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SMOKE EXTRACTION BY ENTRAINMENT INTO A DUCTED WATER SPRAY

by

H P Morgan and M L Bullen

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FIRE
RESEARCH
STATION

51051

Fire Research Station
BOREHAMWOOD
Hertfordshire WD6 2BL

Tel: 01 953 6177

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SUMMARY

This report presents a smoke extraction system which has no moving parts in the hot smoky gases, employing momentum transfer from a high velocity water spray in a duct to extract smoke. The gas velocity for different duct configurations and water pressures was measured in an experimental rig. A theory was developed to explain the experimental results and to enable the performance of practical smoke extraction systems to be predicted.

Department of the Environment and Fire Offices' Committee
Joint Fire Research Organization

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1. INTRODUCTION

There is at present a trend towards large covered shopping complexes which may present serious problems in the event of a fire. To permit safe evacuation it is important that the spread of smoke be minimised, but it is expensive to provide a sufficiently large capacity in a mechanical system (eg fans) for removing smoke from a fire. Since the gases may be at an elevated temperature the moving parts of fans might be damaged. There is therefore a need for a cheap, simple system which can withstand high temperatures and extract smoke effectively.

A possible method of achieving this is by means of air entrainment into a water spray situated in a duct. By transferring momentum from a high velocity fluid stream to a slower one an extraction system can be produced with no moving parts in the hot gases. This principle is well established and has many applications, particularly in high-vacuum work and for air extraction in power station generating sets[1].

In the situation described here, a system was required to extract hot gases from a fire compartment through an intumescent-coated honeycomb fire damper. A simple constant cross-section jet pump consisting of a water spray in the ducting leading off the fire compartment was found to be capable of extracting gases through a 0.23 m (9 in) x 0.23 m (9 in) damper at velocities of up to 20 m/s. The performance of the extraction system was investigated and a theory developed to predict the characteristics of this apparatus and give an indication of how well the system might work in practice with larger duct sizes. The method has been found to be reasonably effective and could have applications in other kinds of buildings where smoke might have to be extracted.

2. APPARATUS

The apparatus is shown schematically in Fig. 1. The spray nozzle was a proprietary swirl type and was fitted along the centre-line of a 0.3 m (1 ft) diameter circular open ended pipe. This was angled at 30° to the horizontal to allow the water to drain away. The water drained into a tank supplying a portable water pump which was capable of producing a pressure of up to 13.5 bar (200 psi) at the spray nozzle. For the honeycomb extraction tests a horizontal 0.23 m (9 in) square section duct leading into the fire compartment was bolted to the circular pipe. The square section duct was removed for the tests on the extraction characteristics of the system and an additional 1 m (3 ft) length of circular pipe added to the inlet end of the entrainment section, giving an overall length of 4 m (13 ft).

The frictional load on the extraction system was varied by means of orifice plates placed over the inlet to the duct. The static air pressure difference between the region immediately upstream of the spray and atmosphere was measured by a pressure tapping at that point and a U-tube manometer. The air velocity in the duct was measured by an electronic vane anemometer which was also situated just upstream of the spray.

An alternative entrainment section was also tried, consisting of a conical divergent section with an angle of divergence just less than the cone angle of the spray. It was felt that this geometry would give an increased efficiency as splashing on the sides of the duct would be minimised, and that the entrainment process would be improved because the air velocity would be reduced.

3. EXPERIMENTAL PROCEDURE

Readings of air pressure difference and velocity were taken over the range of water pressure supplied to the nozzle. The readings were repeated for different inlet orifice sizes and also with the inlet completely blocked. The conical entrainment pipe was then fitted and the test procedure repeated. All these tests were carried out using air at ambient temperature.

4. THEORY

A key to the symbols used is given in section 11.

A complete theory has not yet been produced for liquid/air jet pumps[1]. The theory given here models the momentum exchange between liquid drops and the surrounding air by friction.

For any body moving in a fluid a drag coefficient can be defined as[2]

$$C_D = \frac{D}{\frac{1}{2}\rho_a u^2 S} \tag{1}$$

hence for a sphere of diameter d

$$D = C_D . \tfrac{1}{8}\rho_a \pi d^2 u^2 \tag{2}$$

and if the force on the air due to the drag on a water drop is defined as being positive

$$D = \tfrac{1}{8} \rho_a u^2 \pi d_w^{\ 2} C_D \tag{3}$$

where u is the velocity of the water drop relative to the air. The following relations are used to transform equation (3) into a stationary frame of reference

$$\begin{aligned} u &= v - V & ds &= dx - Vdt \\ du &= dv \ (V = \text{constant}) & t &= t \\ s &= x - Vt & dt &= dt \end{aligned} \tag{4}$$

Thus $$D = \tfrac{1}{8} \rho_a (v - V)^2 \pi d_w^{\ 2} C_D \tag{5}$$

The number of drops in a length δx of pipe is:

$$\frac{\text{mass flow rate}}{\text{mass of 1 drop}} \text{ x time taken for drops to travel } \delta x.$$

$$= \frac{\dot{m}\, dt}{4/3 \pi \frac{d_w^{\ 3}}{8} \rho_w}$$

$$= \frac{6\, \dot{m}}{\pi \rho_w d_w^{\ 3}} \quad \frac{dx}{v} \tag{6}$$

The total force exerted by the drops on the air in a distance δx.

$$= \delta E \quad = \sum D = D \text{ x no. of drops}$$

$$= \tfrac{1}{8} \rho_a \pi d_w^{\ 2} C_D (v - V)^2 \ \frac{6\, \dot{m}\ dx}{\pi \rho_w d_w^{\ 3}\ v}$$

$$= \frac{3 \rho_a C_D \dot{m}}{4 \rho_w d_w} \quad \frac{(v - V)^2}{v} \ dx \tag{7}$$

Thus for an entrainment length x, the 'spray motive force' E is given by

$$E = \int_o^E dE = \int_o^x \frac{3 \rho_a C_D \dot{m} (v - V)^2}{4 \rho_w d_w\ v} dx \tag{8}$$

Consider the acceleration force on a water drop

$$F = ma = m\frac{du}{dt} = m\,u\,\frac{du}{ds} \tag{9}$$

Using equation (3) and our previous sign convention

$$-\tfrac{1}{8}\rho_a \pi d_w^2 C_D u^2 = \tfrac{1}{6}\rho_w \pi d_w^3\, u\, \frac{du}{ds} \tag{10}$$

$$\therefore \quad -\frac{3\rho_a C_D}{4\rho_w d_w}\, ds = \frac{du}{u} \tag{11}$$

and equations (4) give

$$-\frac{3\rho_a C_D}{4 d_w \rho_w}(dx - V\,dt) = \frac{dv}{(v - V)} \tag{12}$$

but equation (9) also gives

$$-\frac{3\rho_a C_D}{4 d_w \rho_w}\, dt = \frac{du}{u^2} = \frac{dv}{(v - V)^2} \tag{13}$$

Combining equations (12) and (13)

$$\frac{-3\rho_a C_D}{4 d_w \rho_w}\, dx - \frac{V\,dv}{(v - V)^2} = \frac{dv}{(v - V)} \tag{14}$$

$$\therefore \quad \int_0^x \frac{3\rho_a C_D}{4 d_w \rho_w}\, dx = \int_{v_i}^{v}\left[\frac{V}{(v - V)^2} + \frac{1}{(v - V)}\right] dv \tag{15}$$

where v_i is the velocity of the water leaving the spray nozzle.

If C_D is assumed to be constant (this will be justified later) the integration can be carried out:

$$-\frac{3\rho_a C_D}{4 d_w \rho_w}\, x = \left[\ln(v - V) - \frac{V}{(v - V)}\right]_{v_i}^{v}$$

$$-\frac{3\rho_a C_D}{4 d_w \rho_w}\, x = \ln\left(\frac{v - V}{v_i - V}\right) \quad \frac{V(v_i - v)}{(v - V)(v_i - V)} \tag{16}$$

Substituting dx from (14) into (8) gives

$$E = -\int_{v_i}^{v} \frac{\dot{m}(v-V)^2}{v}\left[\frac{v}{(v-V)^2}\right] dv \tag{17}$$

$$= -\dot{m}\int_{v_i}^{v} dv \tag{18}$$

$$= \dot{m}(v_i - v) \tag{19}$$

The force resisting flow in the pipe is equal to $\Delta p.A$ (which is equal to the spray motive force) since there is no acceleration of the air.

$$\therefore \quad E = \Delta p.A \tag{20}$$

For turbulent flow, the pressure drop along a length of pipe can be represented by D'Arcy's formula for a circular cross-section pipe:

$$\Delta p = \frac{2\rho_a f l_e}{d} V^2 \tag{21}$$

Combining equations (19) and (21)

$$\frac{2 f l_e \rho_a A}{\dot{m}d} V^2 = v_i - v \tag{22}$$

Thus eliminating v from equations (16) and (22) gives

$$\beta = \frac{V(v_i + \alpha V^2 - v_i)}{(v_i - \alpha V^2 - V)(v_i - V)} - \ln\left(\frac{v_i - \alpha V^2 - V}{v_i - V}\right) \tag{23}$$

$$= \frac{\alpha V^3}{\gamma} - \ln\left(\frac{\gamma}{(v_i - V)^2}\right) \tag{24}$$

where

$$\left.\begin{aligned} \alpha &= \frac{2 f l_e \rho_a A}{\dot{m} d} \\ \beta &= \frac{3\rho_a C_D x}{4\rho_w d_w} \\ \gamma &= (v_i - V)(v_i - \alpha V^2 - V) \end{aligned}\right\} \tag{25}$$

Since v_i and $\dot{m}$ are known in terms of the water pressure (p_w) for a particular nozzle, a solution of V against p_w for different pipe systems can be obtained by solving equation (24). The solution of this equation is by no means easy as conventional numerical techniques cannot be used. The function is not defined for $v < V < v_i$ and with $V > v_i$, the solution has no practical significance for the problem considered here. Iterative methods, such as the Newton-Raphson method may give incorrect answers or break down if the iteration takes place partly or wholly in the non-analytic region. However, when graphs of β against V were plotted in the region of interest $(0 < V < v)$ for different values of p_w the curves were found to be continuous, and increasing monotonically. It was therefore possible to apply a simple iterative technique of incrementing V and evaluating the right-hand side of equation (24) until it became equal to β. This procedure was carried out by computer and proved satisfactory except that many more iterations were required than would be needed by more standard numerical methods giving a faster convergence.

The initial water velocity v_i is given by

$$v_i = \frac{R}{\dot{m}} \qquad \text{(conservation of momentum)} \tag{26}$$

R and m were found experimentally in terms of p_w :

$$R = k_1 \, p_w \qquad \text{where} \quad k_1 = 0.0001 \; (p_w \text{ in N/m}^2) \tag{27}$$

$$m = k_2 \sqrt{p_w} \qquad \text{where} \quad k_2 = 0.0038 \tag{28}$$

$$\therefore \quad v_i = 0.026 \sqrt{p_w} \tag{29}$$

Inlet and outlet head losses were calculated for the different pipe configurations by taking the velocity head loss to be $\frac{1.0 \, V^2_{IN}}{2g}$ for the inlet and $0.5 \, \frac{V^2}{2g}$ for the outlet. The sum of these losses was equated to the pressure loss given by equation (21). For example, for the experimental ducting with a 0.205 m (8 in) inlet orifice and 4 m (13 ft) length of 0.305 m (1 ft) diameter pipe:

$$\frac{2 \, f \, l_e \, V^2}{d} = \frac{1.0 \, V^2_{IN}}{2} + \frac{0.5 \, V^2}{2}$$

$$V_{IN} = \frac{AV}{A_{IN}}$$

$$\therefore \quad l_e = \frac{d}{4f} \left(\frac{1.0 \, A^2}{A^2_{IN}} + 0.5 \right)$$

$$= 71.1 \text{ m} \qquad \text{with } f = 0.006$$

This is added to the actual length of 4 m (13 ft) which gives a total length of 75.1 m (243 ft). The total effective lengths for the different inlet conditions are given in Table 1.

Table 1

Theoretical effective lengths of ducting

Orifice diameter (mm)	50	100	150	205	255	305
Duct effective length (m)	16480	1040	214	75	37	24

Equation (24) was solved using the following values:

$f = 0.006$

$\rho_a = 1.2\ \mathrm{kg/m^3}$

$x = 2.0\ \mathrm{m}$

$d_w = 1$ mm, a typical value given by Rasbash and Stark[3]

The drag coefficient C_D was taken to be constant and of value 0.4. This is justified by considering the Reynolds number of the spray drops. As the drops slowed from their initial velocity their Reynolds number $\left(\frac{(v - V)d_w}{\nu}\right)$ fell from $\sim 10^4$ to $\sim 10^3$. Figure 2 shows the variation of C_D with Re for a sphere[2] and it can be seen that C_D is reasonably constant between Re = 600 and Re = 10^5.

The theoretical air velocity vs. water pressure curves for each orifice size are shown in Fig. 3.

A special case of the above analysis which is of theoretical interest is when the inlet is completely blocked. The spray motive force can be equated to the force due to the pressure difference developed across the spray.

$$E = \Delta p\, A \qquad (30)$$

Equation (16), with $V = 0$ becomes

$$\frac{-3\rho_a C_D x}{4 d_w \rho_w} = \ln\left(\frac{v}{v_i}\right) \qquad (31)$$

$\therefore$ Equations (19) and (31) give

$$\Delta p = \frac{\dot{m} v_i}{A}\left(1 - \exp\left(-\frac{3\rho_a C_D x}{4\rho_w d_w}\right)\right) \qquad (32)$$

Thus the suction pressure (fore-pressure) can be evaluated for the no-flow condition and compared with the experiment (Fig. 4).

An analysis of the behaviour of the divergent entrainment pipe was attempted, but since the air velocity is now a function of distance, integration of equation (14) becomes extremely difficult.

5. RESULTS

The experimental readings for the straight and for the conical entrainment pipes are given in Tables 2 and 3 respectively. The data are plotted in Figs 3 and 5 which show the effect of the inlet orifice size on the air velocity-water pressure relationship.

(continued on page 11)

Table 2. Experimental data for straight entrainment section

p_w	Inlet orifice diameter (mm)												
	0	50		100		150		205		255		305	
(kN/m^2)	Δp (N/m^2)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)
175	48	46	0.2	34	0.7	30	1.5	25	2.5	22	3.4	15	4.1
345	170	110	0.6	85	1.3	80	3.0	55	4.3	39	6.1	30	6.8
520	290	170	0.85	145	1.6	125	3.8	95	5.9	80	7.6	50	9.4
690	430	270	1.0	240	2.25	170	4.4	120	7.6	105	9.9	85	11.4
865	550	385	1.2	335	2.5	230	5.2	175	8.6	135	11.9	110	13.4
1030	650	595	1.3	530	3.0	420	6.2	320	10.2	205	13.2	150	15.5
1380	820	750	1.5	700	4.1	630	7.6	470	12.7	300	15.7	220	19.5

Table 3. Experimental data for conical entrainment section

p_w	Inlet orifice diameter (mm)												
	0	50		100		150		205		255		305	
(kN/m^2)	Δp (N/m^2)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)	Δp (N/m^2)	V (m/s)
345	110	100	0.7	95	1.2	85	2.5	70	3.8	60	5.8	50	7.6
690	320	300	0.9	290	2.2	250	4.4	190	6.9	170	10.7	130	14.0
1030	600	560	1.3	510	3.0	440	5.8	350	9.1	280	14.5	230	17.7
1380	750	720	1.8	710	3.8	620	6.6	510	11.2	400	17.0	325	21.3

(continued from page 8)

An 'efficiency' for the entrainment process can be defined in terms of the momentum transferred to the air, ie:

$$\eta = \frac{\dot{M}V}{\dot{m}v} \times 100 \text{ per cent}$$

$$= \frac{100\, \rho_a\, A\, V^2}{k_1\, p_w} \quad \text{per cent}$$

This relationship was evaluated for the experimental readings for the straight entrainment pipe and is plotted in Fig. 6.

6. DISCUSSION

In general the results show that although this method of extraction does not have a high 'mechanical' efficiency (Fig. 6) it is nevertheless an effective means of extracting hot gases in certain situations. For example, a water flow of 4.5 l/s (60 gal/min) with a pressure at the nozzle of 14 bar (200 psi) in a 300 mm (1 ft) diameter tube could extract up to 1.5 m^3/s (3180 ft^3/min). To extract the gases from a real fire in a shopping mall requires an extraction rate of say 12 kg/s (26.5 lb/s) and one could use a water flow of 12 l/s (160 gal/min) at a pressure of 11 bar (160 psi) in a 1.125 m^2 duct, such water flows and pressures being easily obtainable in practice.

The theory which has been developed to model the behaviour of the system gives a good agreement with the experimental results within the limits of the apparatus and experimental error and the assumptions of the theory. It should be remembered that the apparatus was designed primarily to extract hot gases through a honeycomb rather than to test the theory.

Figure 4 showing the suction characteristics of the pump provides some useful information. Firstly, it shows that equation (32) gives a good agreement with the experimental results. The discrepancy between the two lines is almost certainly due to interference between the spray and the duct walls, resulting in a loss of available water momentum, although the action of the moving gas stream on the water spray means that much less water will impinge on the walls of the duct than would be expected from the shape of the spray in the open air. Also, since the theoretical line does not lie far from the experimental line, it suggests that the evaluation of the motive force produced by the spray is valid.

Comparison of the air velocity vs. water pressure graphs (Figs 3 and 5) for the straight and conical entrainment pipes shows that the conical pipe gives a marginal improvement in performance when the resistance is low (approx. 10 per cent increase in performance in the characteristics for the situation where there is no inlet orifice). With a high resistance, ie. with small inlet orifices, there is little difference in the performance of the two systems. It is also noticeable that the suction characteristic for the diverging pipe is marginally less good than for the straight pipe. This is probable due to leakage back upstream between the jet and the walls of the duct. Qualitatively the divergent section should be more effective since the momentum transfer occurs with the air at a reduced velocity relative to the observer, and thus a higher droplet velocity (v_i - V) relative to the air, and also because splashing on the walls of duct is reduced. The two systems used show that by careful design of the interaction of the spray and the ducting the performance of the system may be improved but the precise positioning of the jet becomes more critical and the straight pipe geometry may be a better practical compromise. An alternative configuration would be a relatively small duct joined to one of larger cross-section in which the spray is situated.

Similarity between the experimental and theoretical curves is apparent (Fig. 3), several points arising from the analysis which help explain the discrepancies. The theory and experiment agree fairly well for the intermediate sizes of inlet orifice (100, 150, 205 mm) but less well for the remaining sizes. Since the analysis equated the spray motive force with the resistance to flow and the use of the spray motive force gives good results for the suction (no-flow) experiment, there might be an error in the representation of the flow resistance in the pipe (equation (21)). This is confirmed by reference to Table 4 which compares the effective lengths produced by the theory with those derived from the experimental readings of air pressure drop in the pipe. It can be seen that there are relatively large percentage differences in the values for the 50, 255 and 305 mm orifices, where the theory does not seem to have predicted practice particularly well. For the 305 mm orifice (corresponding to the inlet of the pipe being completely open) a better correlation with the experimental curve is given by using the 14.25 m (47 ft) effective length of the pipe based on the experimental pressure readings. This line is also shown in Fig. 3. It would seem that the theory works reasonably well, except for some of the extreme inlet conditions produced by the experimental arrangement, when discrepancies between the predicted and actual pipe resistances become large.

Table 4. Comparison of theoretical and experimentally derived effective duct lengths

p_w (kN/m^2)	Inlet orifice diameter (mm)					
	50	100	150	205	255	305
	Effective duct length (l_e) based on experimental readings (m)					
345	5600	1300	217	58	30	14
690	6600	1015	220	71	31	16
1030	2800	990	208	64	27	14
1380	1900	910	220	60	25	13
	Theoretical effective length (from Table 1)					
	16480	1040	214	75	37	24

D'Arcy's pipe friction formula in the form given (equation (21)) applies only to turbulent flow. As the Reynolds number based on the pipe diameter is approximately 200 at $V = 1$ m/s rising to 4000 at $V = 20$ m/s, the flow at the lower velocities is almost certainly laminar. For a laminar flow the pipe resistance is more likely to be proportional to the velocity rather than its square and the friction factor (f) will also be altered. This will lead to discrepancies at low velocities, although this error will tend to result in an underestimation of the flow rate.

The effect of air temperature on the performance must also be considered. A change in air viscosity and density will affect Reynolds number. The reduction of density will also decrease the air mass flow for a given volume flow rate. However, it was found during the extraction experiments with intumescent-coated honeycombs that the spray cooled the hot gases considerably. Qualitatively the reduction in temperature of the gases by the spray, combined with the reduced frictional losses in the duct tend to offset the reduction in mass flow rate caused by the fall in density.

In the absence of data concerning the drop diameter, the value produced by Rasbash and Stark[3] has been used. Although this may not be correct in this application and the drop size will almost certainly vary with water pressure, the air velocity V is relatively insensitive to the value of β (equation (24)) providing $\beta > \sim 0.1$. This is because the value of β implies the proportion of the water momentum transferred to the air: this is illustrated more clearly by comparing equations (25) and (32). The momentum transfer falls exponentially with x, so that if β is sufficiently large to give a reasonable transfer, any further change in β will not give much improvement in momentum transfer.

7. PRACTICAL APPLICATION

The application of the theory to other duct systems of uniform cross-section entrainment pipe can be considered by referring to equation (24). If a given water supply (known in terms of the pressure and mass flow rate) is available then p_w can be substituted directly and the mass flow rate in the form $n\dot{m}$ where n is the number of spray nozzles used and $\dot{m}$ the mass flow rate for 1 nozzle (known in terms of the water pressure for the nozzle used) can be found. The droplet size will also have to be measured or a typical size assumed. The effective length of the duct and its diameter, if it is of circular cross-section, can also be used directly. For non-circular ducting D'Arcy's formula can be re-written

$$p = \frac{f\, l\, \rho_a\, V^2}{2\,\Delta} \tag{21a}$$

where Δ = hydraulic mean depth

$$= \frac{\text{area}}{\text{perimeter}} \text{ of duct}$$

Thus for a circular duct

$$\Delta = \frac{\pi d^2}{4\,\pi d} = d/4$$ which gives the original result when substituted into (21a).

This calculation has been done for a variety of duct configurations and the design data are shown in Fig. 7. For any of the duct systems given, corresponding values of water flow rate, water pressure and air flow rate can be read. For instance; an air mass flow rate of 5 kg/s (69 gal/min) can be achieved in a 1 m^2 (10.8 ft^2) duct containing 2 nozzles, using a water flow rate of 4.2 kg/s (58 gal/min) and a pressure of 320 kN/m^2 (46 psi).

An external wind could affect the performance of the extraction system. This may be of importance in designing for a mall. The reduction in extraction velocity due to the adverse pressure of the wind for a typical duct outlet is given in Fig. 8. This shows the effect of external wind in the worst possible case: the duct outlet has no form of cowl and the wind is blowing directly into the duct outlet. For an external wind of up to 5 m/s (11 mph) the extraction velocity is reduced by less than 10 per cent.

8. CONCLUSIONS

1) Entrainment into a water spray is an effective means of extracting gases especially in fire situations where the high temperature of the gases means that a fan cannot be used without special protection.

2) The performance of the system was predicted reasonably well by the theory.

3) An indication of the performance of other systems using a parallel-sided entrainment tube can be obtained by minor modifications to the theory.

4) It may be possible to use this system to extract large quantities of smoke in situations where there is not a large adverse pressure head between inlet and outlet. The system has the advantages of having no moving parts and also of cooling the hot gases, so that further extraction can be effected by conventional means.

9. ACKNOWLEDGMENT

The idea of using a water spray in a duct to extract smoke was suggested by experiments several years ago at the Fire Research Station by Z Rogowski, who was investigating the performance of explosion relief vents in a duct.

10. REFERENCES

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11. NOMENCLATURE

A	cross-sectional area of duct	m^2
C_D	drag coefficient	
D	drag force	N
d	duct diameter	m
d_w	diameter of water drop	m
E	spray 'motive-force'	N
f	friction factor of duct	
l_e	effective length of duct	m
$\dot{M}$	mass flow rate of air	kg/s
$\dot{m}$	mass flow rate of water	kg/s
P	plan area of a body	m^2
p_w	water pressure	N/m^2
R	spray reaction force	N
Re	Reynolds number	
S	cross-sectional area of body, normal to direction of motion	m^2
s	displacement in a moving frame of reference	m
t	time	s
u	velocity of a body relative to the air stream (ie in a frame of reference travelling at the air velocity)	m/s
V	air velocity	m/s
v	velocity of a water drop	m/s
v_i	initial velocity of a water drop	m/s
x	displacement from nozzle along duct	m
Δ	hydraulic mean depth	m
Δp	pressure difference across spray	N/m^2
ρ_a	air density	kg/m^3
ρ_w	water density	kg/m^3

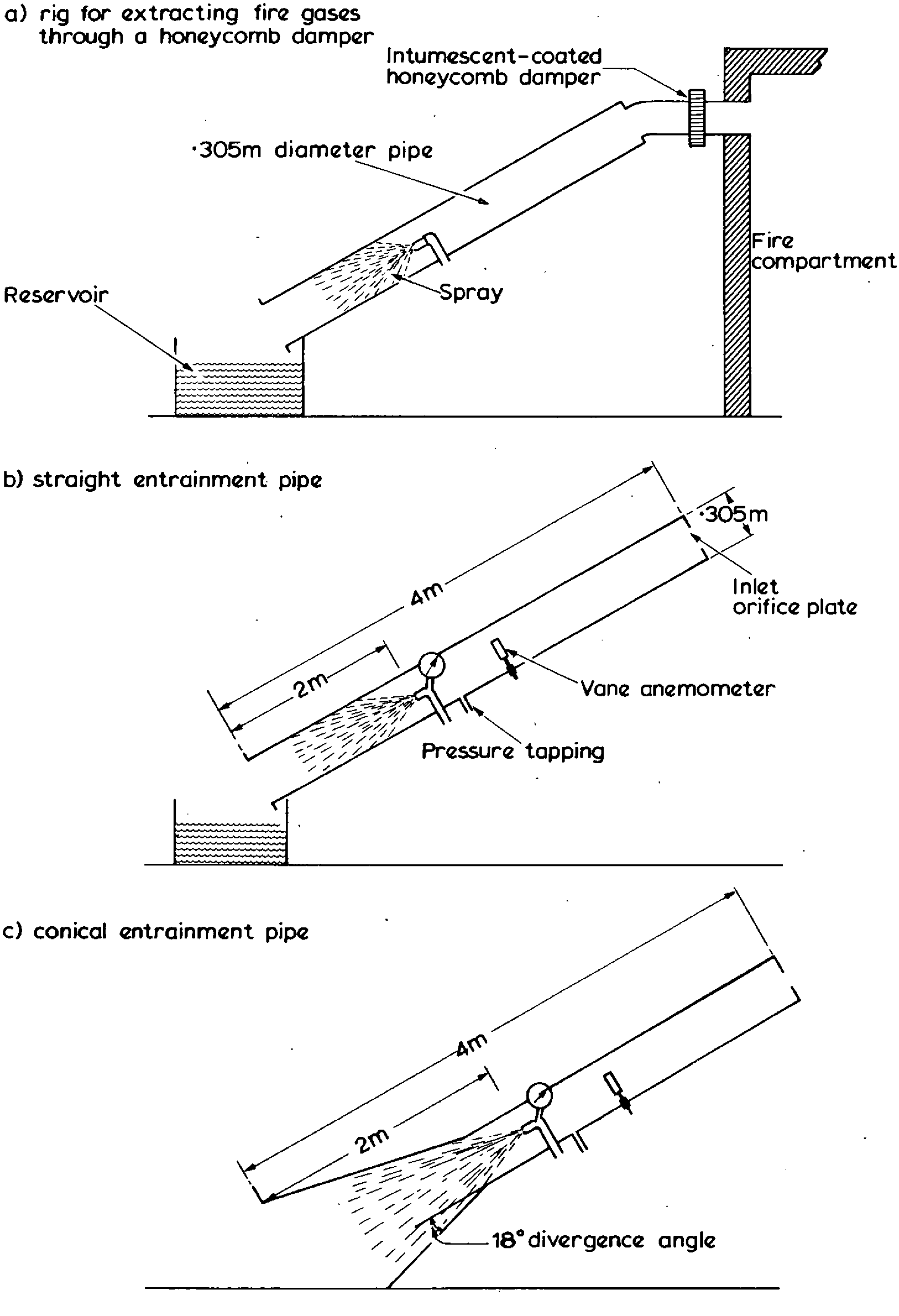


Figure 1 Arrangements of experimental apparatus

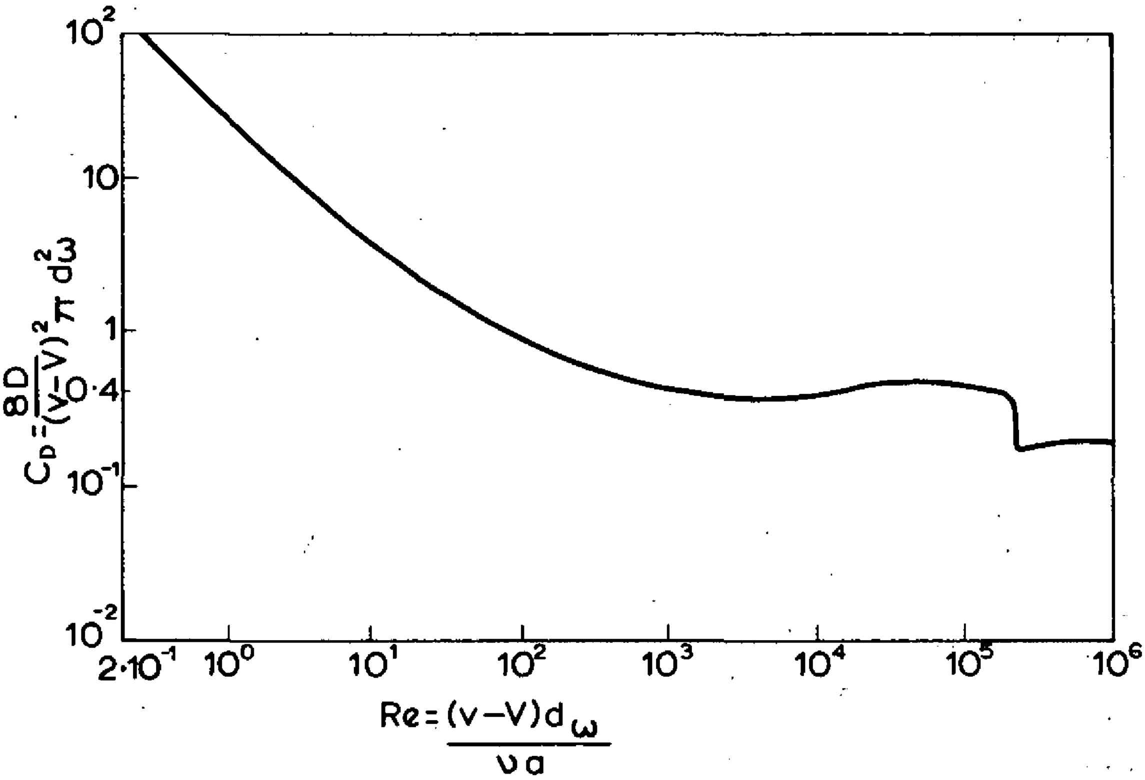


Figure 2 Variation of drag coefficient with Reynolds number for a spherical body

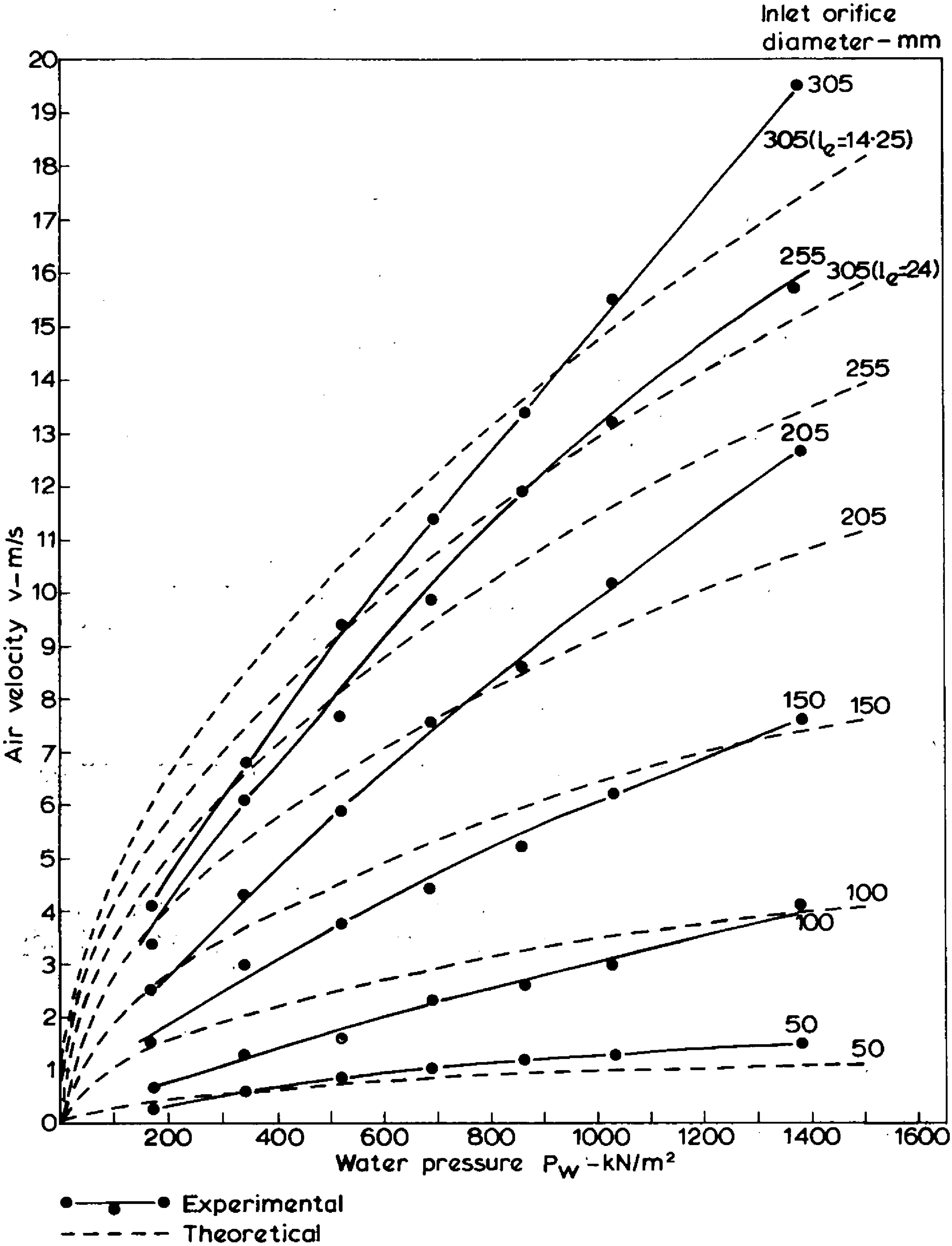


Figure 3 Theoretical and experimental flow characteristics for a straight entrainment pipe

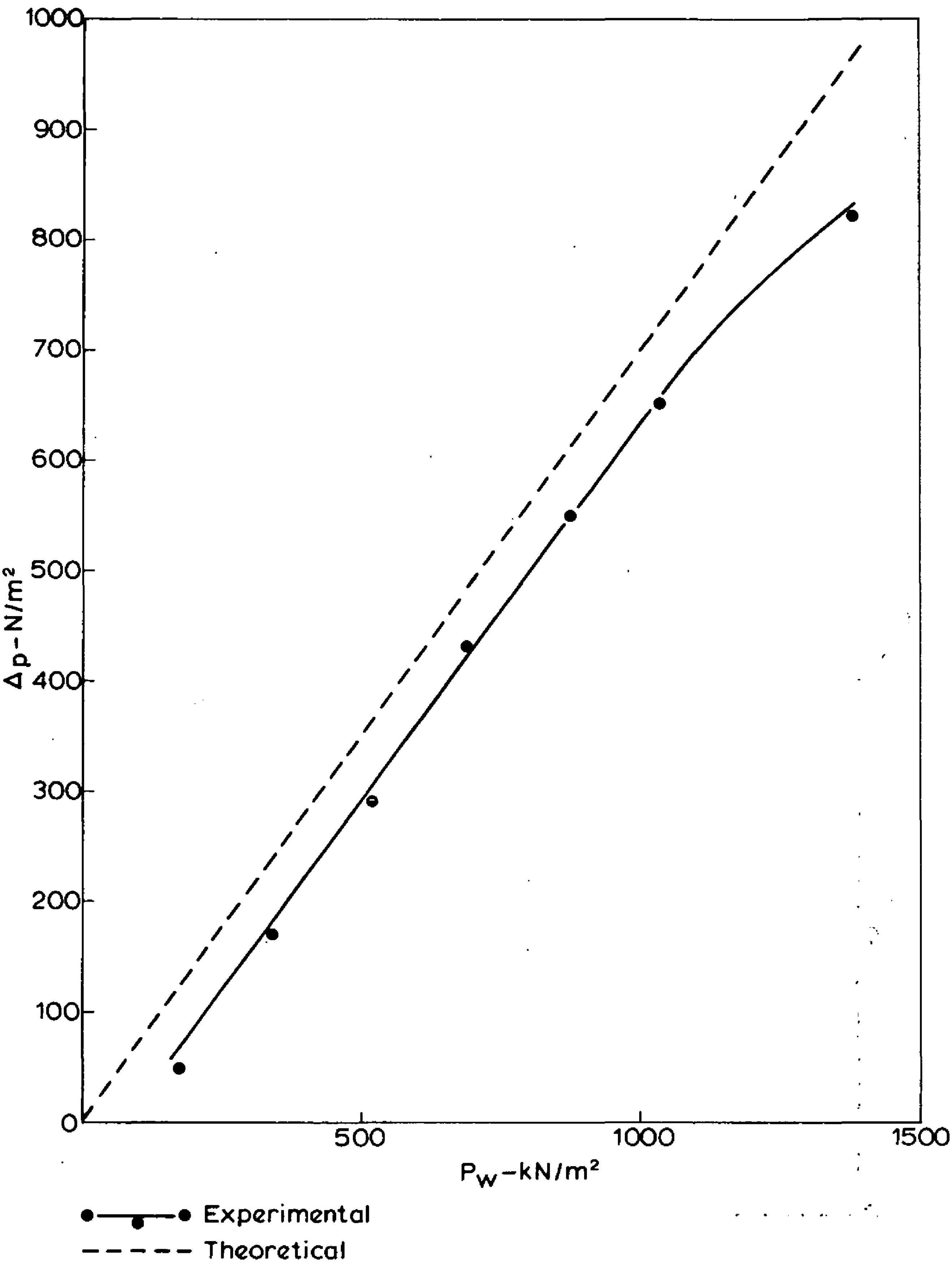


Figure 4 Suction pressure (forepressure) against water pressure (inlet closed)

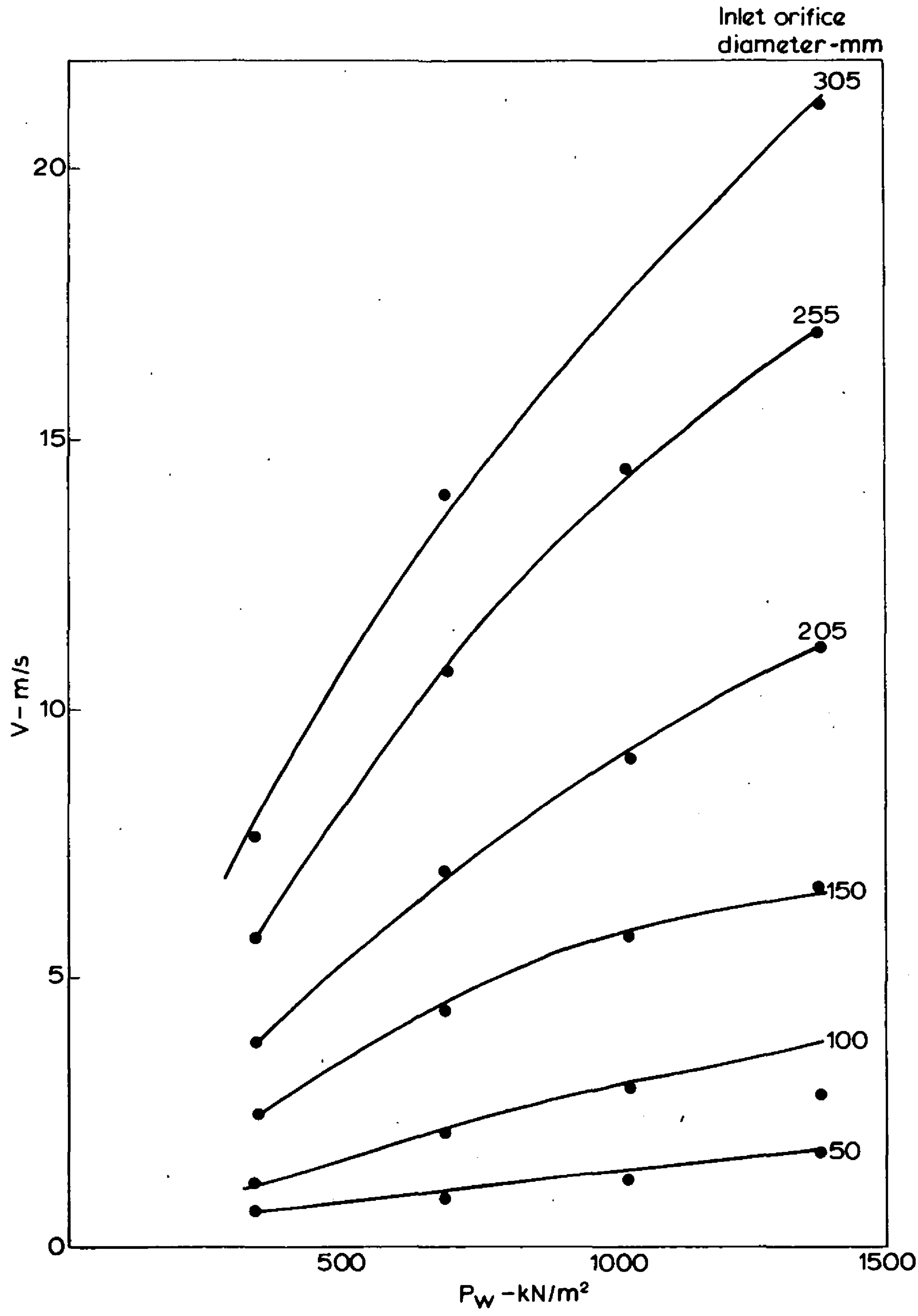


Figure 5 Experimental relationship between air velocity and water pressure for different inlet orifices and a divergent entrainment pipe

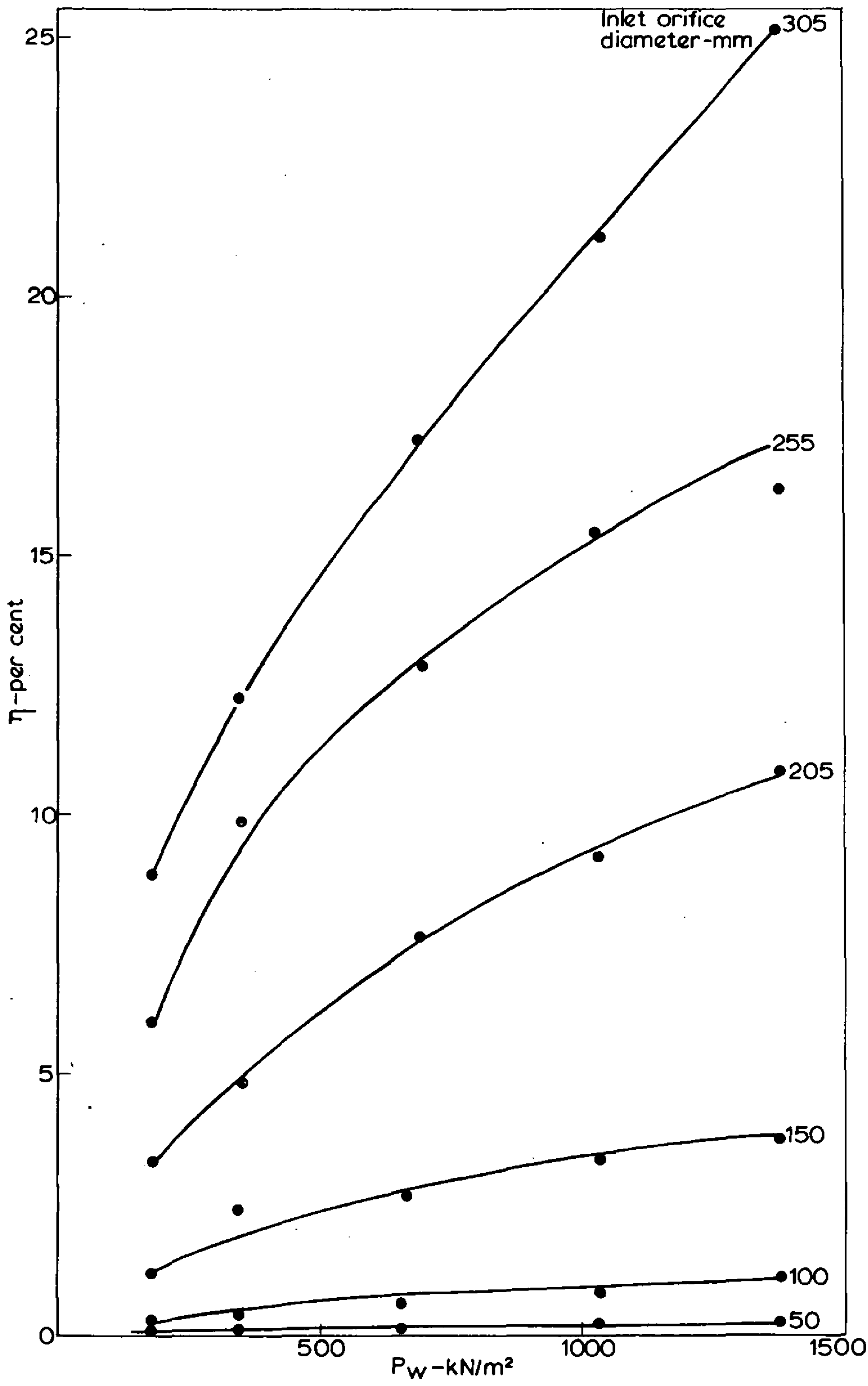
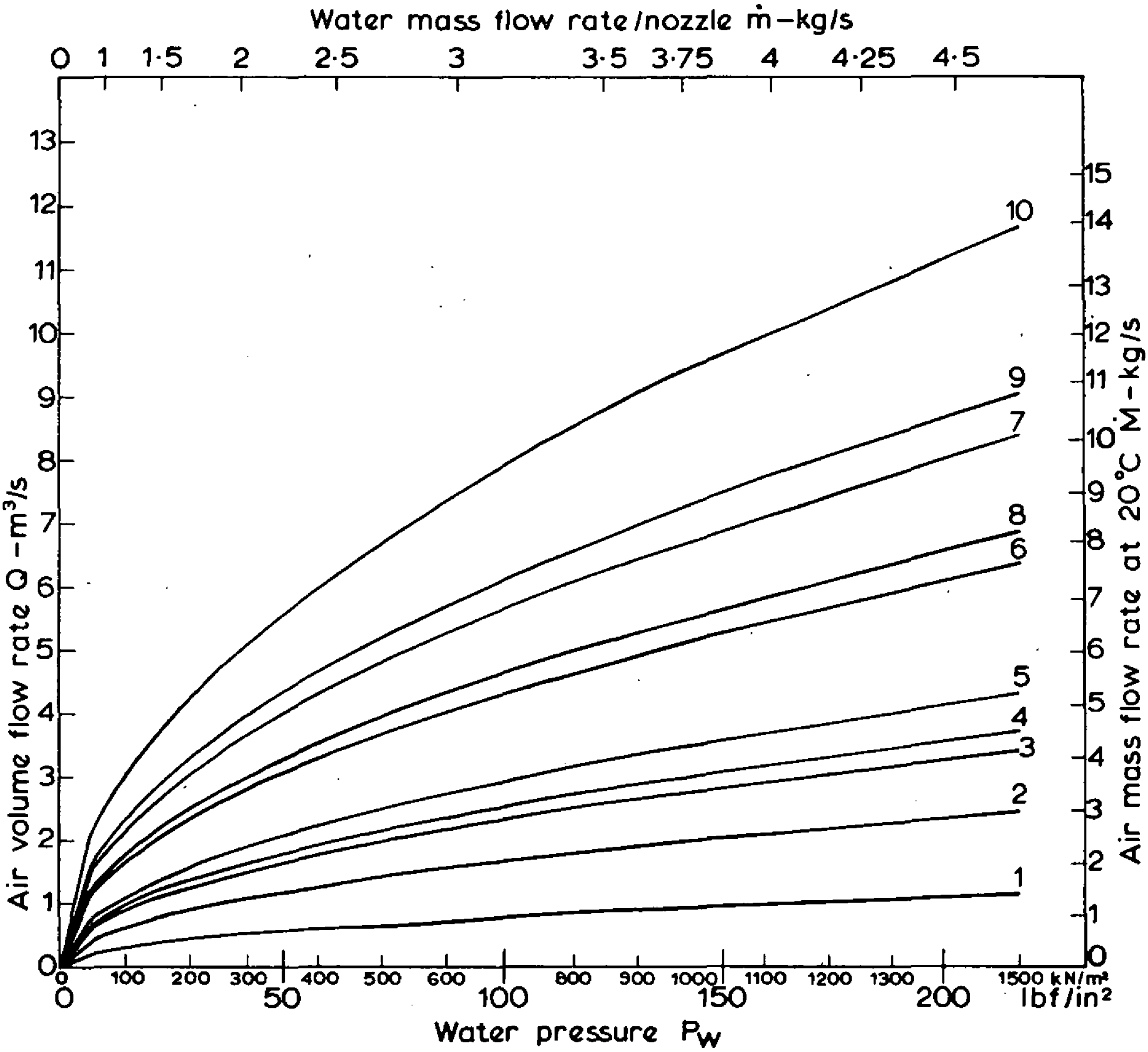


Figure 6 Efficiency against water pressure (straight entrainment pipe)



1 0·305m (1ft) square section duct, 1 spray nozzle

2 0·5m square section duct, 1 spray nozzle

3 0·5m square section duct, with 1m x 1m square entrainment section, 1 nozzle

4 0·75m diameter circular duct, 1 nozzle

5 0·75m square section duct, 1 nozzle

6 1·0m square section duct, 1 nozzle

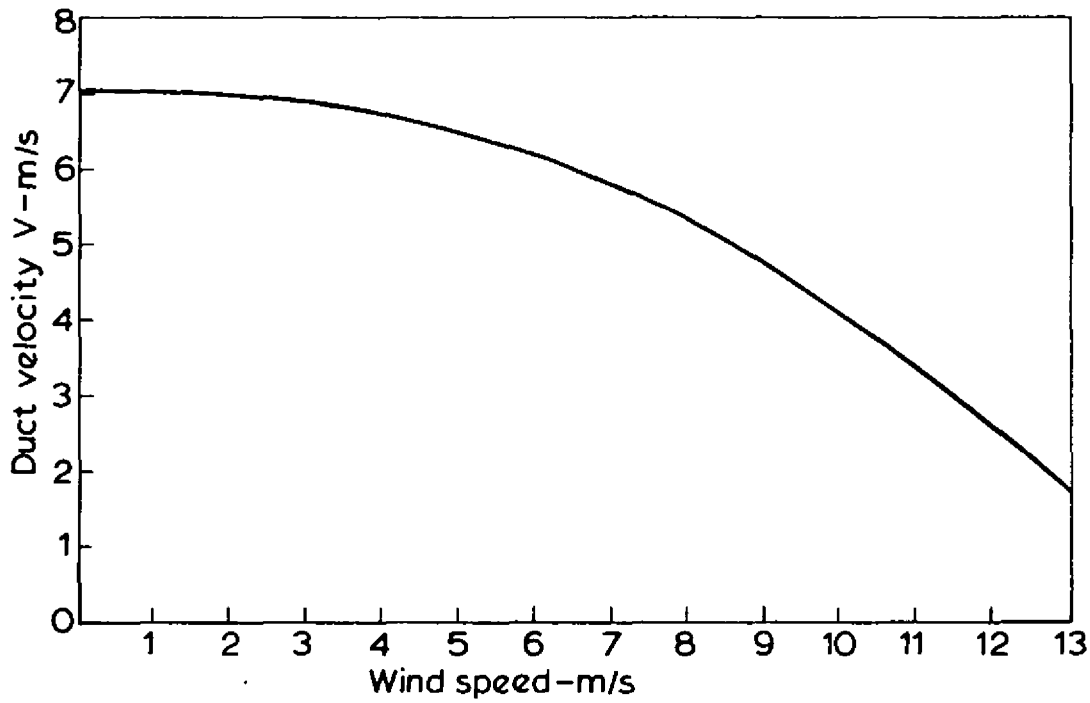
7 1·0m square section duct, 2 nozzles

8 1·5 x 0·75m rectangular section duct, 1 nozzle

9 1·5 x 0·75m rectangular section duct, 2 nozzles

10 1·5 x 0·75m rectangular section duct, 3 nozzles

Figure 7 Design data for various duct configurations (for 20m duct resistance including inlet and outlet velocity head losses)



The effect of wind on the outlet of a 0·75m² duct is shown
The wind is blowing along the axis of the duct exit and there is no deflecting cowl. Water pressure 1400 kN/m²

Figure 8 The effect of external wind on extraction velocity

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