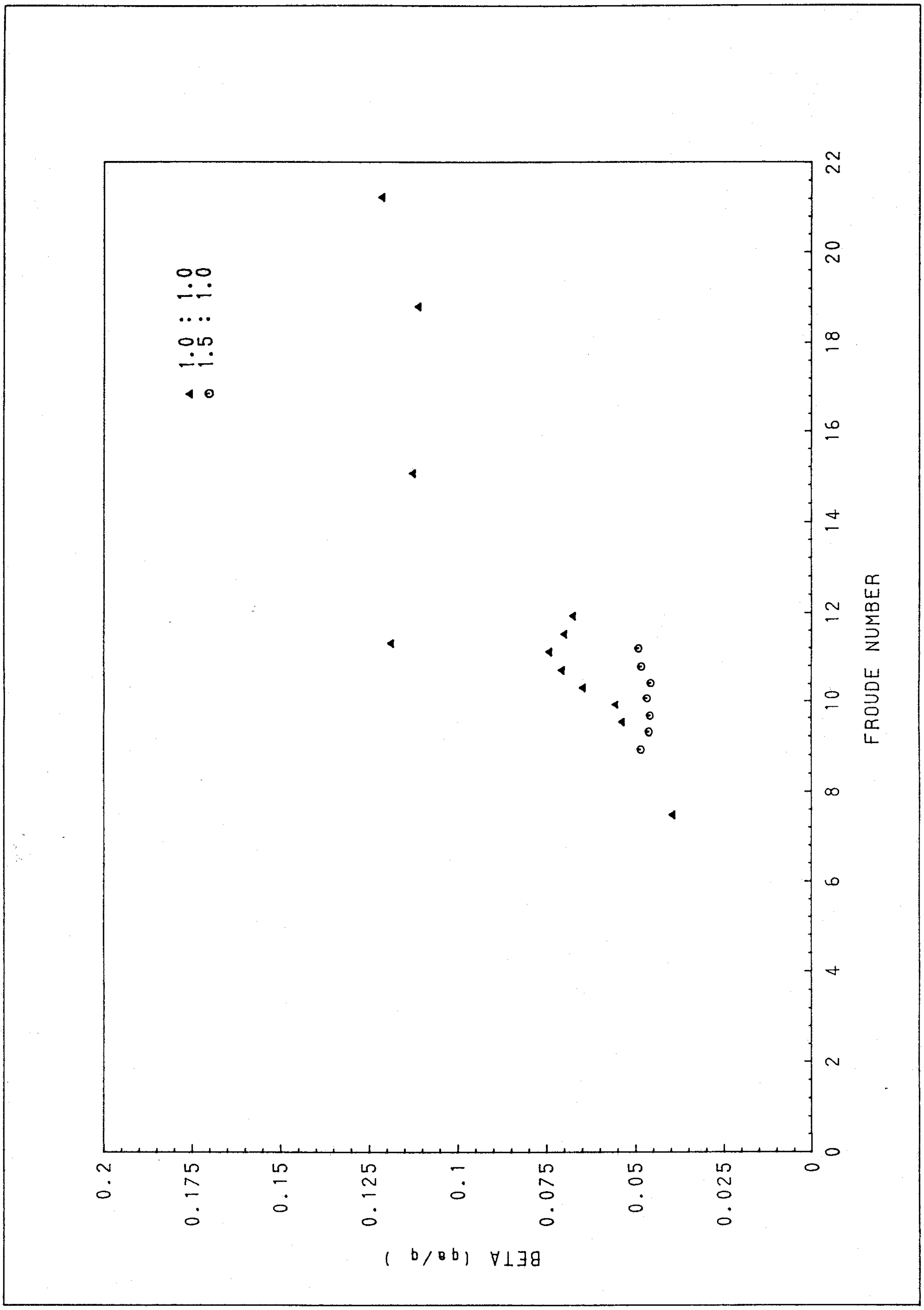


Fig 42 Effect of scale on air demand for ramps with angle of $\phi = 4.6°$

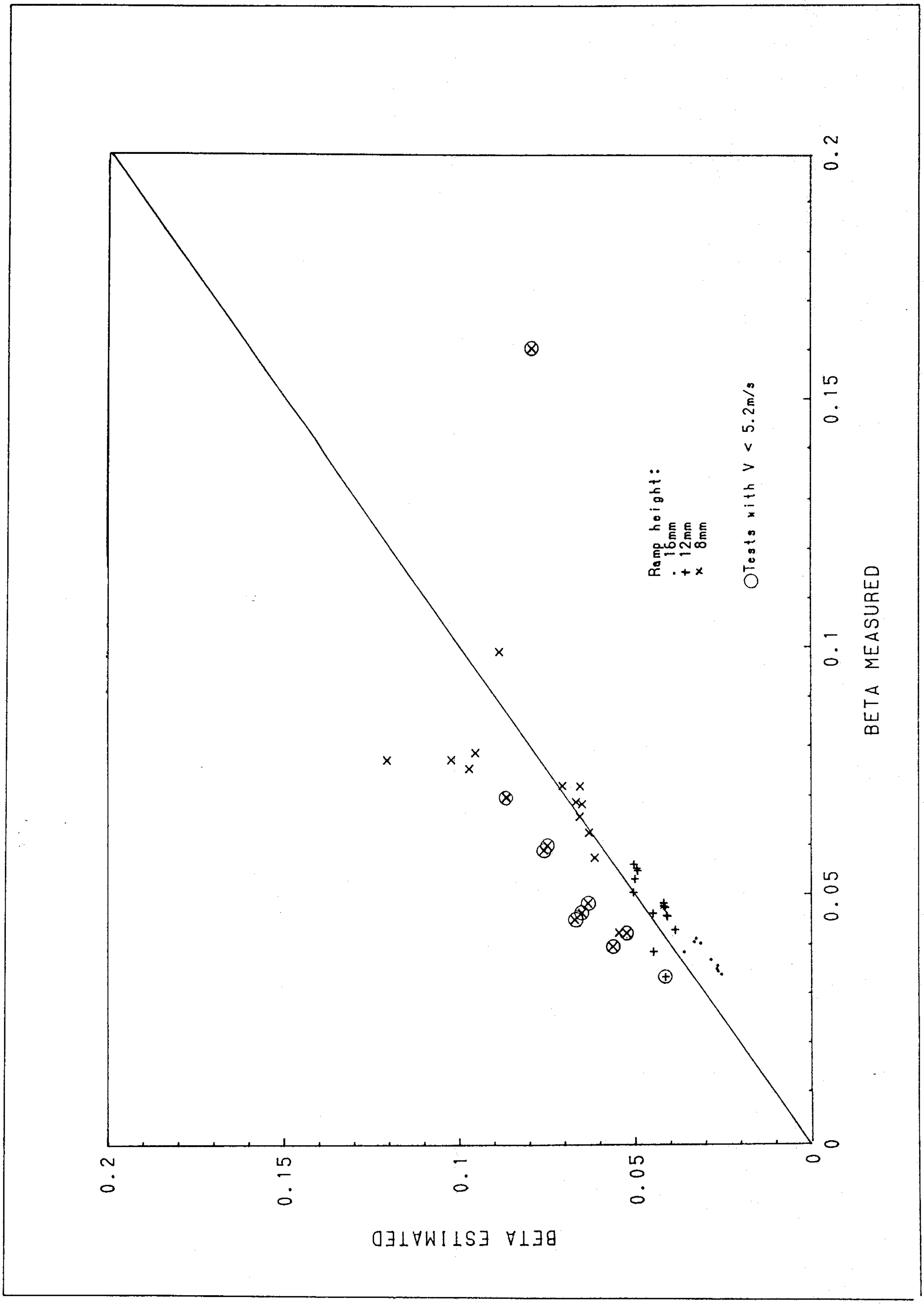


Fig 43 Measured and estimated air demands for scale tests with ramp angle of $\phi = 9.1°$

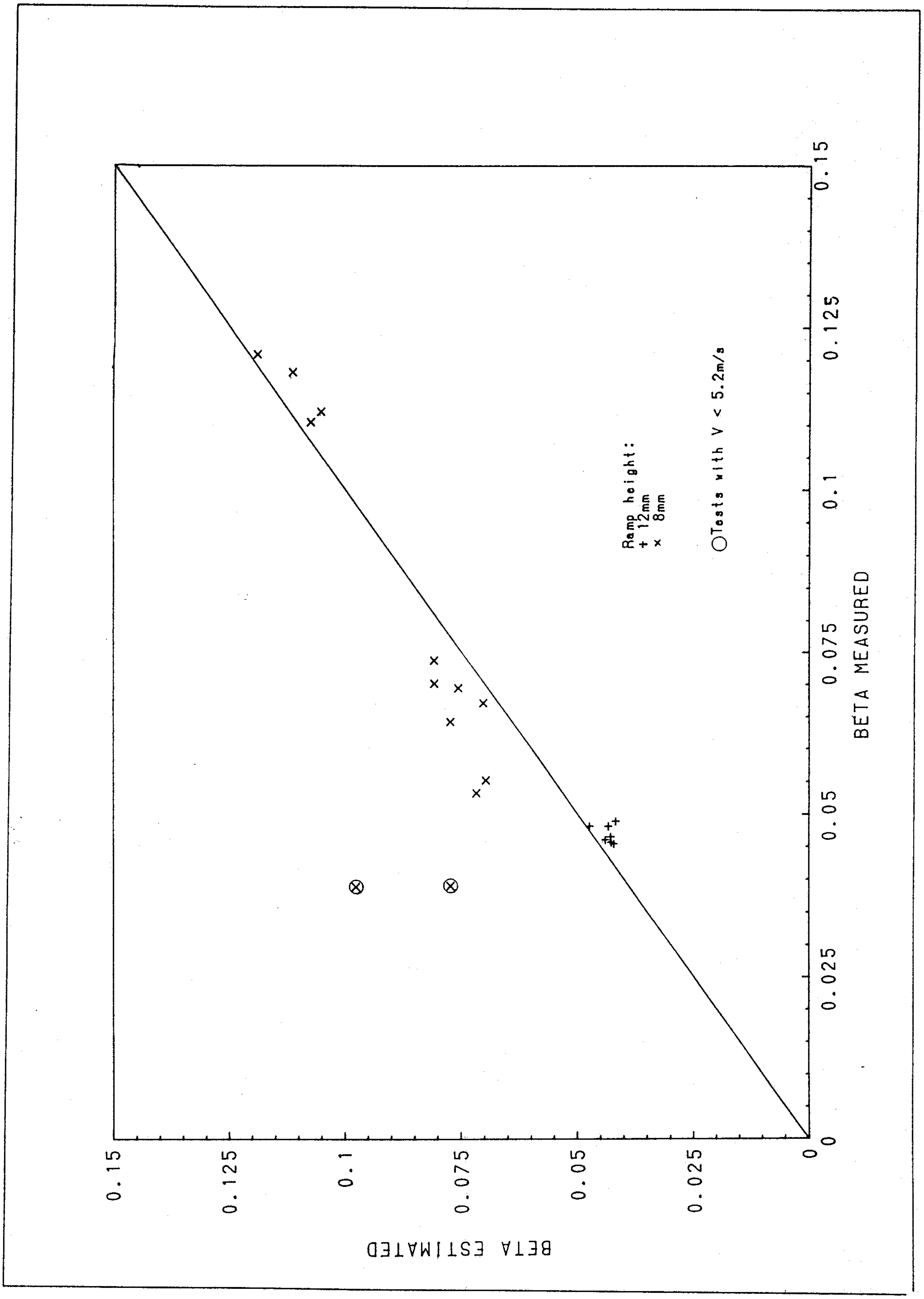


Fig 44 Measured and estimated air demands for scale tests with ramp angle of $\phi = 4.6°$

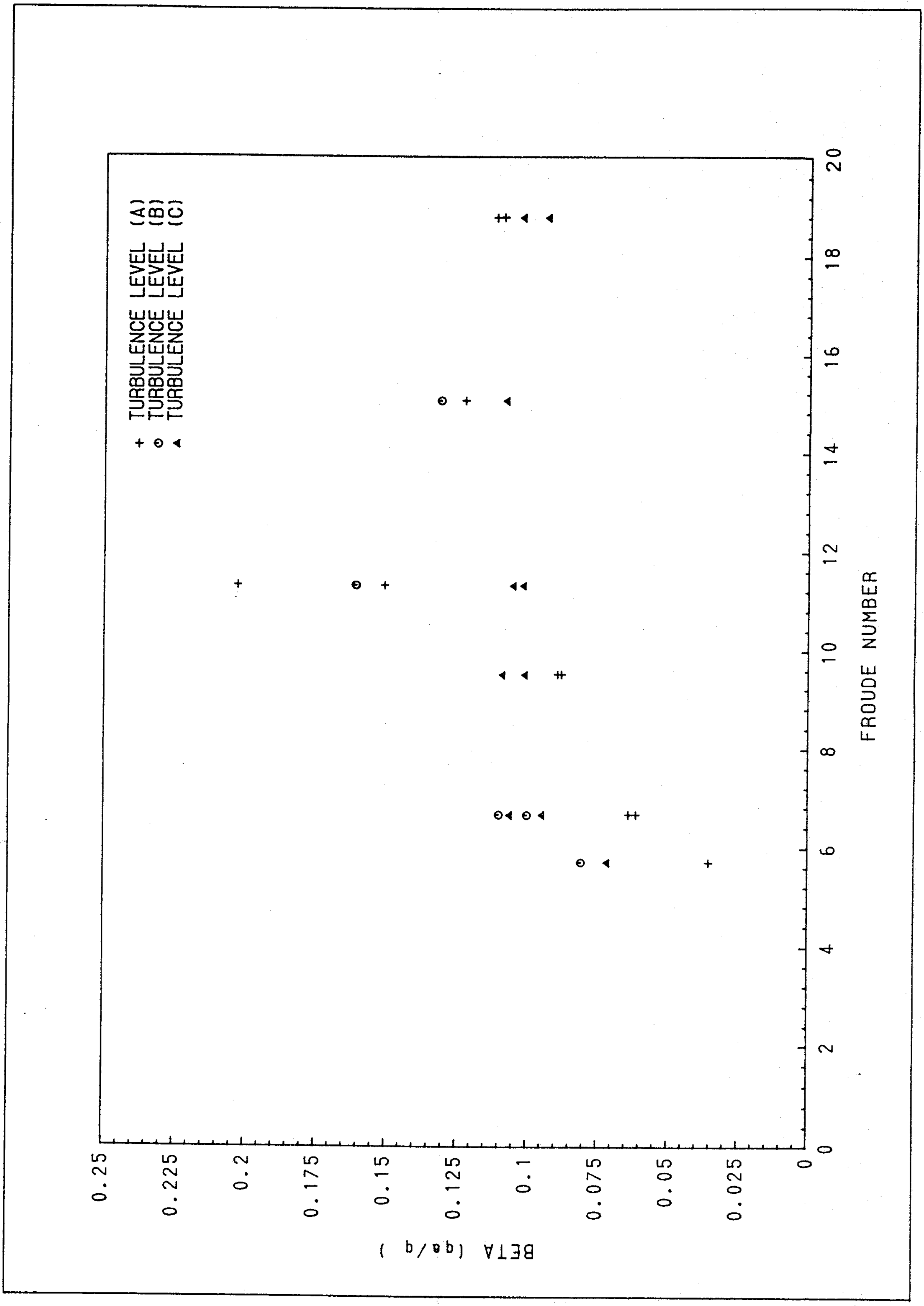


Fig 45 Effect of turbulence intensity on air demand for Aerator 1

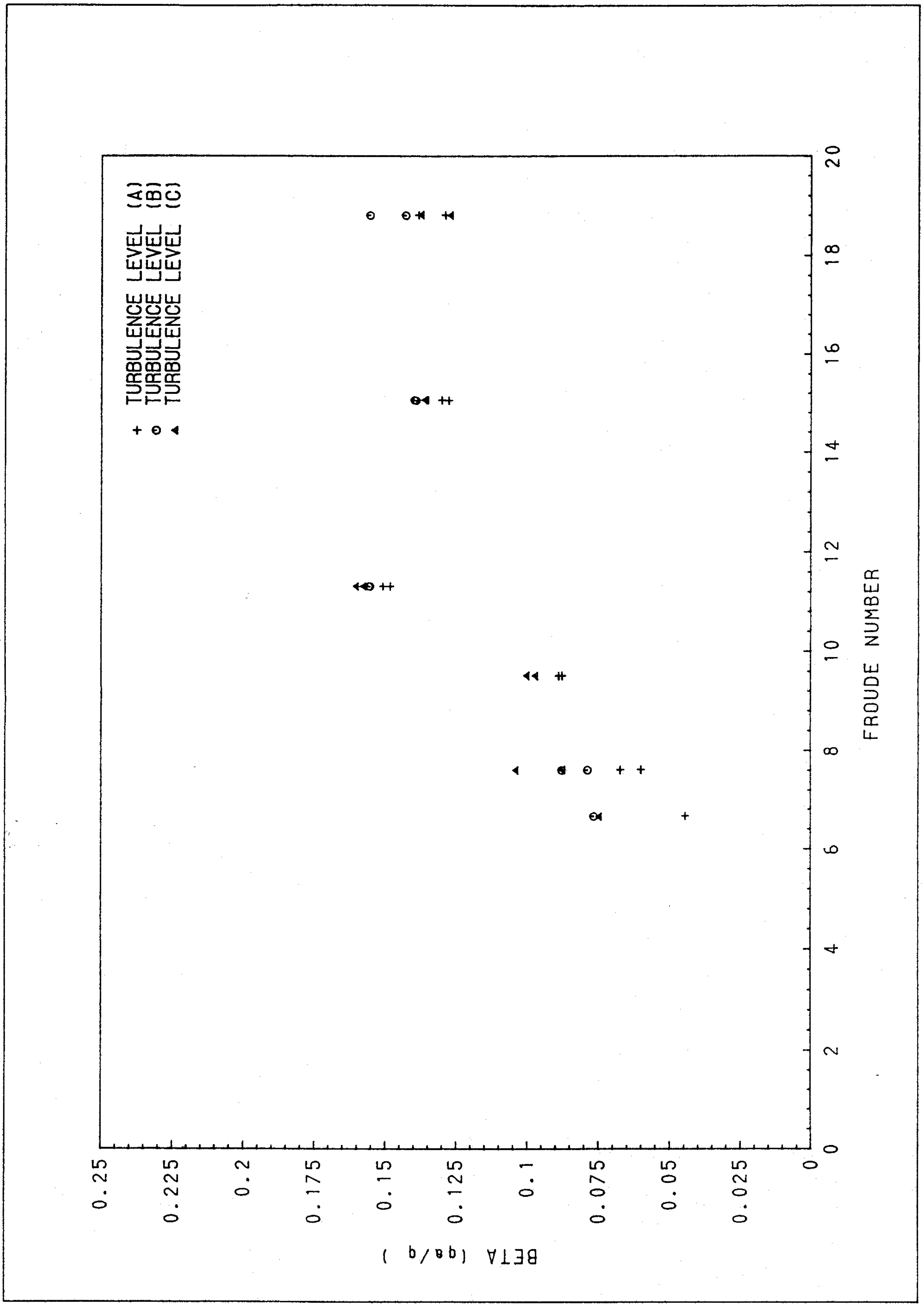


Fig 46 Effect of turbulence intensity on air demand for Aerator 3

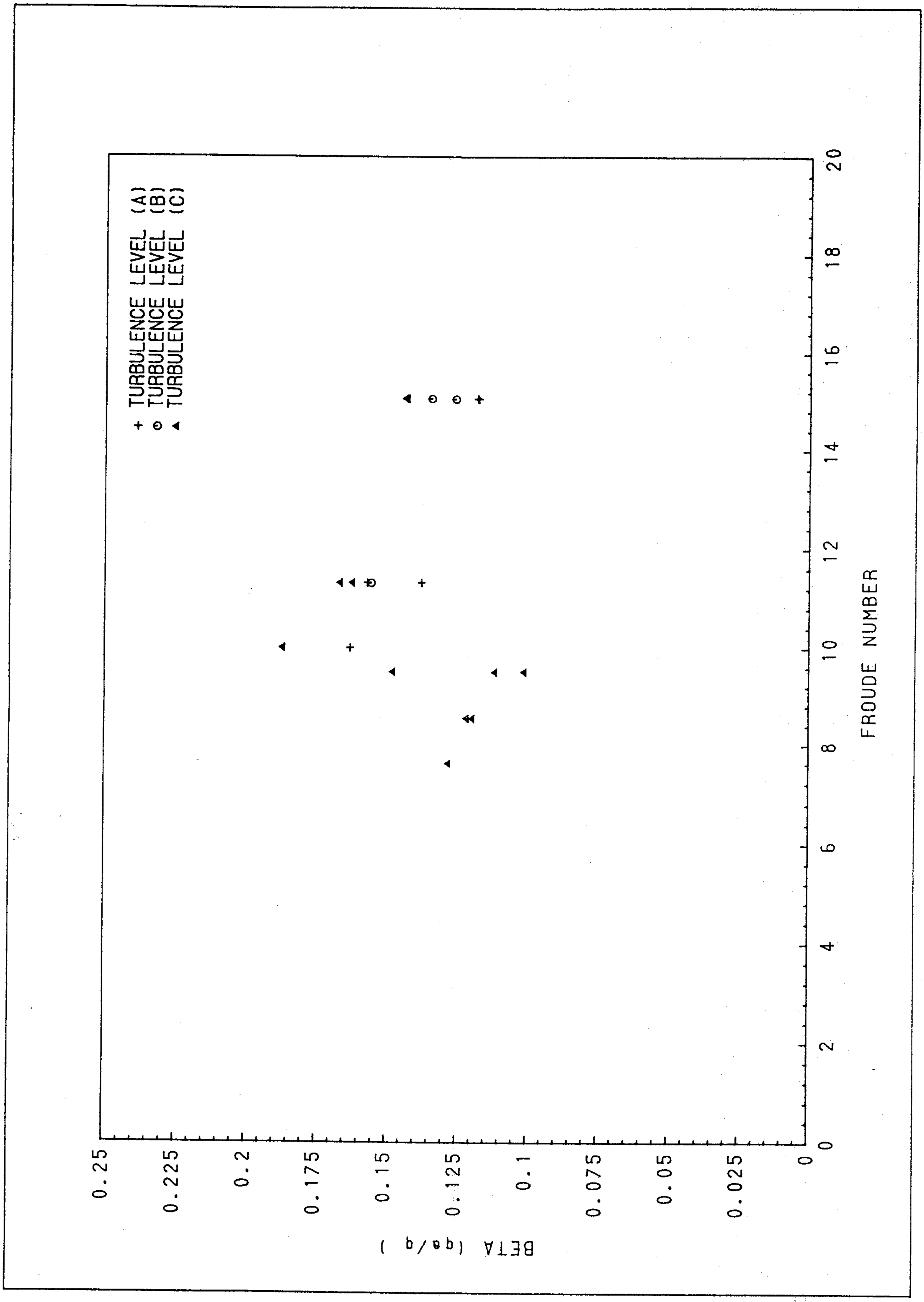


Fig 47 Effect of turbulence intensity on air demand for Aerator 7

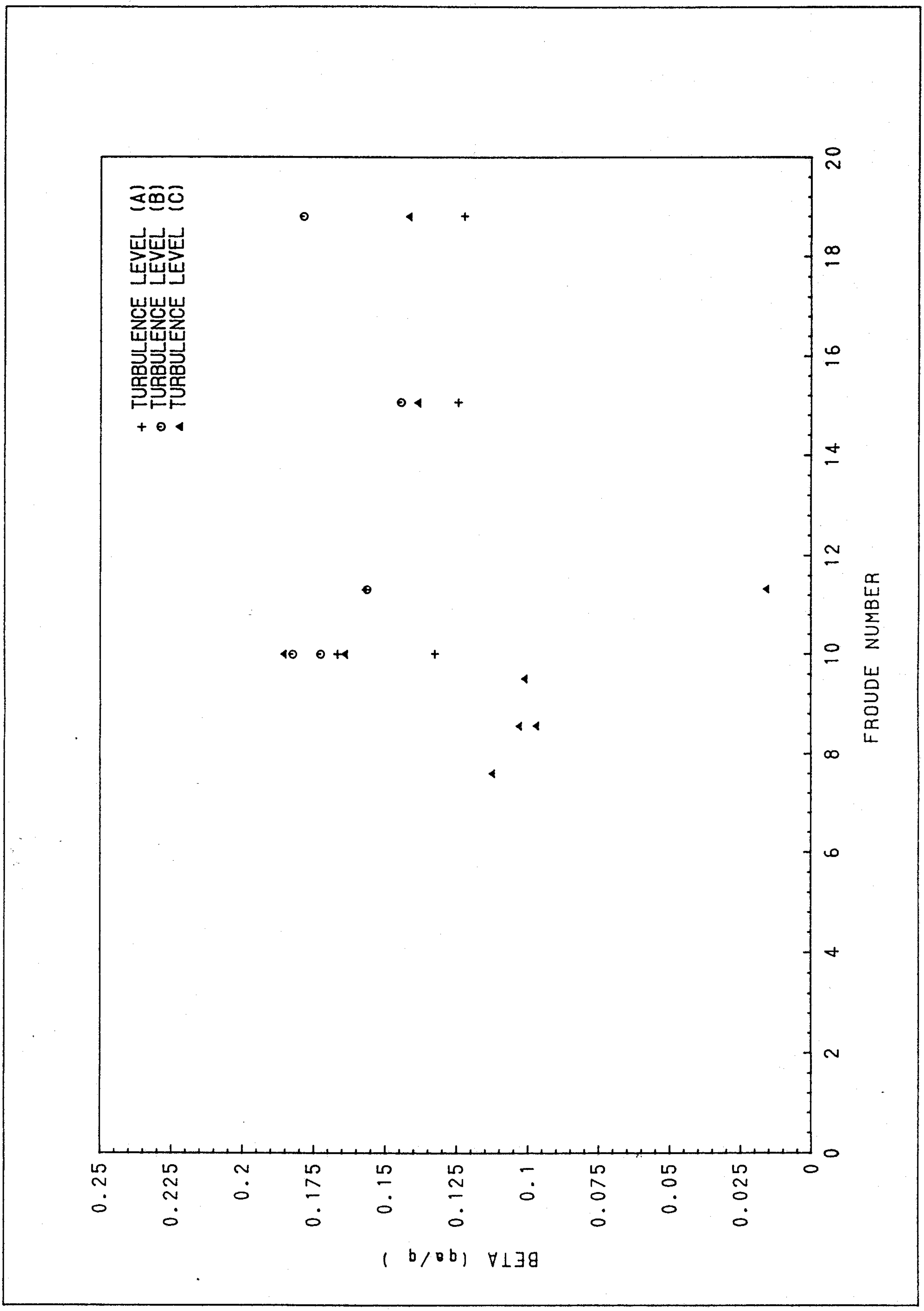


Fig 48   Effect of turbulence intensity on air demand for Aerator 9

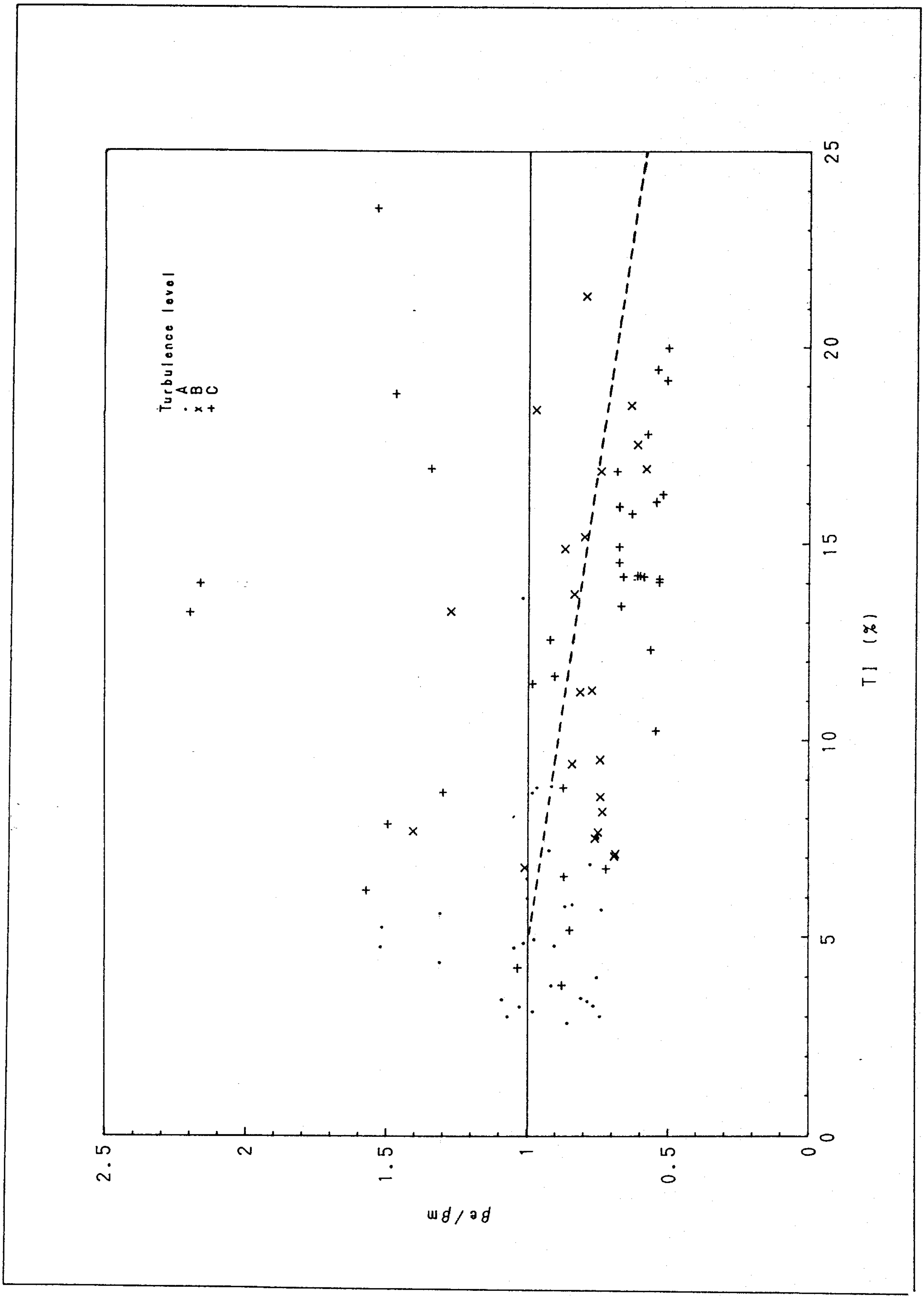


Fig 49 Comparison of measured and estimated values of air demand for turbulence tests

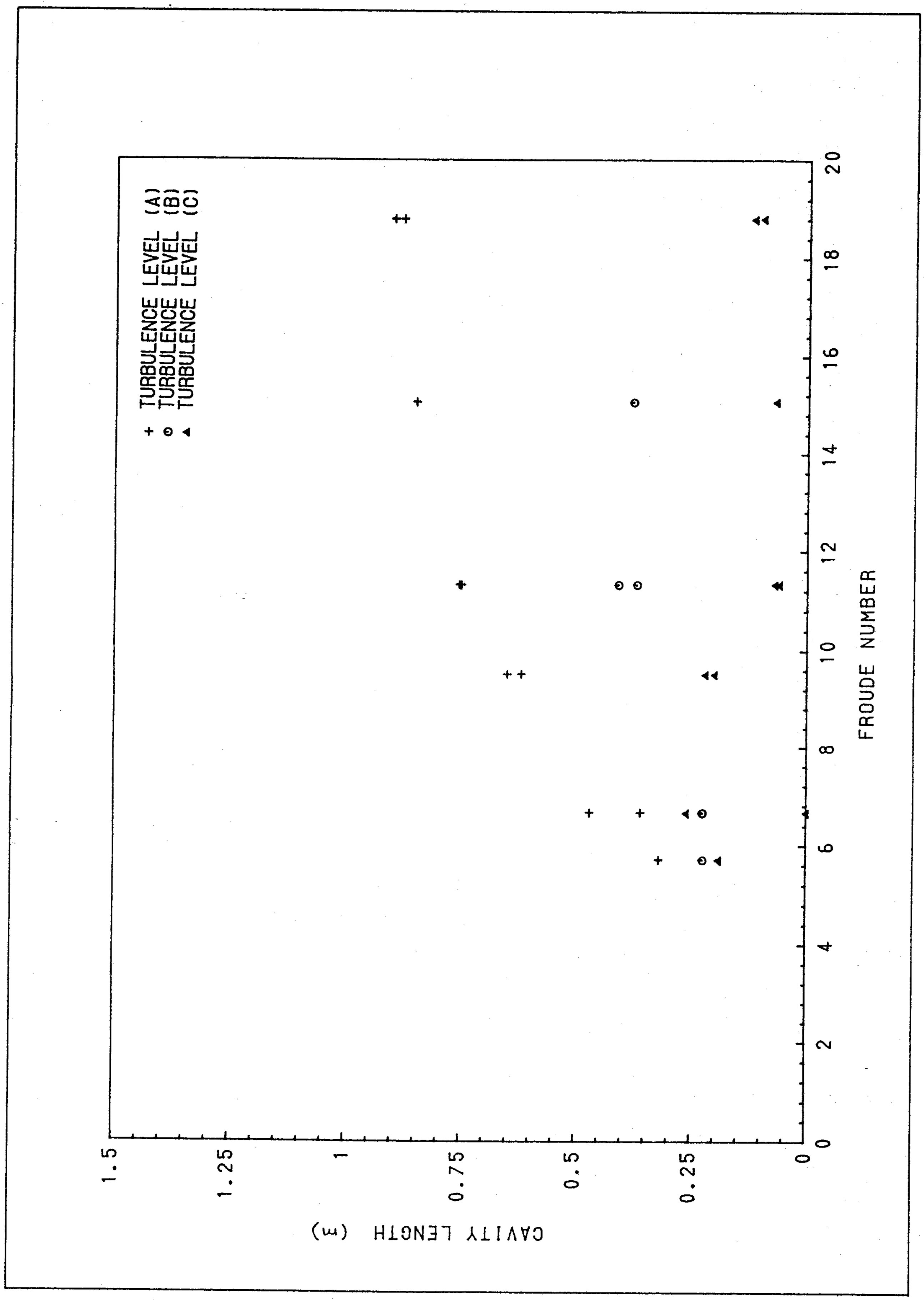


Fig 50 Effect of turbulence intensity on cavity length for Aerator 1

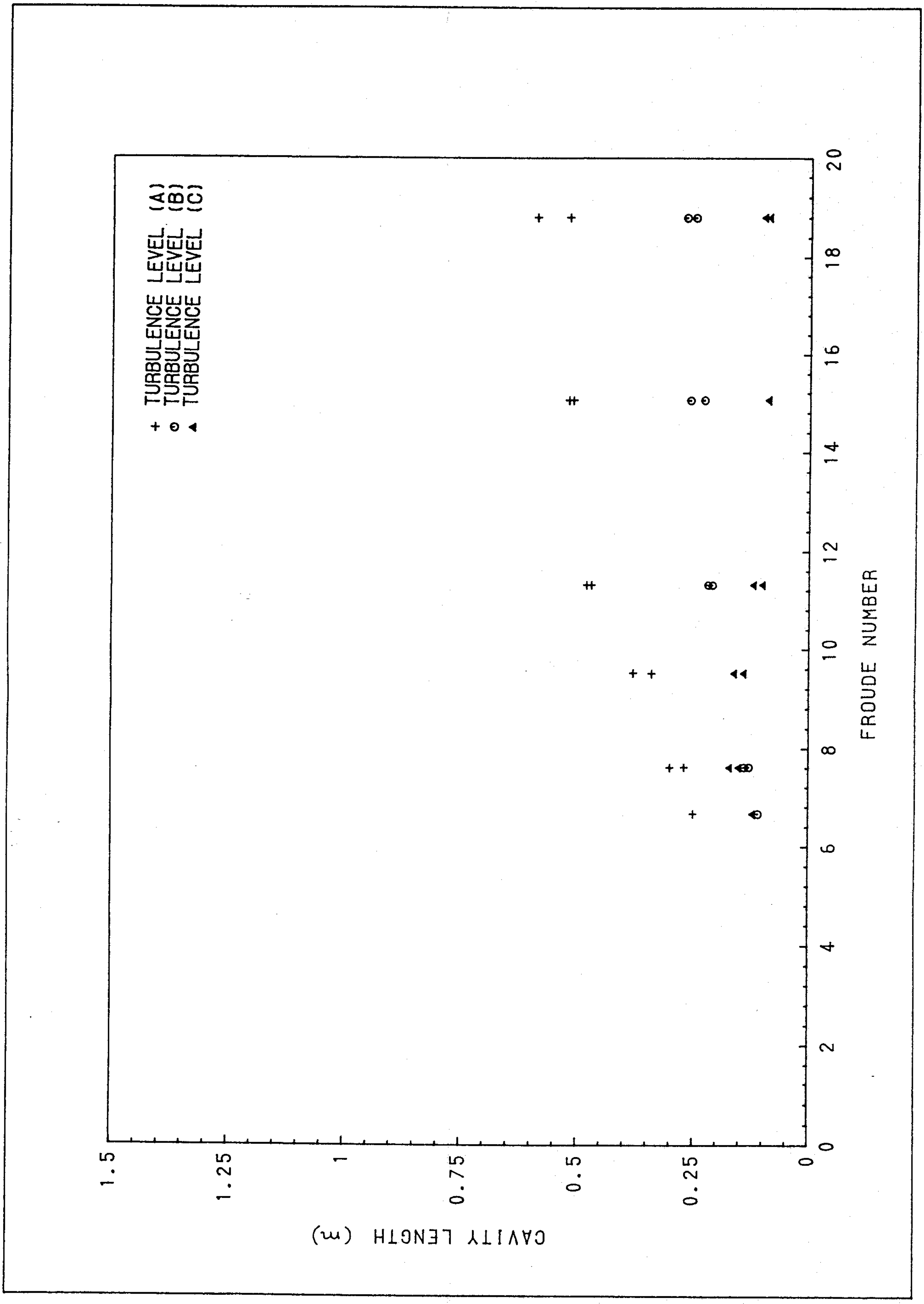


Fig 51 Effect of turbulence intensity on cavity length for Aerator 3

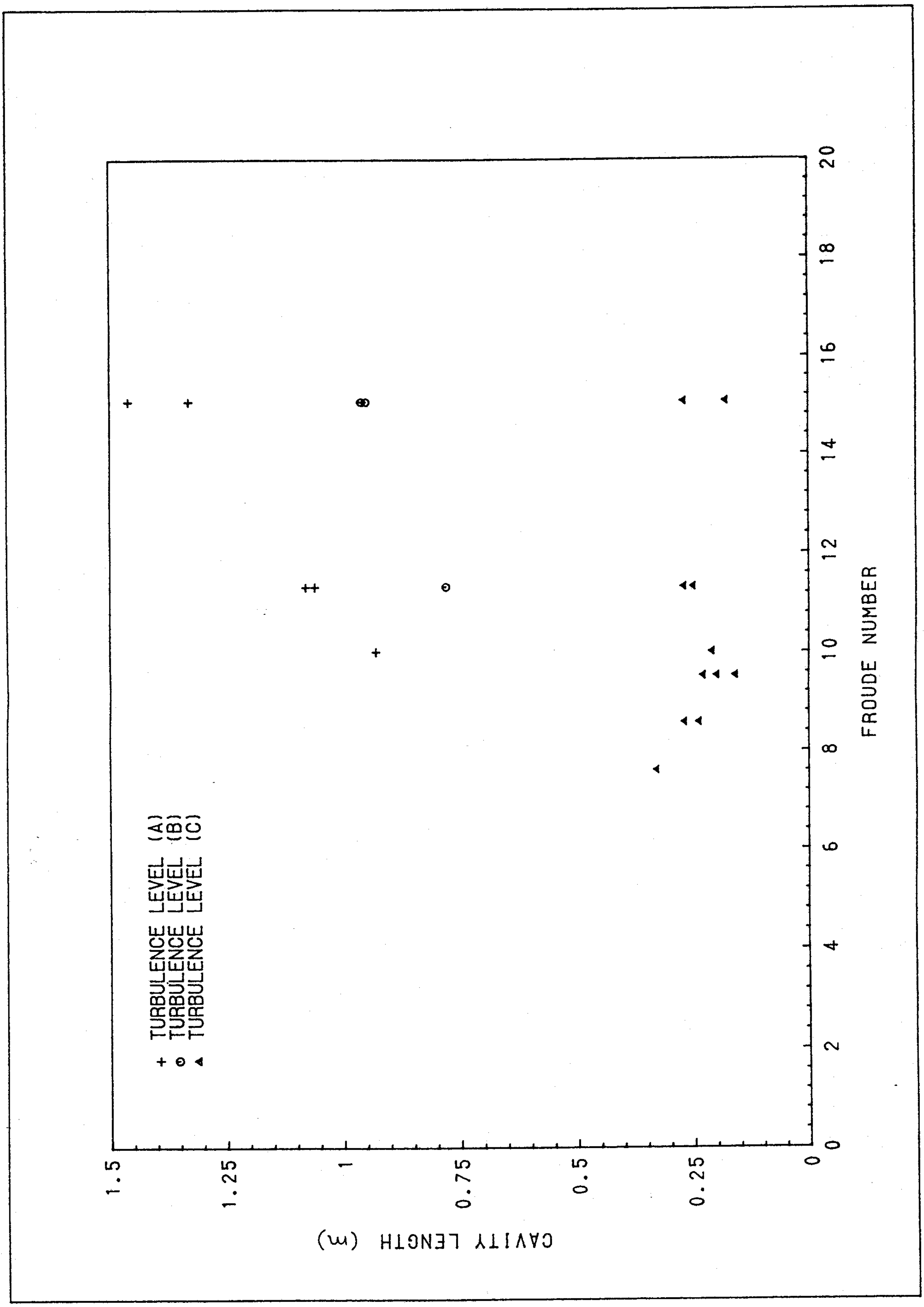


Fig 52   Effect of turbulence intensity on cavity length for Aerator 7

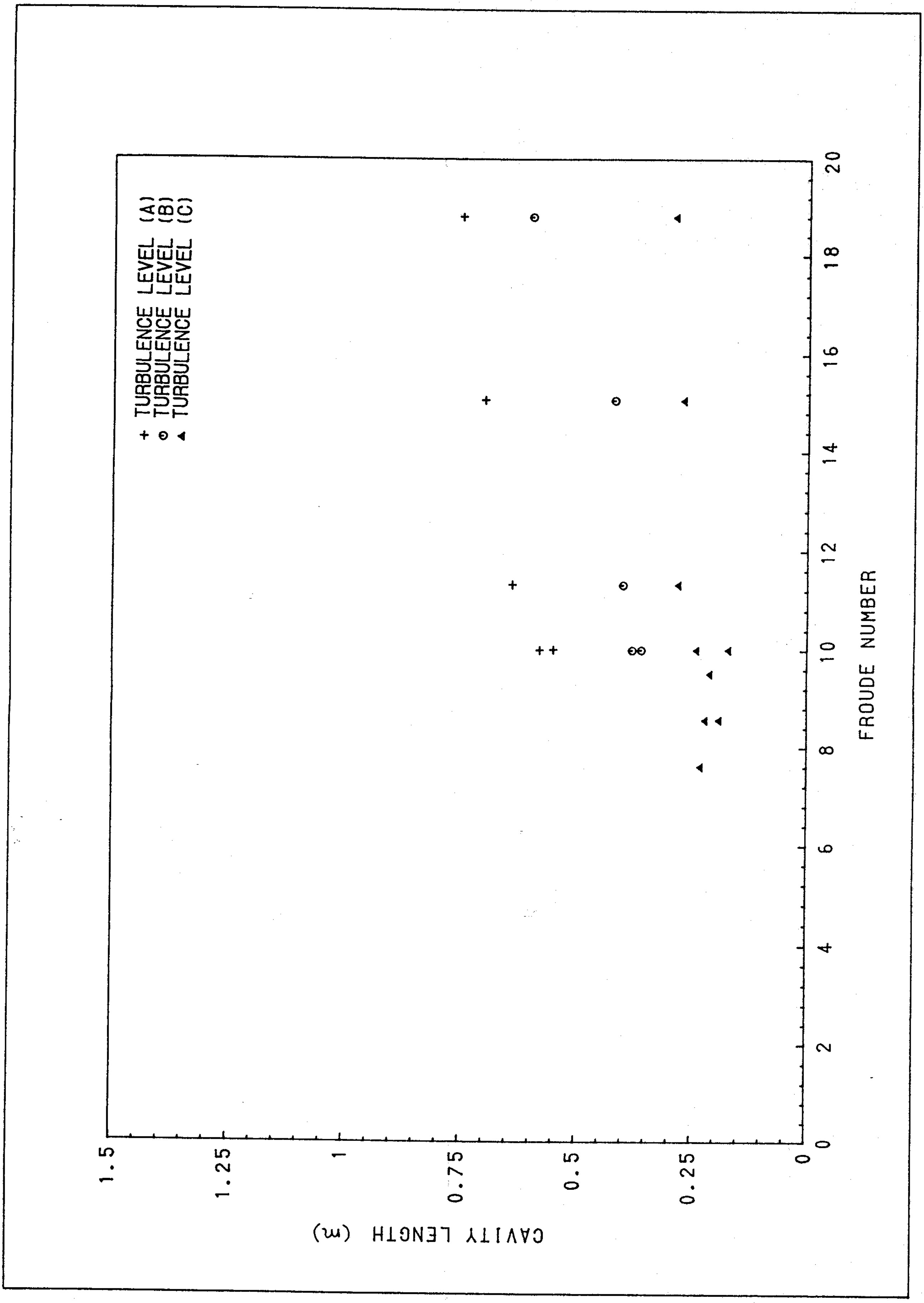


Fig 53 Effect of turbulence intensity on cavity length for Aerator 9

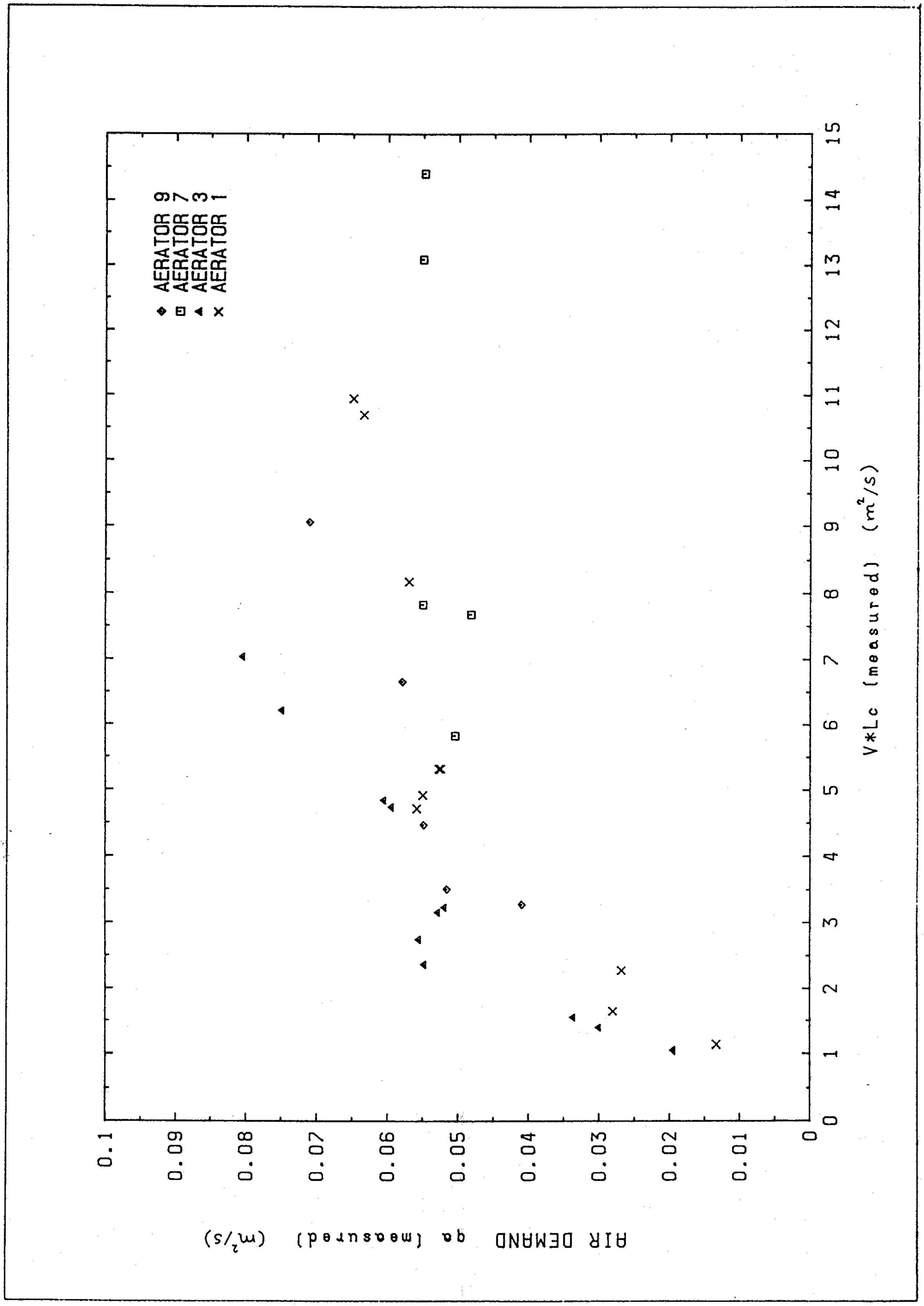
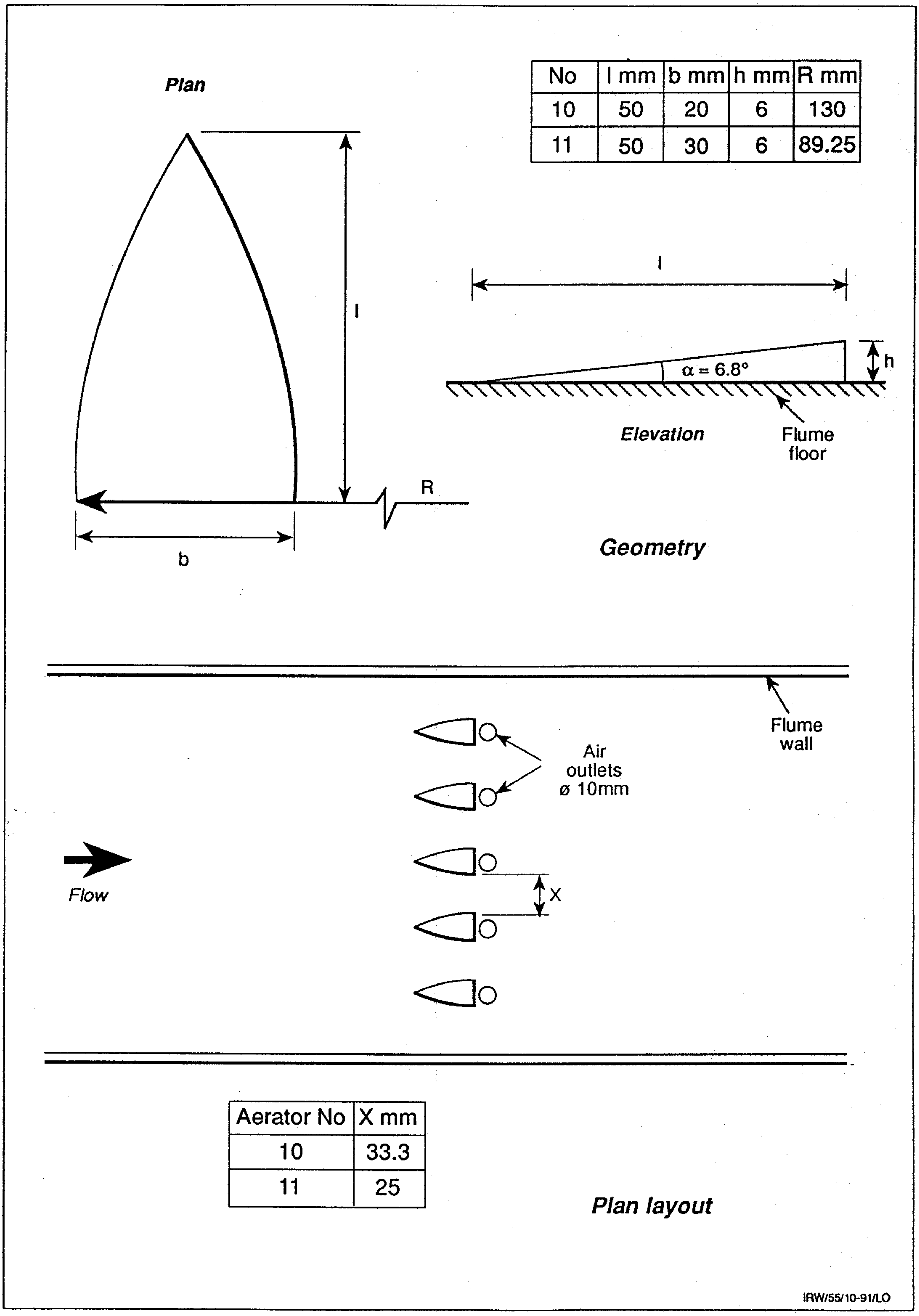


Fig 54 Variation of air demand with velocity and cavity length

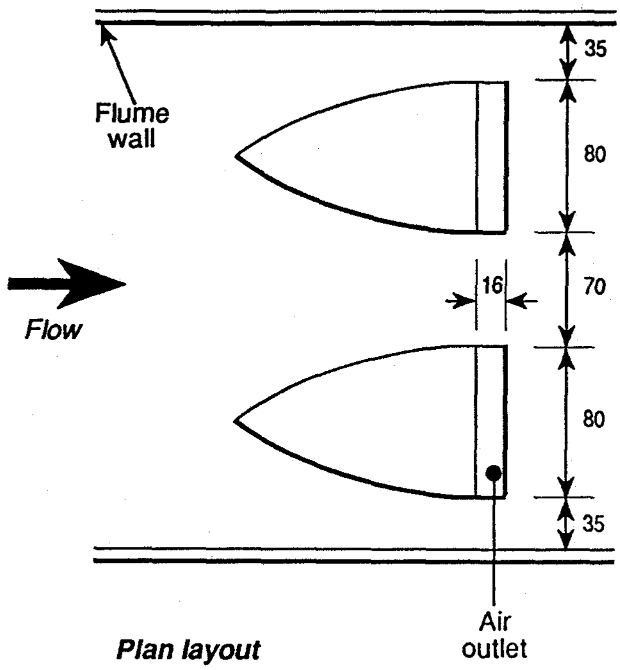


No	l mm	b mm	h mm	R mm
10	50	20	6	130
11	50	30	6	89.25

Aerator No	X mm
10	33.3
11	25

Fig 55 Geometry and layout of Aerators 10 and 11

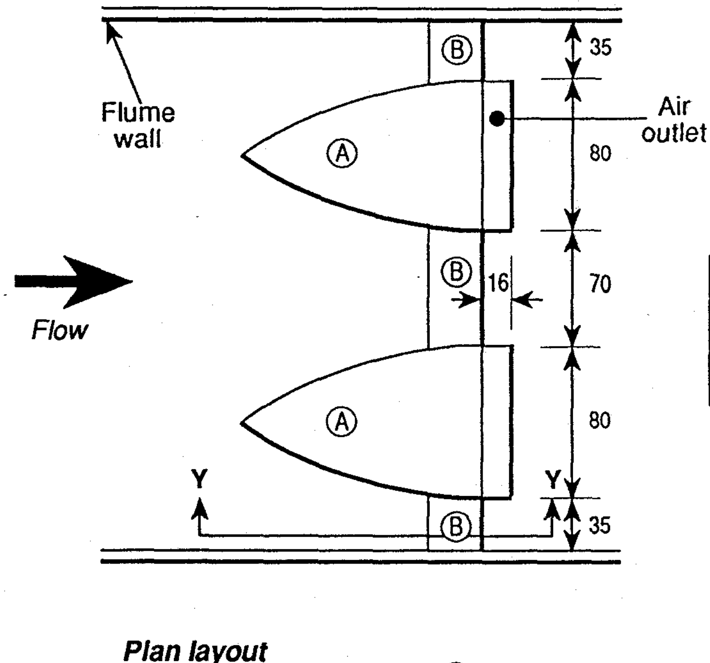
Not to scale



l mm	b mm	h mm	R mm
133.3	80	16	245

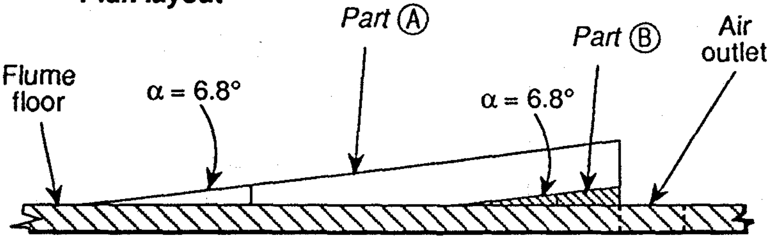
Geometry (Ref Fig 55)

Aerator 12



Part	l mm	b mm	h mm	R mm
Ⓐ	133.3	80	16	245
Ⓑ	33.3	35/70	4	n/a

Geometry (Ref Fig 55)



Section Y-Y

Aerator 13

IRW/56/10-91/LO

Fig 56 Geometry and layout of Aerators 12 and 13

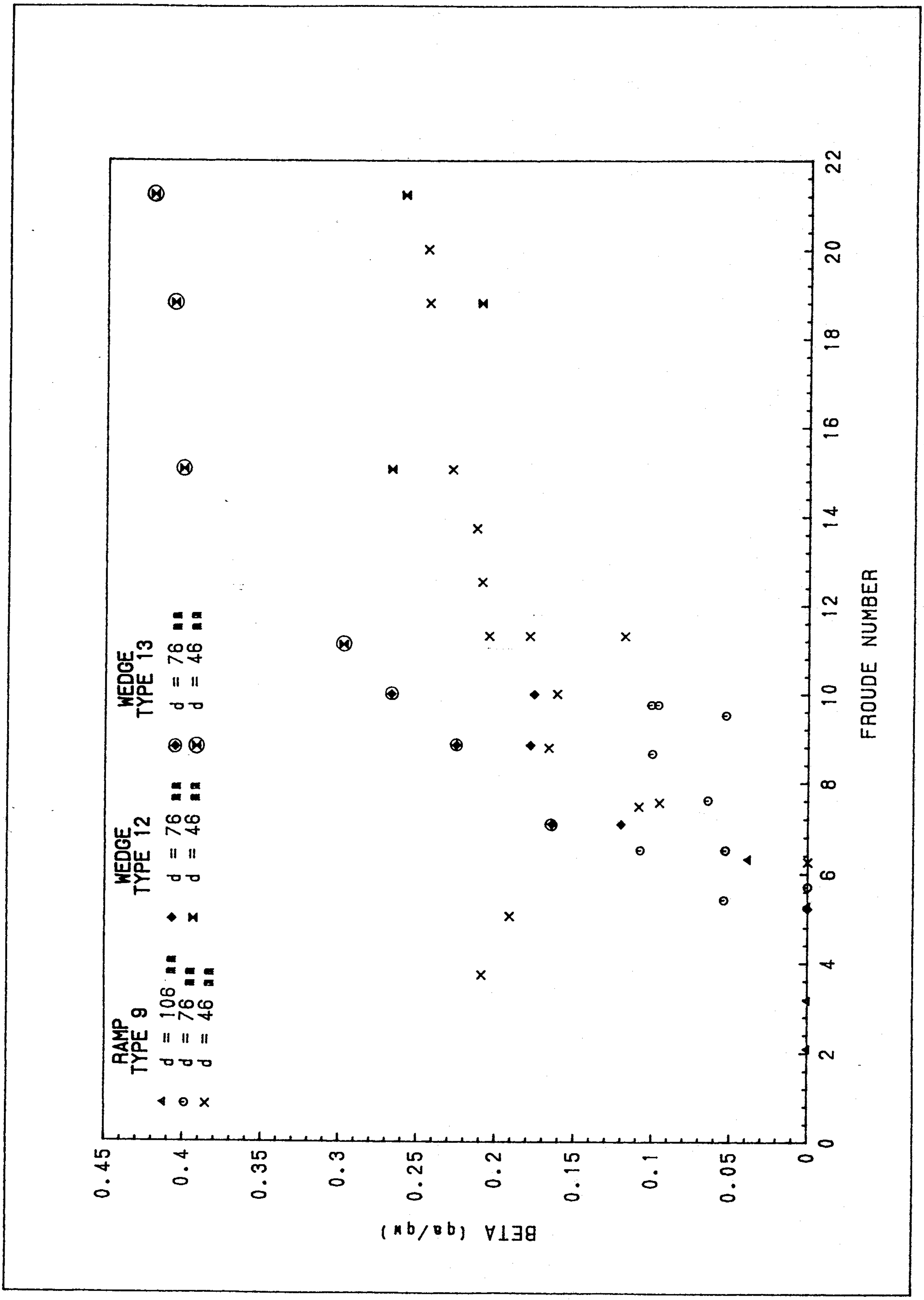


Fig 57 Comparison of air demands for wedges and ramps

PLATES

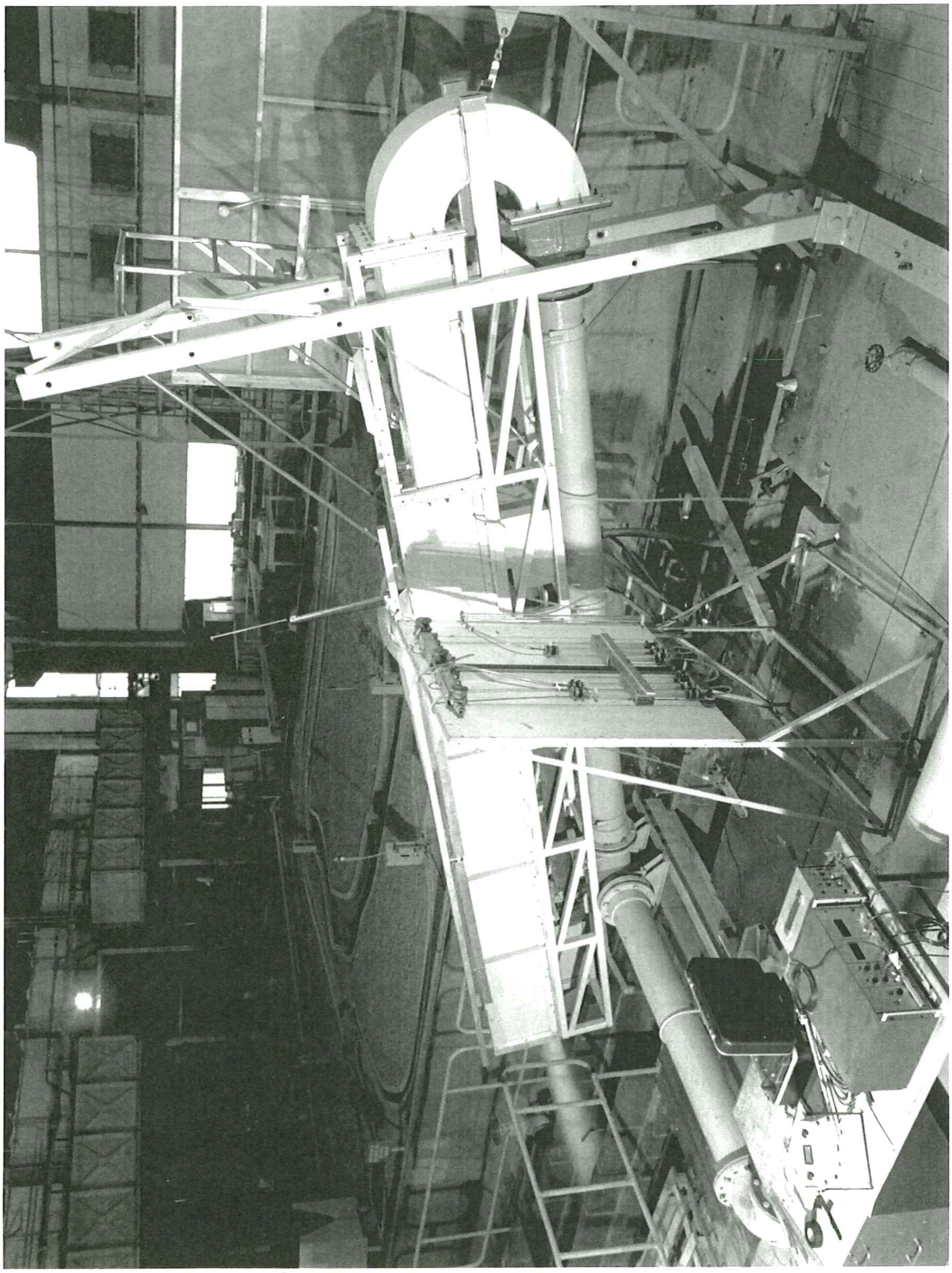


Plate 1   High-velocity flume

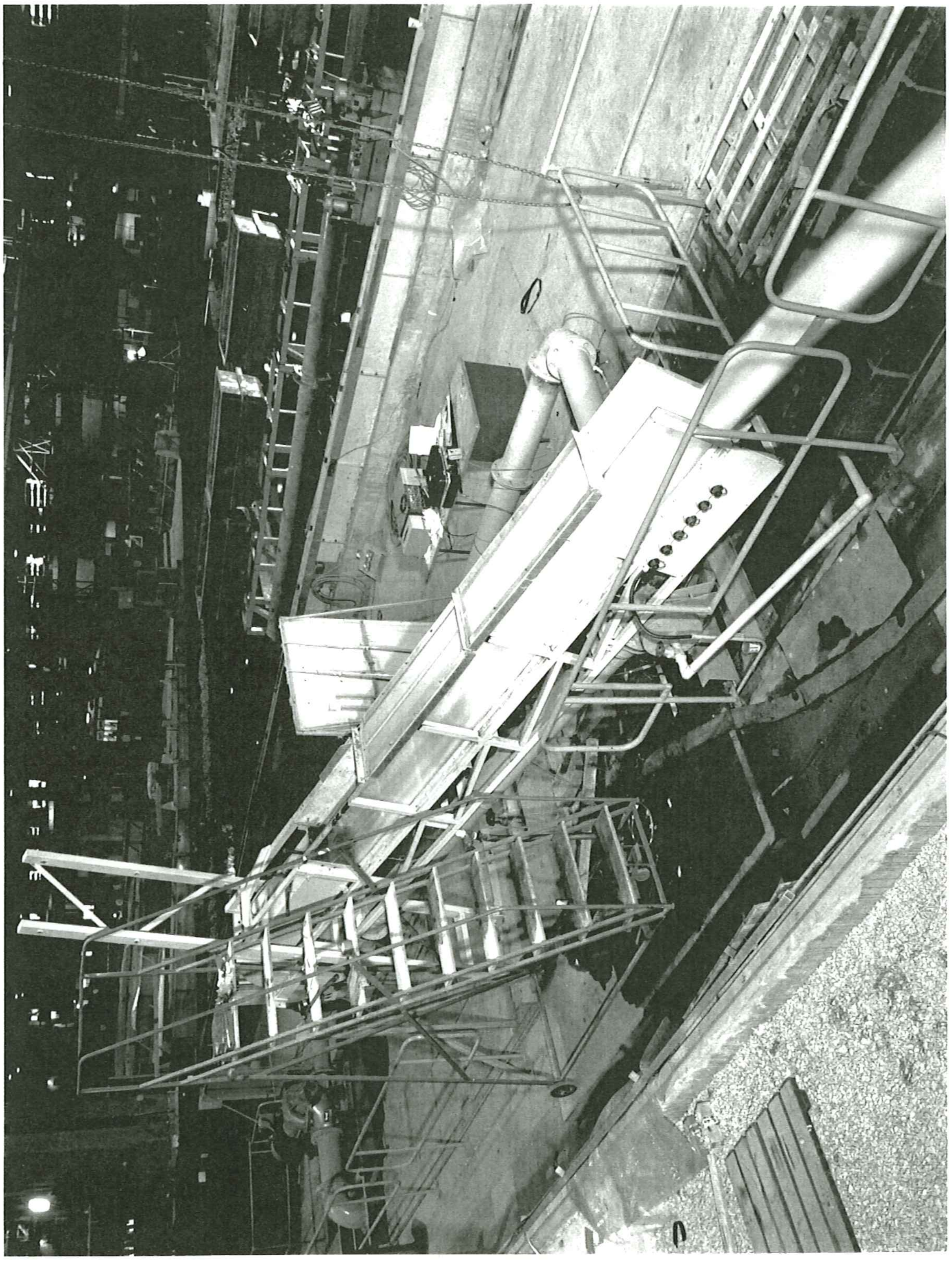


Plate 2   High-velocity flume

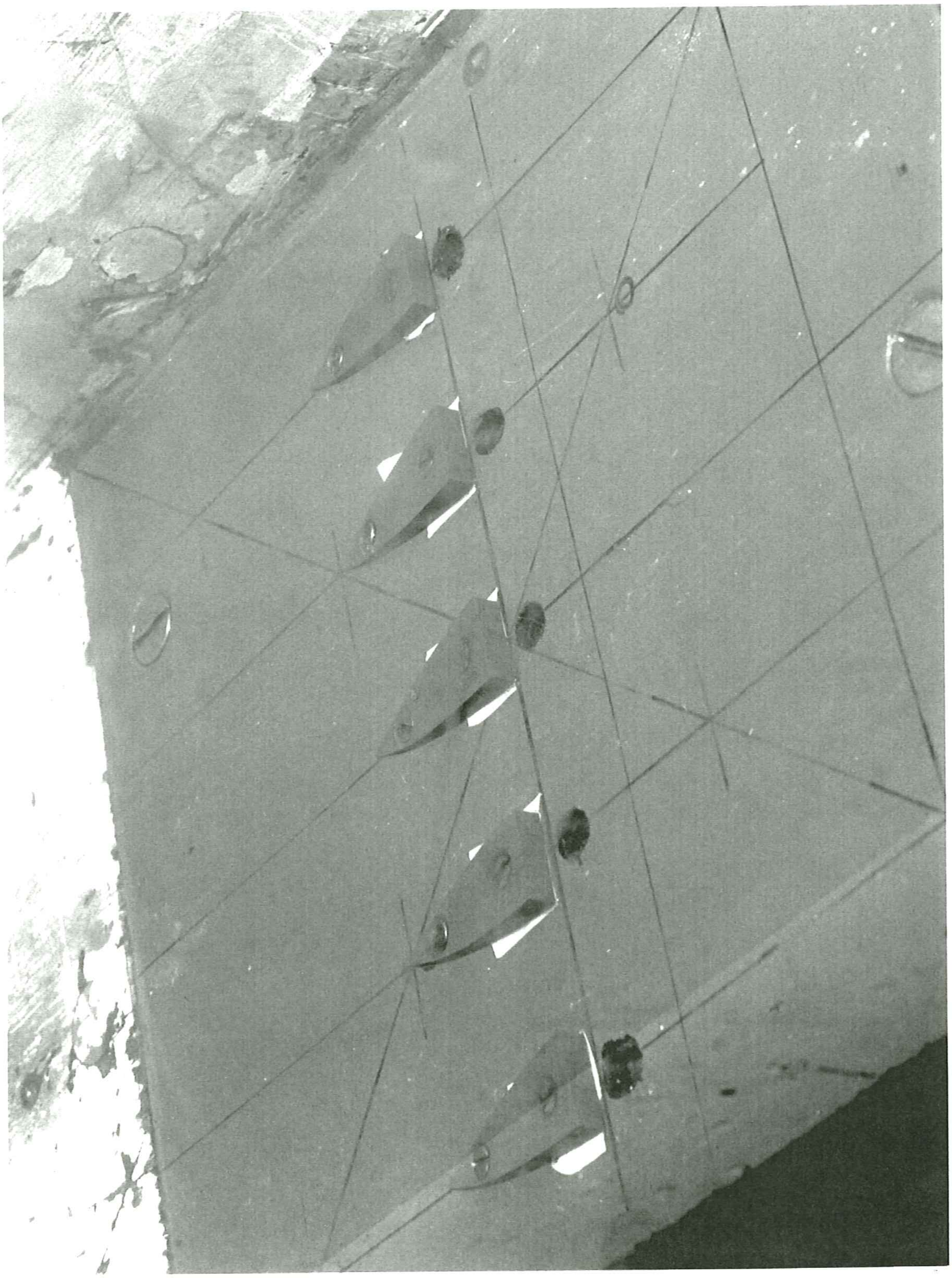


Plate 3   Type 10 wedge-shaped aerators

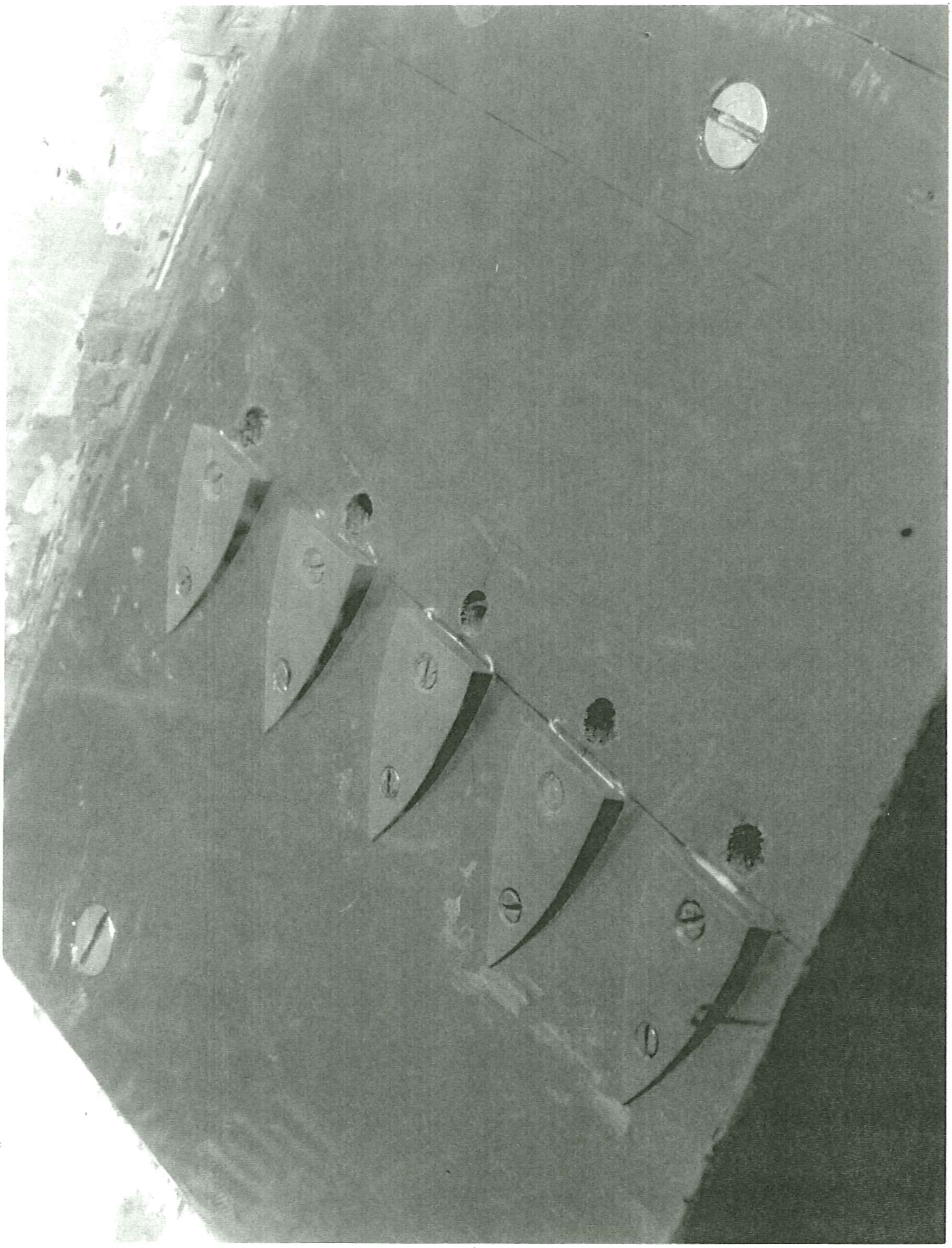


Plate 4 Type 11 wedge-shaped aerators

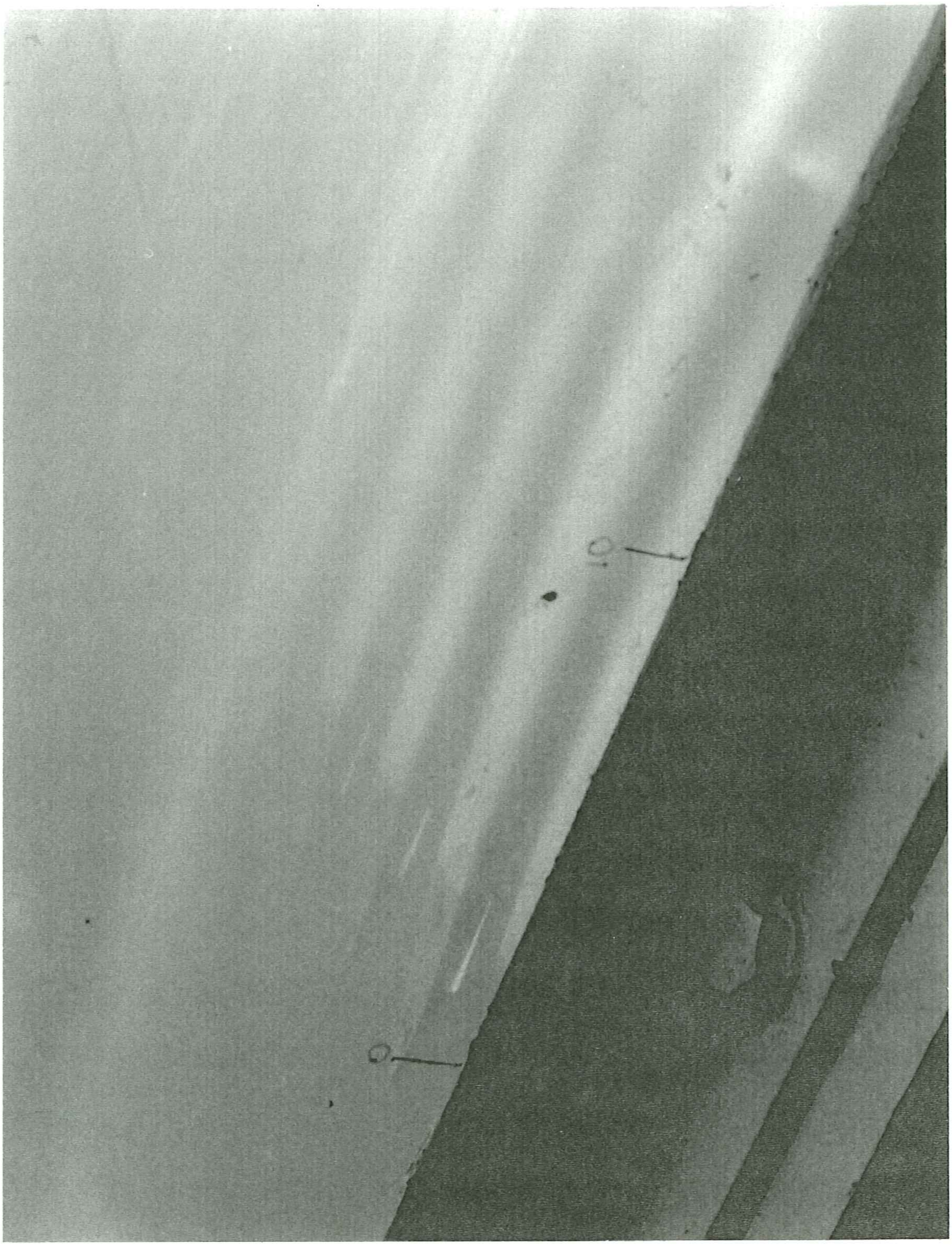


Plate 5   Type 10 aerators in operation

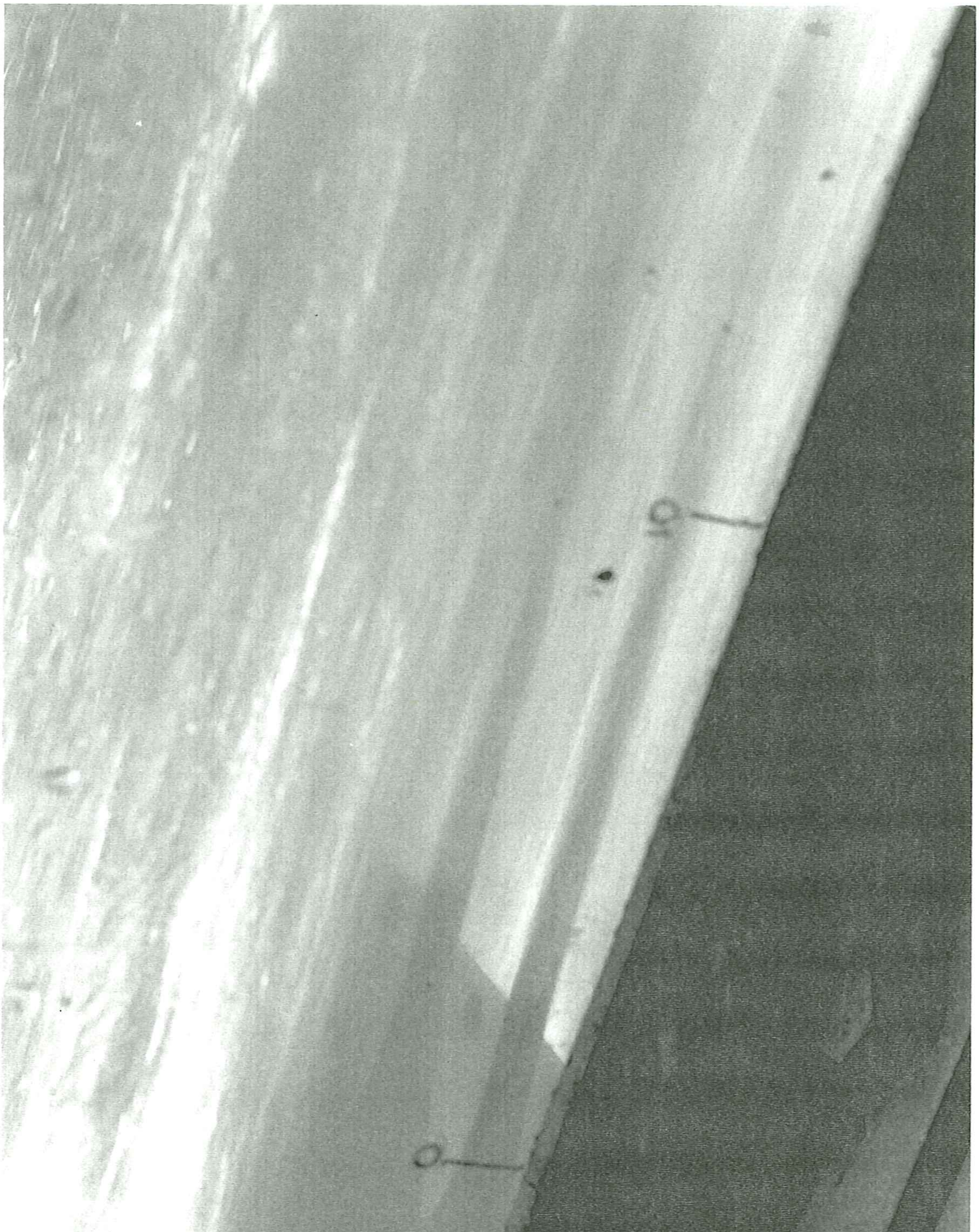


Plate 6   Type 11 aerators in operation

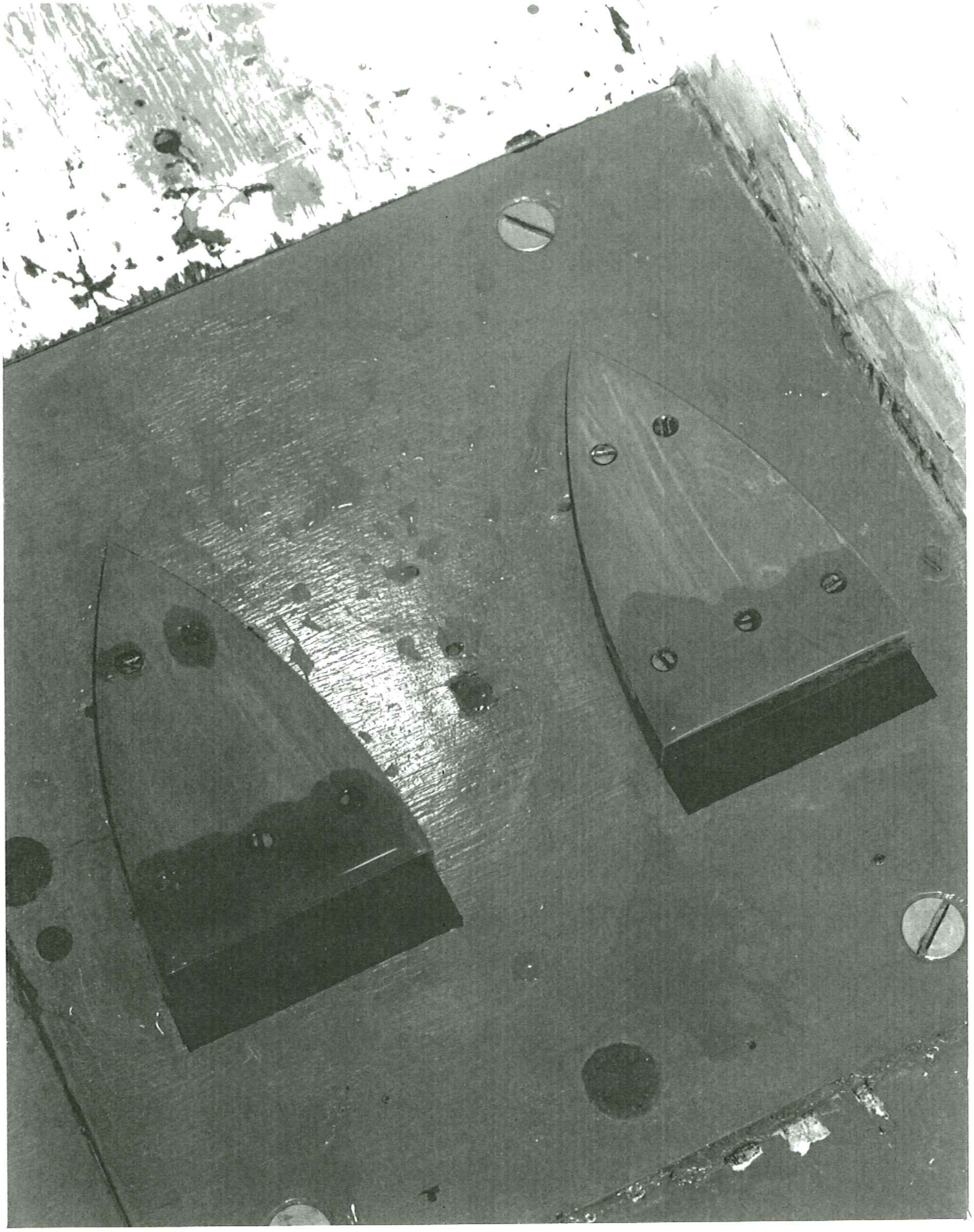


Plate 7   Type 12 wedge-shaped aerators

Plate 8   Type 12 aerators in operation

APPENDIX

APPENDIX A : Calculation of J and M for air supply system

The conveyance parameters J and M of the air supply system are related to the rate of air flow Q_a and the pressure drop Δp in the air cavity by

$$Q_a = J\, A_a\, (\Delta p/\rho_a)^{½} \tag{A.1}$$

$$M = \frac{J\, A_a}{B\, h_1} \tag{A.2}$$

where A_a is the outlet area of the duct serving width B of the spillway.

Assume that the supply system consists of a series of straight ducts or pipes, and a series of transitions, bends etc. Let an individual duct or pipe have a cross-sectional area A_n, flow perimeter P_n and length L_n with a friction factor λ_n. Similarly, let a transition, bend etc have a cross-sectional area A_m and a corresponding "point" loss coefficient C_m. The pressure drop between atmosphere and the air cavity is then given by

$$\frac{\Delta p}{\rho_a} = ½\, Q_a^2 \left[\frac{1}{A_a^2} + \sum_m \left(\frac{C_m}{A_m^2} \right) \right] + \frac{Q_a^2}{8} \sum_n \left(\frac{\lambda_n P_n L_n}{A_n^3} \right) \tag{A.3}$$

Substituting in Equation (A.1) then gives

$$J^2 = \frac{\quad}{1 + \sum_m C_m (A_a/A_m)^2 + 0.25 \sum_n \lambda_n (P_n L_n/A_a)\ (A_a/A_n)^3} \tag{A.4}$$

The unity term appearing in the denominator on the right-hand side of Equation (A.4) represents the kinetic energy of the air entering the cavity ; the maximum possible value of J is therefore $\sqrt{2} = 1.414$.

Values of the point loss factors C_m (including the inlet from atmosphere to the supply system) can be found in standard texts such as Miller (1990) and Idelchik (1986). The friction factor λ_n of a duct depends on the Reynolds number of the flow ($= 4Q/P_n\nu_a$) and its relative roughness ; values can be determined from suitable resistance equations such as the Colebrook-White equation.

High air velocities in the supply system should be avoided because they can cause compressibility problems and objectionable noise. Falvey (1980) recommends maximum velocities of 30 m/s for continuous operation and 90 m/s for short durations.