AN EVALUATION OF ROOM AIR
DIFFUSION PERFORMANCE

by

AN-PING HOU

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Major Professor

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CHAPTER I

INTRODUCTION

The distribution of air in an occupied space is a very important, but
complex, problem. Major contributions to the understanding of air diffusion
in occupied spaces for different type outlets have been made through the
long-range research program in the Department of Mechanical Engineering at
Kansas State University.

Extensive air diffusion studies were conducted at Kansas State
University by Miller and Nevins, and they found the average room air velocity·
to be a linear function of diffuser outlet velocity. This function is not
universally usable since the slope of the correlation curve is a function
of the type of outlet device.

The objective of this report is to use the data collected in earlier
research to correlate the average room air velocity with the momentum flow
of the air at the outlet. If this possible, it will give the engineer an
additional and very valuable tool to use in the design of room air diffusion
systems.

"Speed" and "velocity", although technically different, are both
interpreted as speed in this report, as is customary in the industry.

CHAPTER II

LITERATURE SURVEY

A comprehensive study of air diffusion was conducted at Kansas State University by Miller and Nevins. Five air diffusion systems: circular cone-type ceiling diffusers, high side wall grilles, sill grilles, ceiling slots and light troffer diffusers, were evaluated to determine the factors which effect air diffusion from outlets and formulate a means for predicting the performance of these systems.

An analysis of the extensive testing results was made based on the Air Diffusion Performance Index (ADPI). This index is defined as the number of measuring positions in which the combination of air speed and temperature are within specified comfort limits. The comfort limits were defined as an "Effective Draft Temperature" between -1.67 and +1.11 ^{o}C, and a local air speed less than 0.36 m/sec. The Effective Draft Temperature (θ) is a function of local air speed (V_L) and difference in air temperature between the local point (T_L) and a control temperature (T_C), viz.

$\theta = (T_L - T_C) - 7.66(V_L - 0.15)$, in degree C. Fig. 1 indicates the relationship between θ and the local velocity and local temperature differences. Conditions inside the boundaries $-1.67 \leqslant \theta \leqslant 1.11$ and $0 \leqslant V \leqslant 0.36$ were considered confortable.

Data on velocity and temperature were taken throughout the test room, under various conditions of load and flow rate, and the number of comfortable locations, expressed as a percentage of the total number of measuring positions, is the ADPI. Thus an ADPI of 100% indicates that all the occupied zone could be considered comfortable, whereas an ADPI of 50% shows that

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$$\theta = (T_x - T_c) - 7.66(V - 0.15)$$
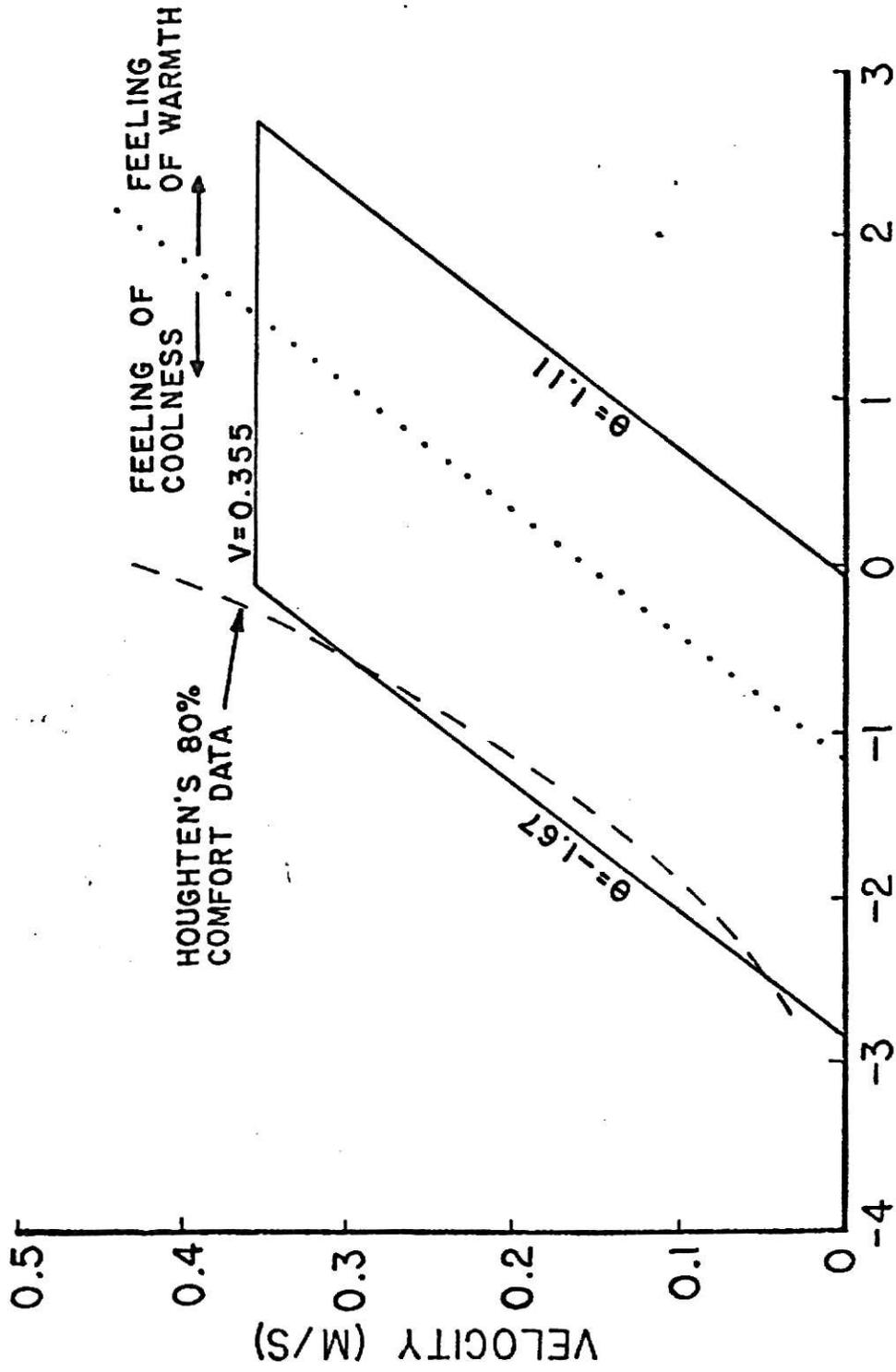
FIG. 1 CRITERIA FOR THE AIR DIFFUSION PERFORMANCE INDEX (ADPI)

conditions were satisfactory at only half of the measuring positions.

Attempts to correlate ADPI with average room velocity were unsuccessful, thereby supporting the conclusion that ADPI and occupant comfort are not functions of velocity or average velocity alone.

If it is possible to predict average room air velocity, it will give the engineer an additional and very valuable tool to use in the design of room air diffusion systems.

The contributions of Jackman are also significant and made more data available to the profession. He was the first to attempt to correlate outlet momentum flow and average room air velocity. His reports also show how these parameters could be used in a design procedure.

As previously stated, Miller and Nevins found, for all diffusers, the average room velocity was a linear function of the outlet velocity, as shown in Fig. 2.

Unfortunately it is also obvious that the type of the device employed has a very definite effect on the slope of the line. Quite a number of correlations have been attempted to remove the effect of the device type as a parameter, but all have been unsuccessful.

HIGH SIDE WALL & SILL GRILLES
SLOT DIFFUSERS
TROFFER DIFFUSERS
0.15 M VER. CONE DIF.
0.15 M HOR CONE DIF.
0.30 M HOR. INT. &
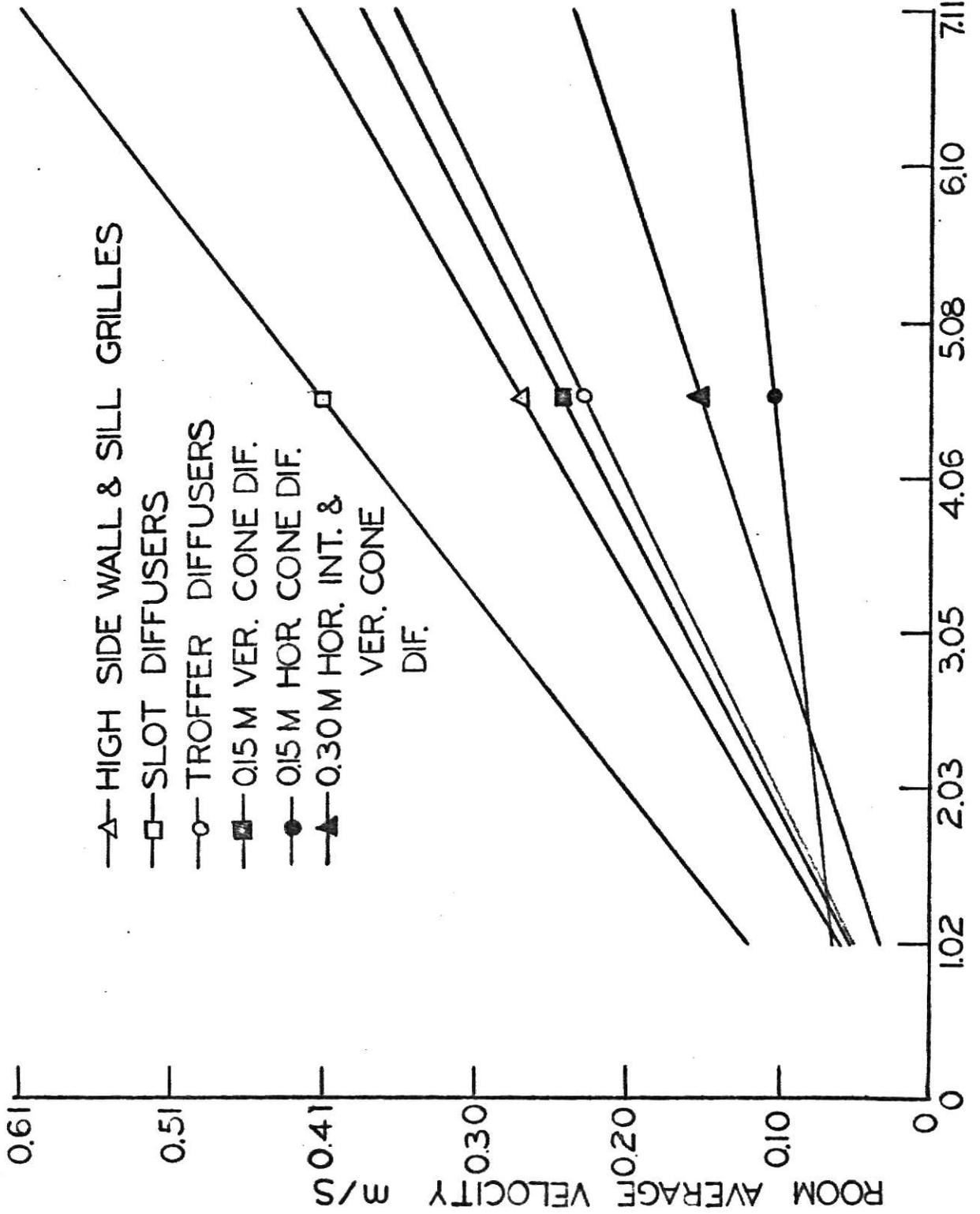VER. CONE
DIF.

OUTLET VELOCITY m/S

FIG. 2 ROOM AVERAGE VELOCITY VS. OUTLET VELOCITY

ROOM AVERAGE VELOCITY m/S

CHAPTER III

TEST FACILITIES

To study and compare various air diffusion systems, two sets of experimental data were available. The test facilities, with necessary instrumentation, were developed by Jackman and by Miller and may be described as follows:

Jackman's Facilities:

The test room was constructed to represent a typical office located in an office block and having one external wall. The dimensions were 4.9 x 3.7 x 2.75 m, the last dimension being the ceiling height.

The air supply system was terminated by a length of flexible ducting to accommodate variations in the position of supply opening into the room. A return air aperture was permanently located just above floor level in the center of the end wall.

Two electrically heated carpets completely covering the floor area were used to provide a uniform heat load within the room.

An inclined manometer was used with a venturi-meter for the measurement of the supply air flow.

Air temperatures and air speeds within the room were measured by six thermocouples and eleven heated thermocouple anemometers mounted on a supporting trolly. These sensors were supported on a traversing framework which enabled manual adjustment of the sensor position throughout the room. The positional adjustment was made by means of a pulley system operated from outside the test room. A total of forty-nine positions, seven transverses and seven longitudiuals, were used for each test, as shown in Fig. 3
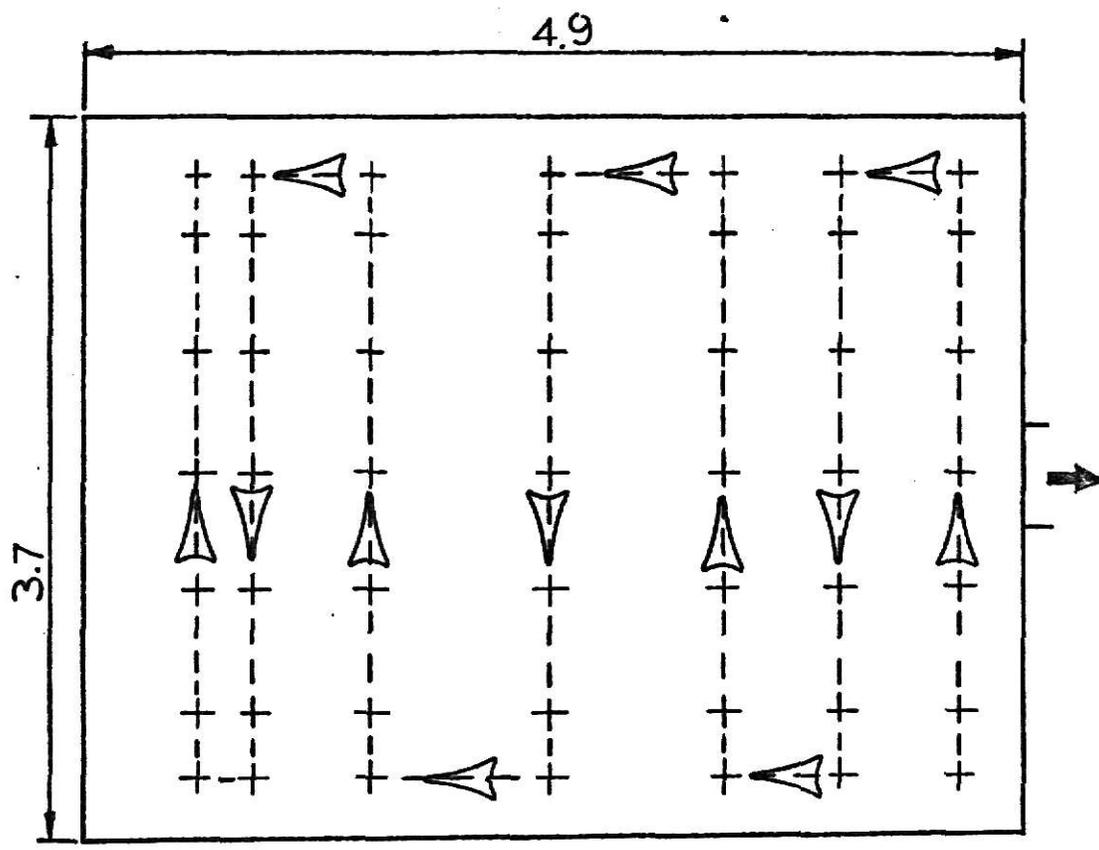
FIG.3 PLAN VIEW OF TEST ROOM
INSTRUMENT TRAVERSE (JACKMAN)

Four types of diffusers were used during Jackman's experimental program: high side wall grille, sill grille, circular ceiling diffuser and linear ceiling slot.

The tests of high side wall grilles were conducted using 150 x 150 mm, 305 x 305 mm, 610 x 150 mm and 1070 x 75 mm supply aperatures, which were located in the center of the wall of the test room, directly above the return air grille, with their horizontal center line 305 mm below the ceiling.

For the first series tests of sill grilles, a 1070 x 75 mm linear grille centrally located at the top of the plenum was used, as shown in Fig. 4a. This was followed by tests with two 1070 x 65 mm linear grilles located as shown in Fig. 4b. In some of the tests the linear grilles were reduced in length by blanking two of the spaces between the blades. The supply air flow rates ranged from 0.071 to 0.280 m^3/sec for both high side wall grille and sill grille tests.

Three types of circular ceiling diffusers were included:

(1) a simple 300 mm diameter plate mounted below a 150 mm diameter opening

(2) a proprietary 150 mm (neck diameter) plaque diffuser

(3) a proprietary 150 mm (neck diameter) multi-cone diffuser

Each diffuser was centrally mounted in the ceiling. The supply air flow rates ranged from 0.045 to 0.165 m^3/sec.

A linear two-slot diffuser equal in length to the width of the test room was mounted flush with the ceiling. A series of tests was conducted as follows:

(1) diffuser centrally located, horizontal discharge in opposite directions, as shown in Fig. 5a. Supply air flow rates ranged from 0.140 to 0.235 m^3/sec.

(2) diffuser located 150 mm from wall, single slot with horizontal discharge

across ceiling, as shown in Fig. 5b. Supply air flow rates ranged from 0.095 to 0.235 m³/sec.

(3) diffuser located 150 mm from wall, two slots with horizontal discharge across ceiling, as shown in Fig. 5b. Supply air flow rate was 0.190 m³/sec.

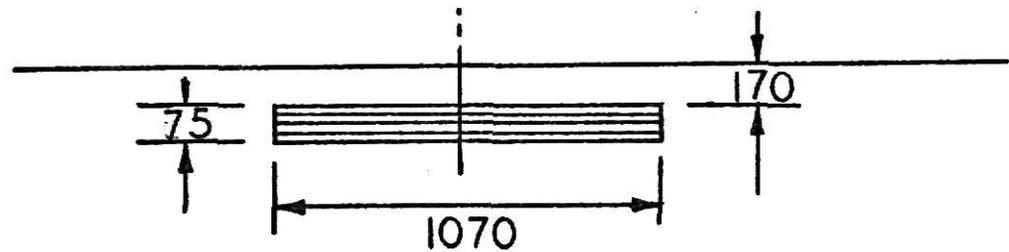The detailed test conditions of Jackman's experimental program are shown in Table 1.

FIG. 4 A
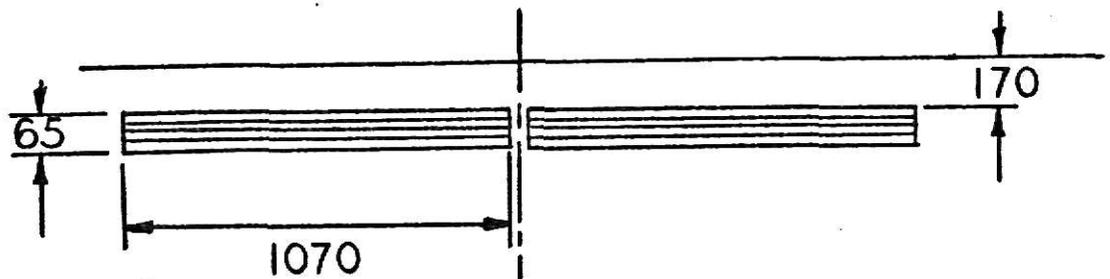
FIG. 4 B

FIG. 4 SILL GRILL SIZE & LOCATION
ALL DIMENSIONS ARE mm

FIG. 5 A

.150 m
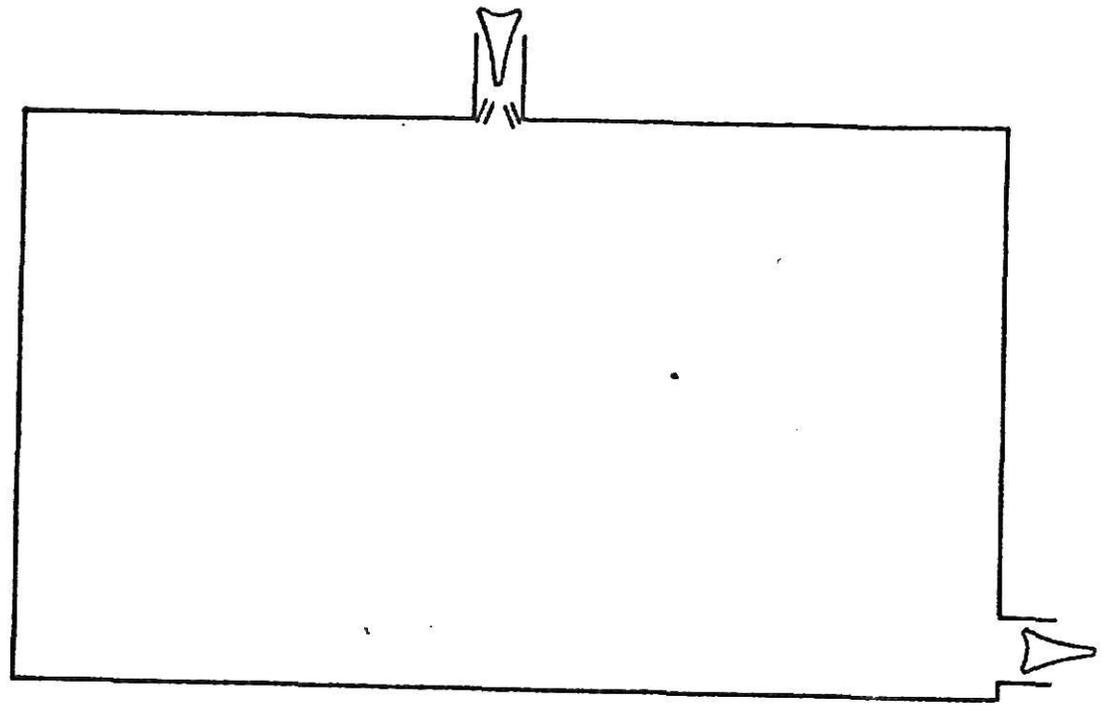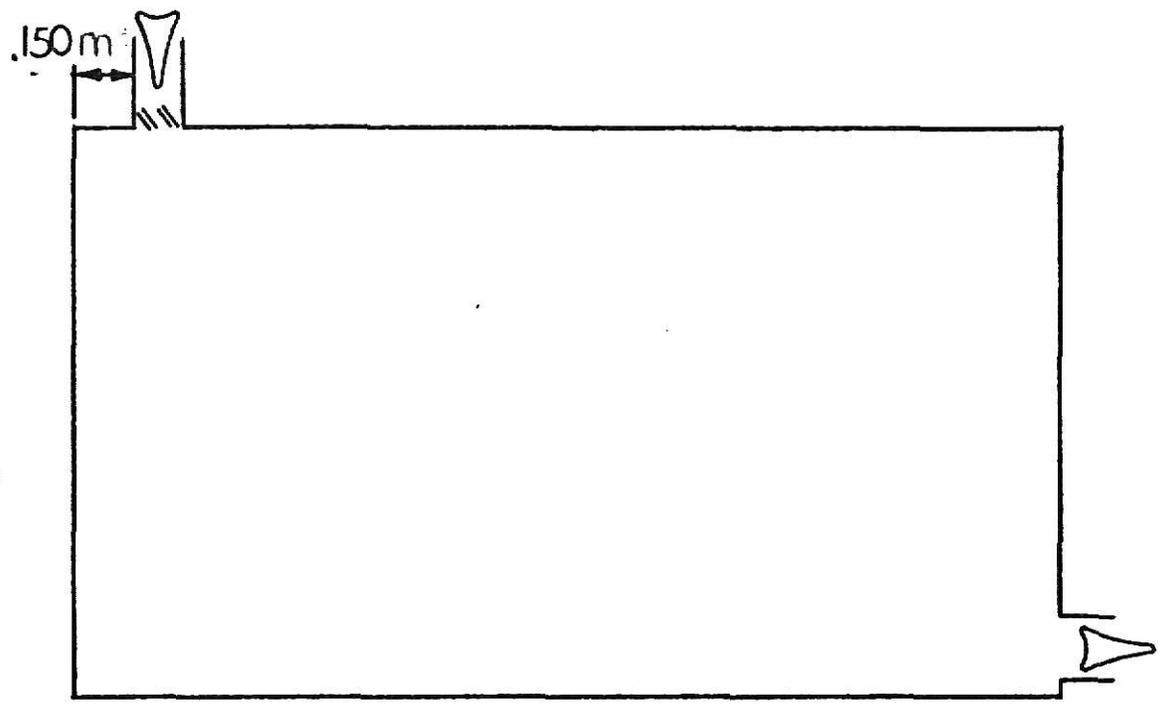
FIG. 5 B

FIG. 5 LINEAR CEILING SLOT
LOCATION

Table 1. Outline of Test Conditions (Jackman)

Devices Tested	Size mm	Supply Air Flow Rate Range m³/sec
High.Side Wall Grille	150 x 150 305 x 305 610 x 150 1070 x 75	from 0.071 to 0.280
Sill Grille	one 1070 x 75 linear grille centrally located (Fig. 4a) two 1070 x 65 linear grilles (Fig. 4b)	from 0.071 to 0.280
Circular Ceiling Diffuser	a 300 diameter plate mounted below a 150 diameter opening 150 plaque diffuser 150 multi-cone diffuser	from 0.045 to 0.160
Linear Slot Diffuser	a linear two-slot diffuser equal in length to the width of the test room	from 0.140 to 0.235
	single slot with horizontal discharge across ceiling	from 0.095 to 0.235
	two slots with horizontal discharge across ceiling (Fig. 5a)	0.190
	diffuser centrally located horizontal discharge in opposite directions (Fig. 5b)	from 0.140 to 0.235

Miller's Facilities:

The test room dimensions were 6.10 x 3.66 x 2.74 m and the room was
constructed to simulate an interior room of an intermediate floor of a multi-
story office building.

The interior heating load consisted of lighting and electrical loads
uniformally distributed on the floor to simulate people and equipment.

The return air grille (0.42 x 0.76 m) was located in the center of the
end wall of the test room 0.48 m from the floor, as shown in Fig. 6.

Air temperatures and speeds were measured with twelve heated thermocouple
anemometers and twelve thermocouples on a remotely controlled test rack. There
were two vertical and nine horizontal positions of the moveable test rack so
that the velocities and temperatures were obtained at 216 locations within the
room on a 0.61 x 0.61 x 0.61 m grid begining 0.305 m from the wall and 0.102 m
from the floor, as shown in Fig. 6.

Five types of diffusers were used during Miller's experimental program:
high side wall grille, sill grille, light troffer diffuser, ceiling slot
diffuser and circular ceiling diffuser.

The high side wall grille was located in the center of the wall of the
test room, directly above the return air grille, with its horizontal center
line 152 mm below the ceiling. Two sizes of grilles, 610 x 152 mm and
406 x 152 mm, with vanes straight were used.

The position of the sill grille is shown in Fig. 7, the grille was
located 965 mm above the floor facing upward. Three sizes, 610 x 76 mm,
813 x 102 mm and 1220 x 152 mm with vanes straight were used, the longer
vanes were on the room side of the grille and parallel to the wall. Tests were
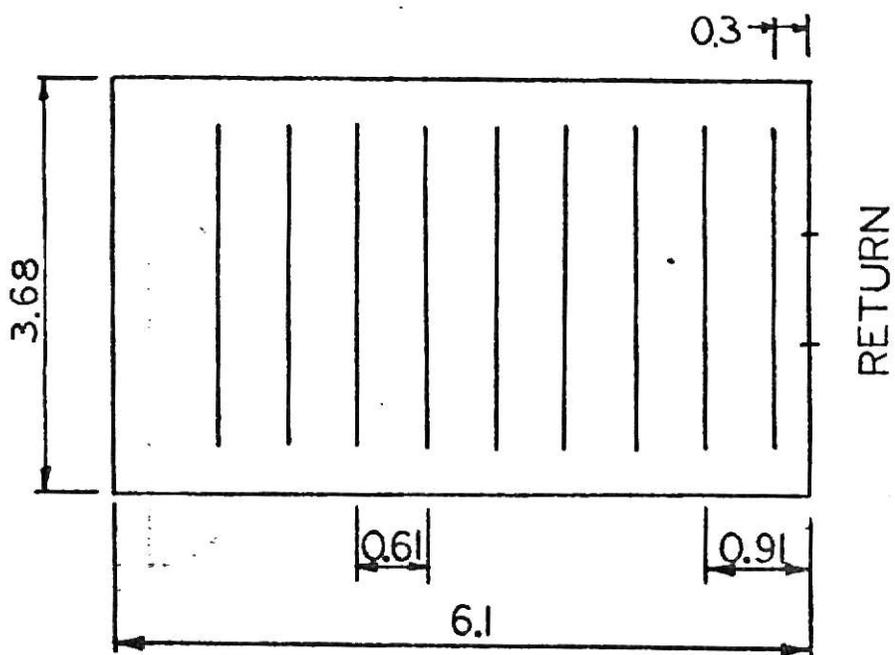
PLAN VIEW
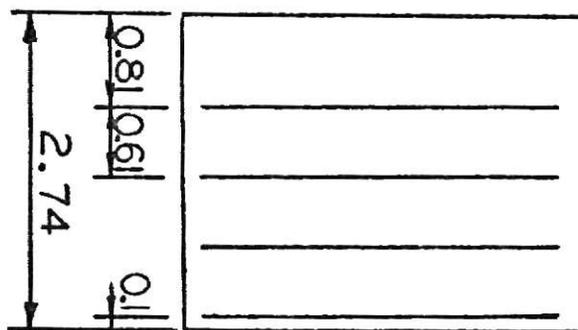
END VIEW

FIG. 6 TEST ROOM AND INSTRUMENT LAYOUT

14
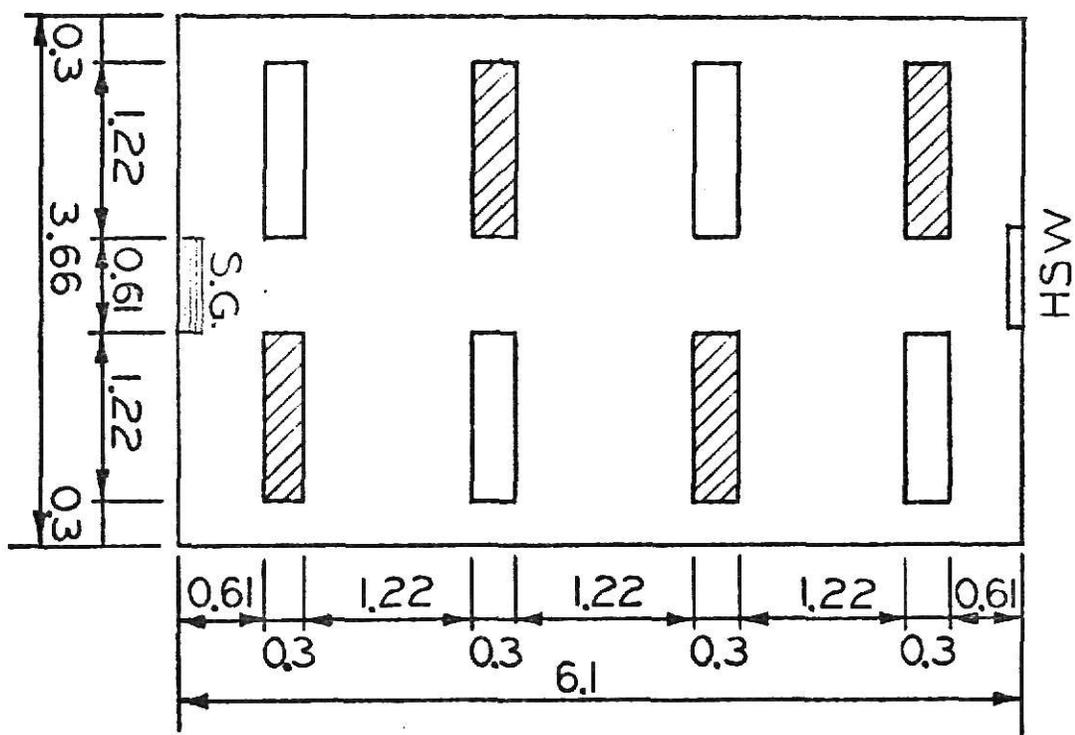
FIG 7 LIGHT TROFFER & SILL GRILLE
DIFFUSER LOCATION
SHADED INDICATE FOUR ACTIVE

ALL DIMENSIONS ARE IN METERS

also conducted with the 610 x 152 mm and 1220 x 152 mm aperture fitted with double- deflection grilles (the vanes set for a deflection of 45° into the room and a spread in both directions of 22.5^o).

The positions of the light troffer diffusers are also shown in Fig. 7. For part of the tests all eight diffusers were active and the flow was adjusted so that each diffuser projected one-eight of the total amount of air into the room. The air diffusion pattern was adjustable and for all tests the air was projected horizontally into the room. For part of the tests only four diffusers were active. These tests were also run with each diffuser delivering an equal amount of air to the room and with horizontal projection of the air. When four diffusers were active they were the ones indicated by the shaded area in Fig. 7.

Fig. 8 shows the location of the ceiling slot diffusers. These diffusers were nominally 1220 mm long and 190 mm wide and each had four slots 1190 mm long and 19 mm wide. Three diffusers were used during all tests and were located along the center line of the room with 25 mm between diffusers and 914 mm between the end walls of the room. For some tests only two of the four slots were active. In this case the outer two slots were active with the center two taped shut. The air deflection was horizontal with the slot on the west side projecting air to the west. In the case where four slots were active, the two slots on the west projected the air horizontally to the west.

The supply air flow rates were 0.057, 0.113, 0.227, 0.453 and 0.566 m^3/sec for all tests of high side wall grilles, sill grilles, light troffer diffusers and ceiling slot diffusers.

The circular ceiling diffusers tested were of typical manufacture and had adjustable cones for changing the air diffusion pattern within the test

1.73
0.19
3.66

0.3 | 1.22 | 0.91

6.1
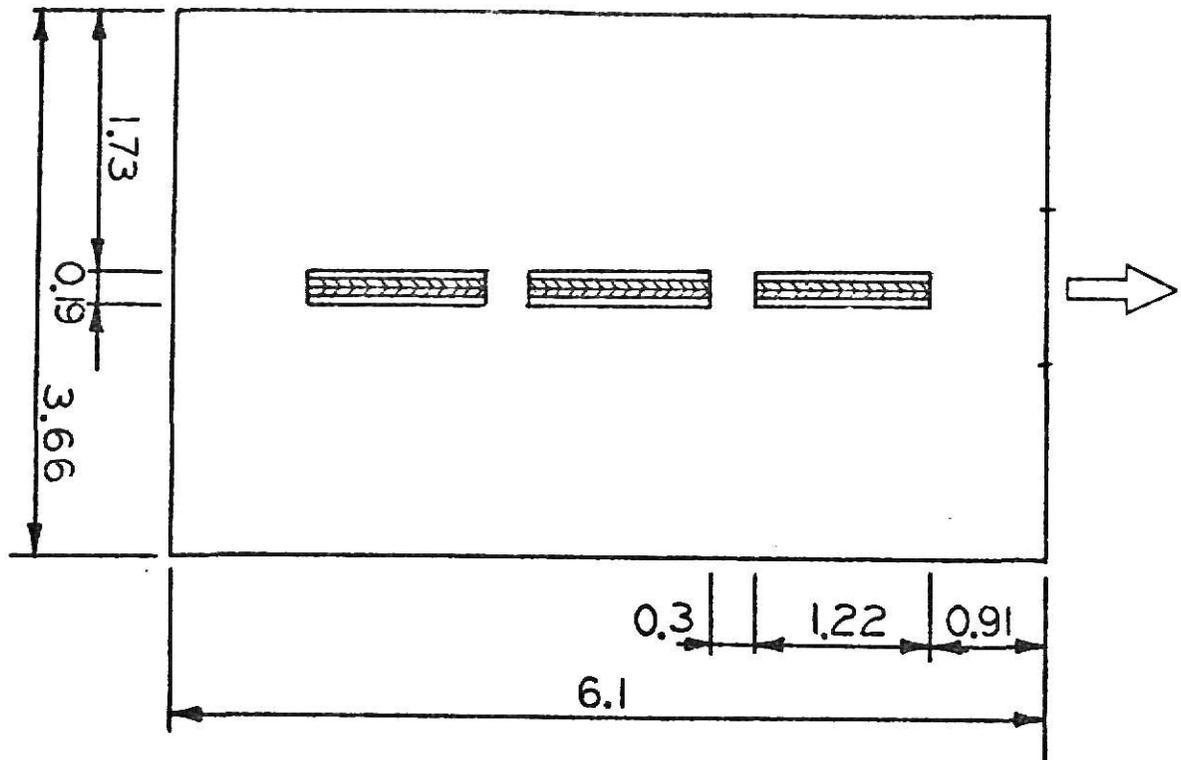
FIG. 8 CEILING SLOT LOCATION

1.52   1.52

1.83

HSW

1.83
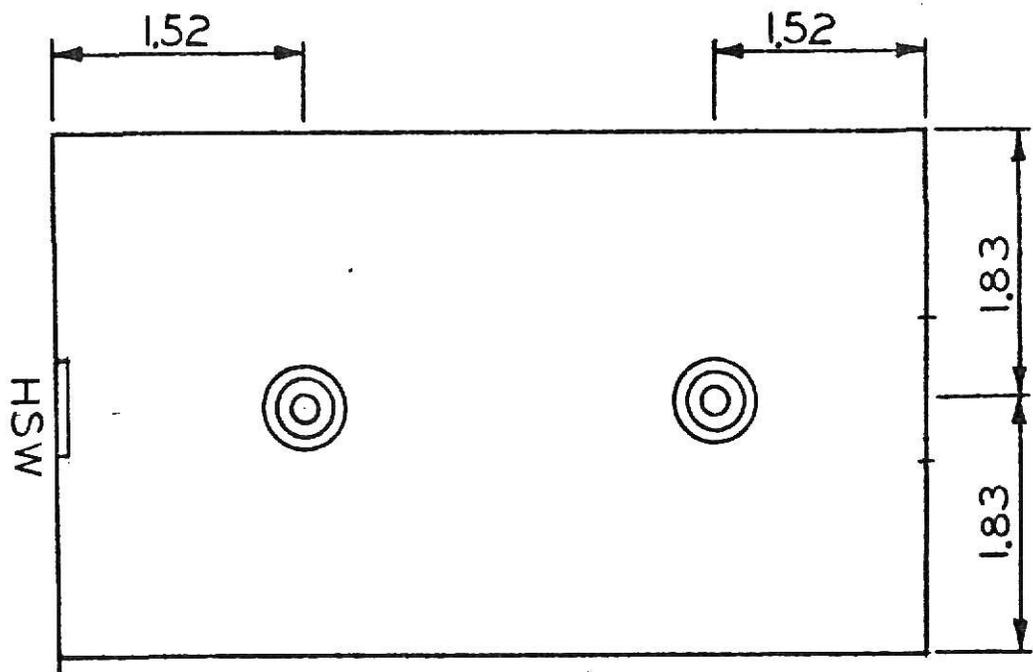
FIG. 9 CIRCULAR CEILING & HIGH SIDE WALL
DIFFUSER LOCATION

room. There were two nominal sizes of diffusers tested, 152 and 305 mm diameter, with a pair of diffusers of the same size used in all tests. The ceiling diffusers were located on the center line of the long dimension (6.10 m) of the test room, with 3.05 m between the diffuser center lines and 1.52 m between each diffuser center line and an end wall as shown in Fig. 9. The supply air flow rates were 0.057, 0.113, 0.566 and 1.133 m³/sec.

The details of the test conditions of Miller's experimental program are shown in Table 2.

Table 2. Outline of Test Conditions

Devices Tested	Size mm	Supply Air Flow Rates m^3/sec
High Side Wall Grille	610 x 152 406 x 152	0.057, 0.113, 0.227, 0.453 and 0.566
Sill Grille	1220 x 152 813 x 102 610 x 76 1220 x 152 vanes 45° and 22.5^o 610 x 152 vanes 45° and 22.5^o	0.057, 0.113, 0.227, 0.453 and 0.566
Light Troffer Diffuser	eight 1220 x 305 troffers: four active eight active	0.057, 0.113, 0.227, 0.453 and 0.566
Ceiling Slot Diffuser	three 1220 x 190 slot diffusers, each has four 1190 x 19 slots two open per slot four open per slot	0.057, 0.113, 0.227, 0.453 and 0.566
Circular Ceiling diffuser	two 305 nominal sizes two 152 nominal sizes	0.057, 0.113, 0.566 and 1.133

CHAPTER IV

ANALYSIS OF DATA

Relation Between Average Room Air Velocity and Supply Air Momentum:

Jackman's Data:

To assess the proportion of high and low air velocities generated in the
occupied zone of the room, an analysis of the number of air velocity readings
in selected velocity ranges was made by Jackman for each test condition.
Fig. 10 shows the results of high side wall grille tests. On the graph, each
test condition is represented by its specific value of momentum flow, in
$kg_{\gamma}m/sec^2$, and at each value of momentum three points are plotted, the lowest
point represents the percentage number of velocity readings that were below
0.10 m/sec, the second point represents the percentage that were below
0.25 m/sec and upper point represents the percentage that were below 0.50 m/sec.
Curves were drawn through the resulting sets of points. The vertical distances
between these lines represent the percentage number of velocity readings that
were within the velocity ranges indicated on the figure.

Fig. 11 shows plots of the mean air velocity in the occupied zone against
the momentum of supply air. The mean air velocity values used in the
calculations are only those velocities in the range 0.10 to 0.50 m/sec, so
they would not be equal to the true mean value, although the error would be
least when the number of velocity readings outside the range was minimal.

Jackman assumed that 60% of the velocity readings were below the mean air
velocity and 40% were above the mean air velocity throughout the range of
supply air momentum. Then the momentum at which the mean air velocity was 0.10
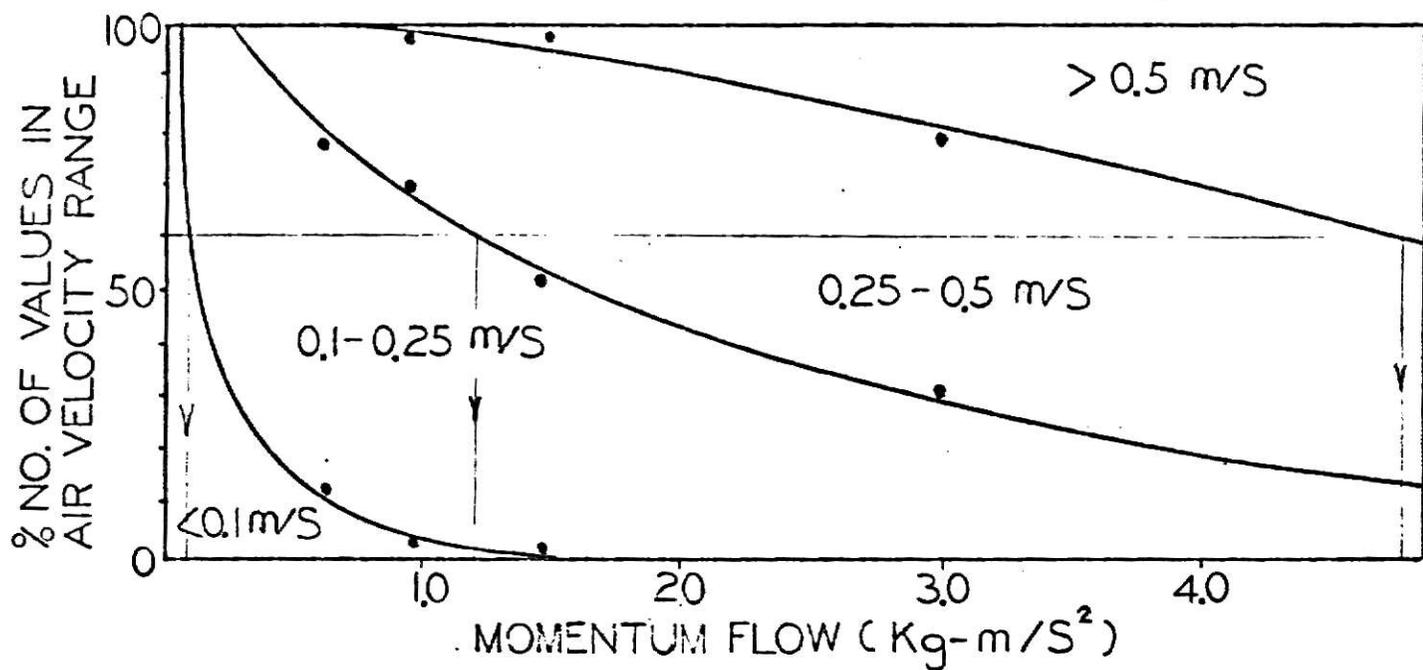0.25 and 0.50 m/sec was found from the intersections of three curves on Fig.10
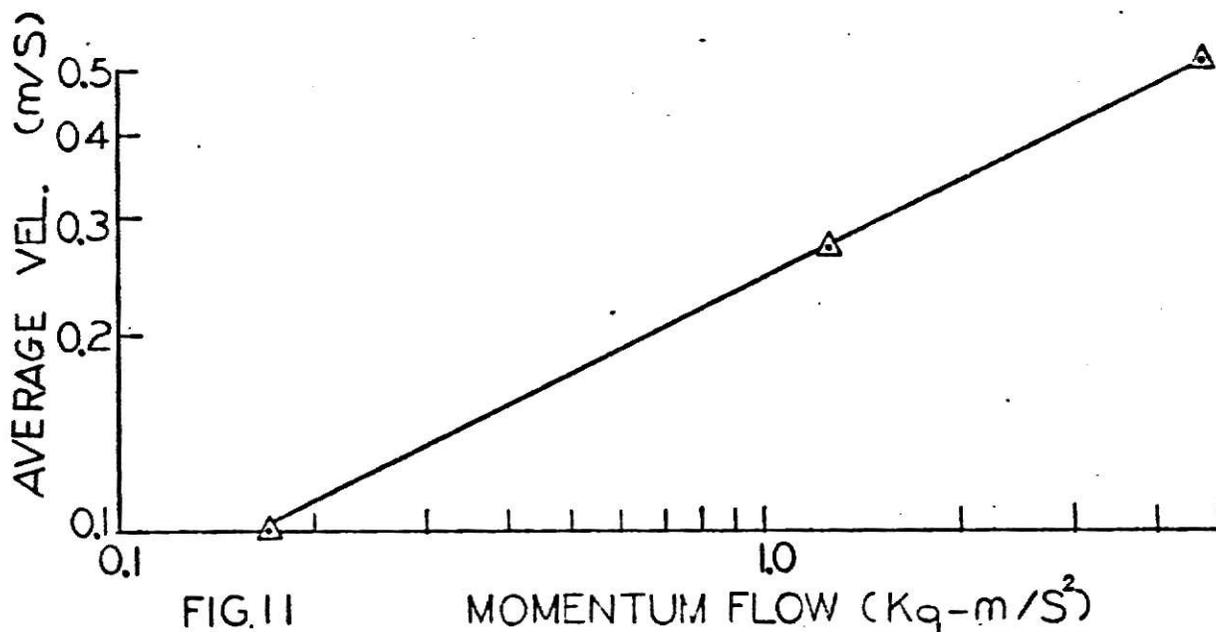
FIG 10 DISTRIBUTION OF AIR VELOCITY

FIG.11   MOMENTUM FLOW (Kg-m/S²)

FIG 11 AVERAGE VELOCITY VS. SUPPLY
AIR MOMENTUM

with 60% level, i.e. 0.19 1.20 and 4.80 kg-m/sec respectively. He plotted
these values of mean air velocity against the supply air momentum on
logarithmic graph axes, as shown on Fig. 11 and found they were on a line.
The equation of this line is $\overline{V} = 0.23 \, M^{0.50}$. The derived equations for all
types of diffusers tested by Jackman are listed in Table 3.

Table 3. Jackman's Derived Equations

Type of Outlet	Equation
High Side Wall Grille	$\overline{V} = 0.23 \, M^{0.50}$
Sill Grille	$\overline{V} = 0.17 \, M^{0.33}$
Circular Ceiling Diffuser	$\overline{V} = 0.21 \, M^{0.50}$
Linear Ceiling Diffuser	$\overline{V} = 0.42 \, M^{0.50}$

Jackman's Data - A More Complete Analysis:

In Jackman's papers, his figures show the experimental data obtained
for mean room air velocity and momentum of supply air. His data for sill
grilles are shown in Fig. 12. The data points from this figure were used to
obtain a least squares correlation between the supply air momentum and mean
air velocity. The equation of which was found to be $\overline{V} = 0.19 \, M^{0.22}$, shown as
the broken line in Fig. 12. The equations obtained by this method from
Jackman's data, for all types of diffusers are tabulated in Table 4.
Jackman's equation for the sill grille given in Table 3, and derived by his
previously described method, is shown as a solid line in Fig. 12.
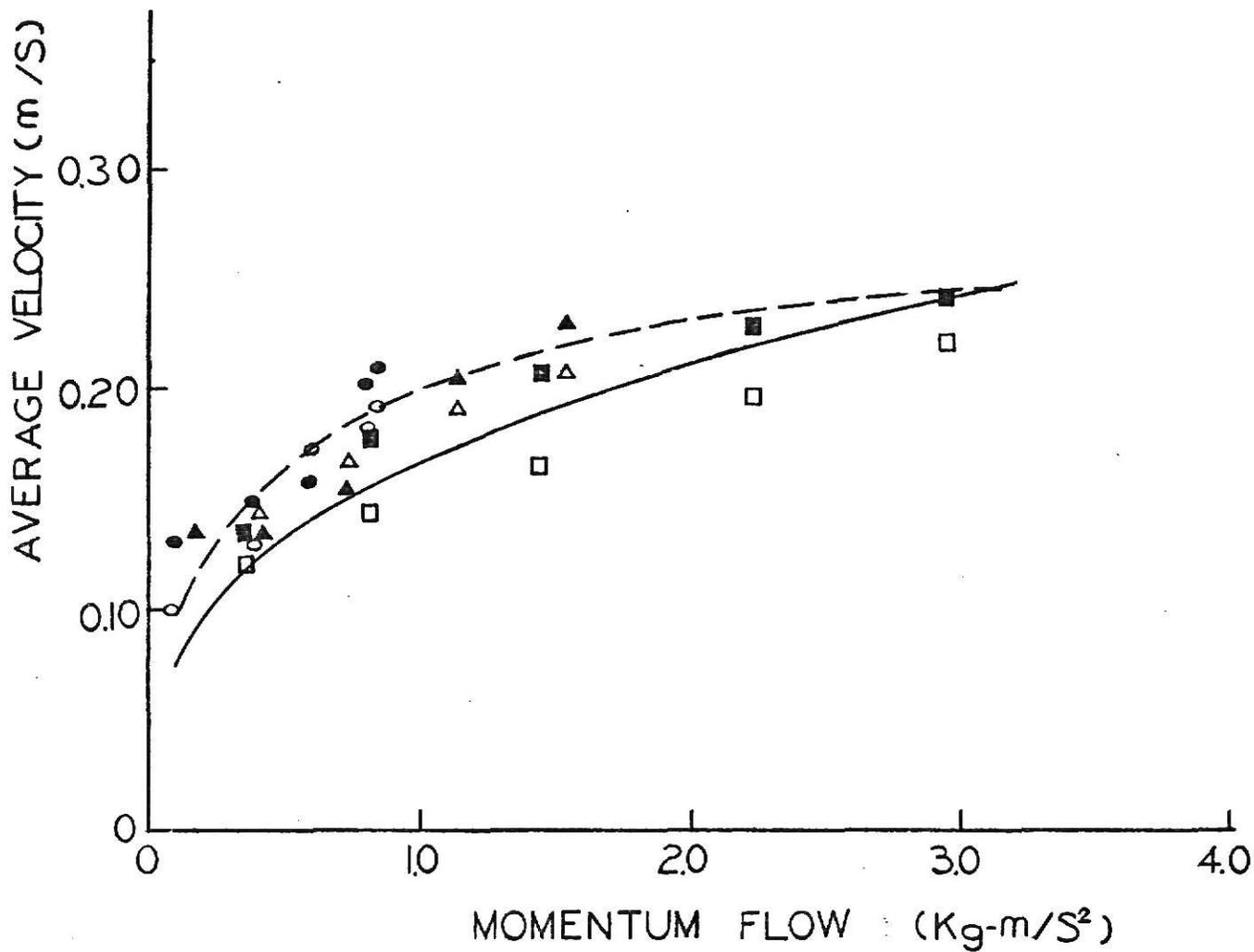
FIG.12 AVERAGE VELOCITY VS. SUPPLY

AIR MOMENTUM FOR SILL GRILLES

Table 4. Equations Obtained from Jackman's Data

Type of Outlet	Data Fit Equations by author	Jackman's Equations
High Side Wall Grille	$\overline{V} = 0.23 \ M^{0.22}$	$\overline{V} = 0.23 \ M^{0.50}$
Sill Grille	$\overline{V} = 0.19 \ M^{0.22}$	$\overline{V} = 0.17 \ M^{0.33}$
Circular Ceiling Diffuser	$\overline{V} = 0.26 \ M^{0.42}$	$\overline{V} = 0.21 \ M^{0.50}$
Linear Ceiling Diffuser	$\overline{V} = 0.22 \ M^{0.30}$	$\overline{V} = 0.42 \ M^{0.50}$

Comparison of equations which Jackman derived and those obtained by the author from Jackman's data indicate that the exponent of the supply air momentum of the former equations are greater than the author's equations.

Miller's Data:

Miller collected the 216 velocity and temperature points from each test run for each supply air flow rate. A computer program was written for calculating the numerical average velocity for the entire room.

Outlet momentum flow is defined as:

$$M = \rho Q V_o \quad \dots\dots\dots\dots\dots\dots\dots (1)$$

where:

M = flow of momentum, $kg\text{-}m/sec^2$

ρ = density of the supply air, kg/m^3

Q = supply air flow rate, m^3/sec

V_o = outlet velocity, m/sec

The outlet velocity was not actually measured, but was calculated from

the measured air flow rate divided by the area factor A_k. (Each kind of diffuser has an area factor A_k provided by the manufacturer.) By definition:

$$A_k = \frac{Q}{V_o} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Solving for V_o and subtituting into Eq. (1) results in

$$M = \frac{\rho Q^2}{A_k} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Now since ρ and Q of the supply air are known, the momentum of supply air is computed by Eq. (3). For multiple identical diffusers with the same air flow rate, air density and area factor A_k, it may be shown that

$$M = \frac{\rho Q^2}{A_k N} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where N = number of diffusers, and Q = total flow rate.

The least squares correlation between the average air velocity and supply air momentum of each different type diffuser was then found. The five equations are tabulated in Table 5 and shown as the solid line on Figs. 13-17.

Table 5. Miller's Equations

Type of Outlet	Equation	Correlation Coefficient
High Side Wall Grille	$\overline{V} = 0.23 \, M^{0.32}$	$r = 0.94$
Sill Grille	$\overline{V} = 0.19 \, M^{0.46}$	$r = 0.95$
Circular Ceiling Diffuser	$\overline{V} = 0.21 \, M^{0.51}$	$r = 0.94$
Ceiling Slot Diffuser	$\overline{V} = 0.25 \, M^{0.35}$	$r = 0.87$
Light Troffer Diffuser	$\overline{V} = 0.17 \, M^{0.30}$	$r = 0.67$
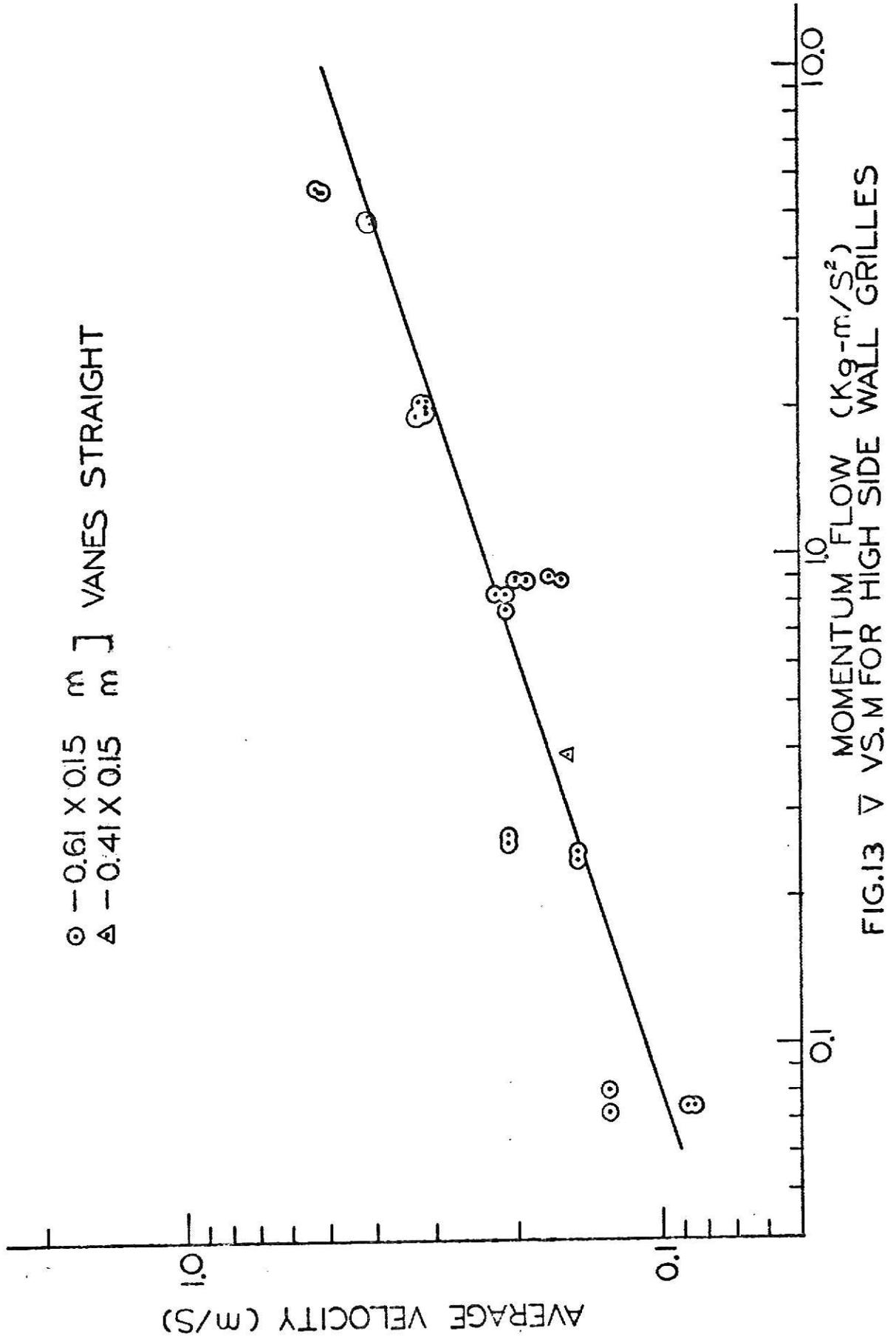
FIG.I3 ▽ VS. M FOR HIGH SIDE WALL GRILLES
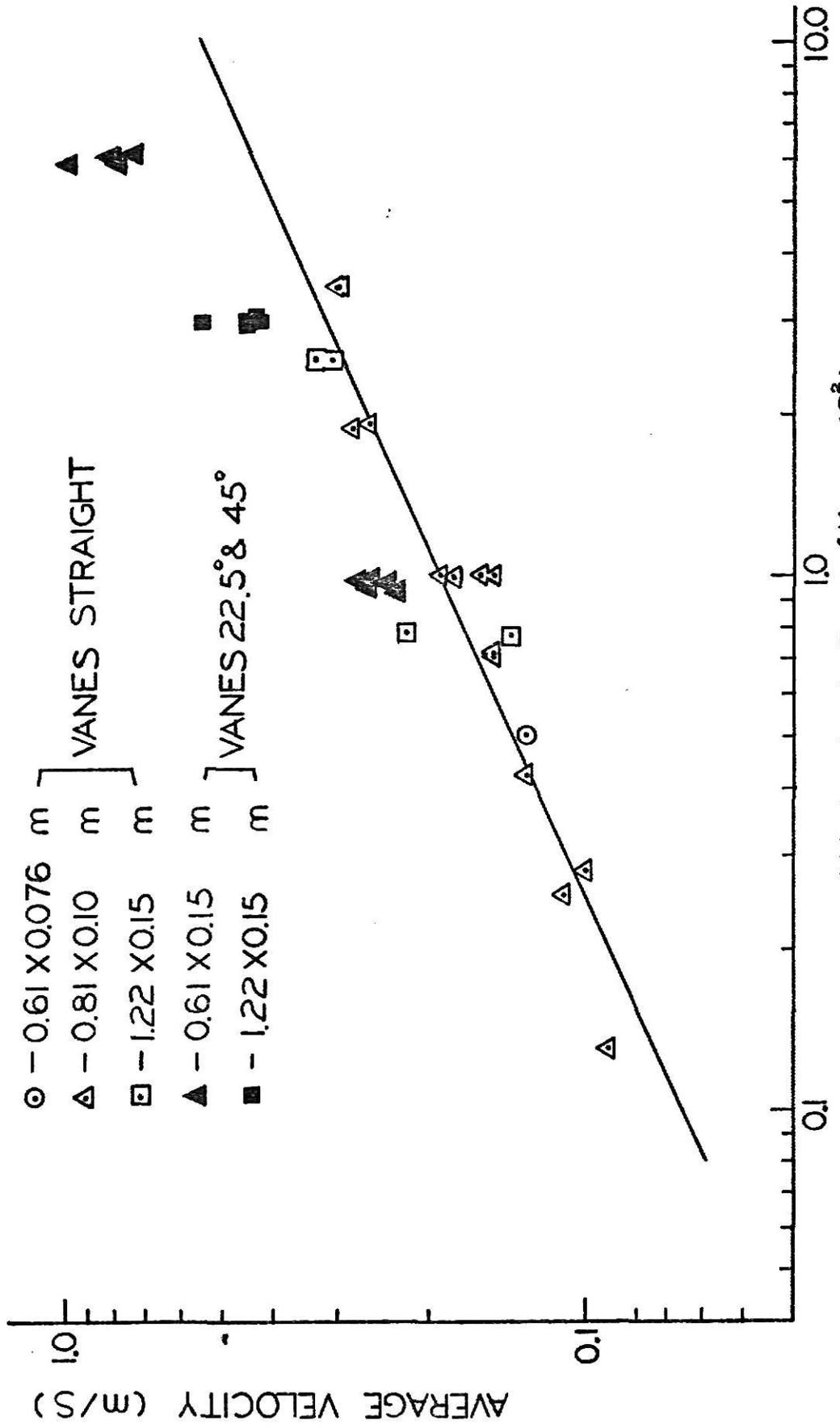
FIG.14 $\bar{V}$ VS. M FOR SILL GRILLES

MOMENTUM FLOM(Kg-m/S^2)

AVERAGE VELOCITY (m/S)

⊙ — 0.61 X 0.076 m
△ — 0.81 X 0.10 m   ] VANES STRAIGHT
⊡ — 1.22 X 0.15 m

▲ — 0.61 X 0.15 m   ] VANES 22.5° & 45°
■ — 1.22 X 0.15 m

⊙ – 0.15 m   DIAMETER ⎤ HORIZONTAL, INTERMEDIATE
△ – 0.31 m   DIAMETER ⎦ & VERTICAL AIR FLOW

MOMENTUM FLOW (Kg-m/S²)
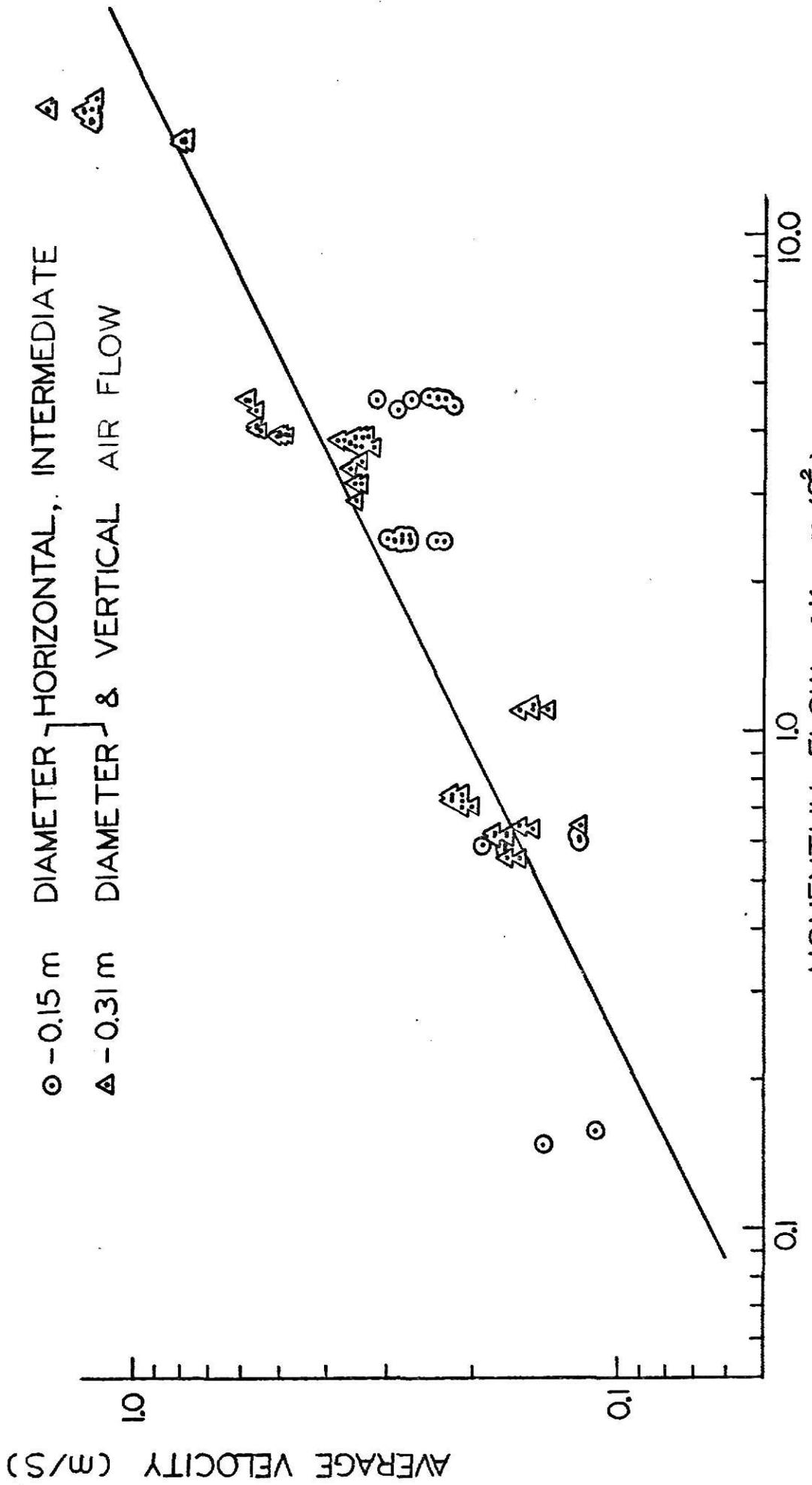
FIG. 15   V̄ VS. M FOR CIRCULAR CONE TYPE DIFFUSERS

AVERAGE VELOCITY (m/S)

FIG.16 $\overline{V}$ VS. M FOR SLOT DIFFUSERS

MOMENTUM FLOW (Kg-m/S^2)

AVERAGE VELOCITY (m/S)

⊙ — 2 SLOTS OPEN
△ — 4 SLOTS OPEN

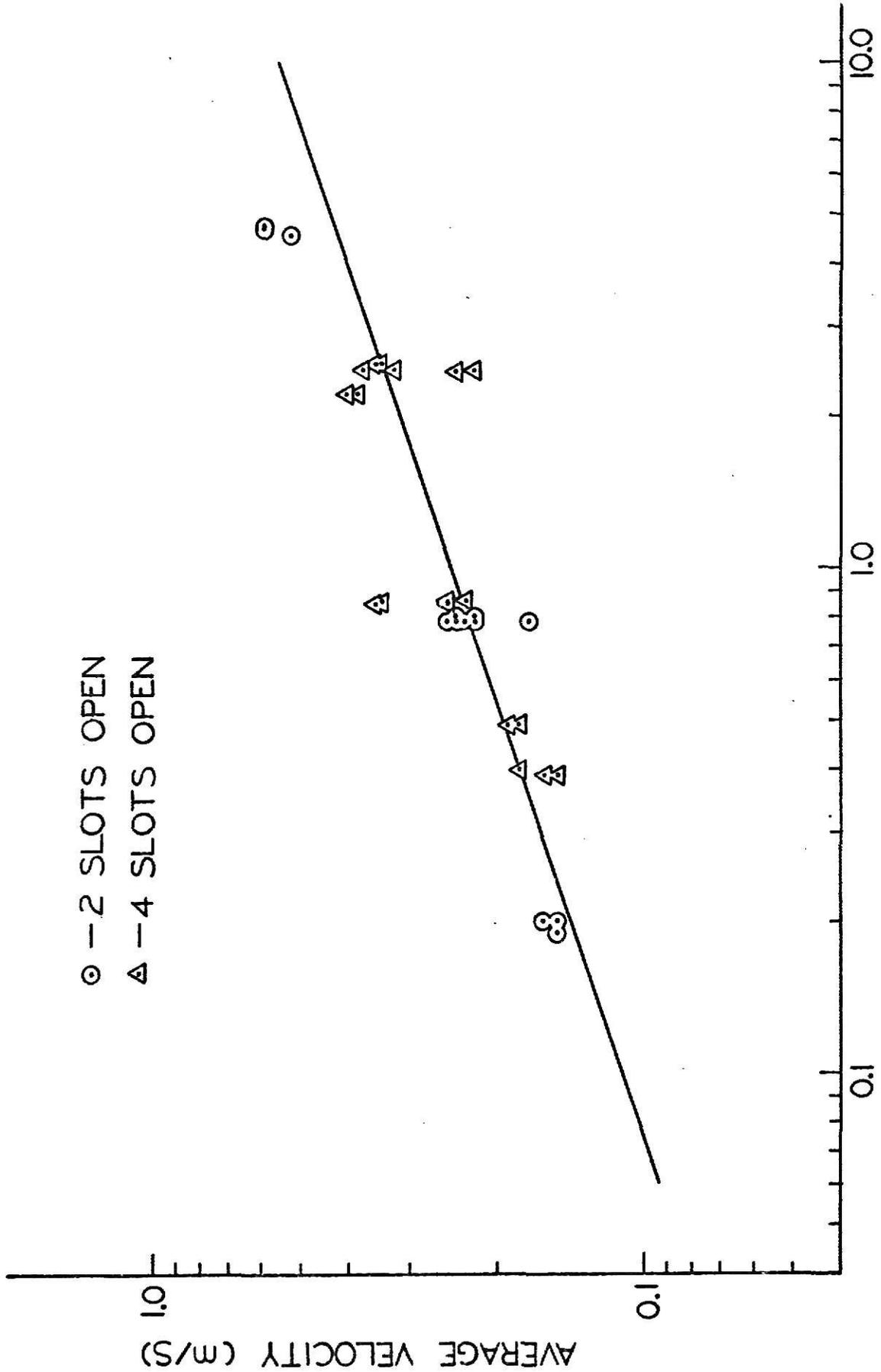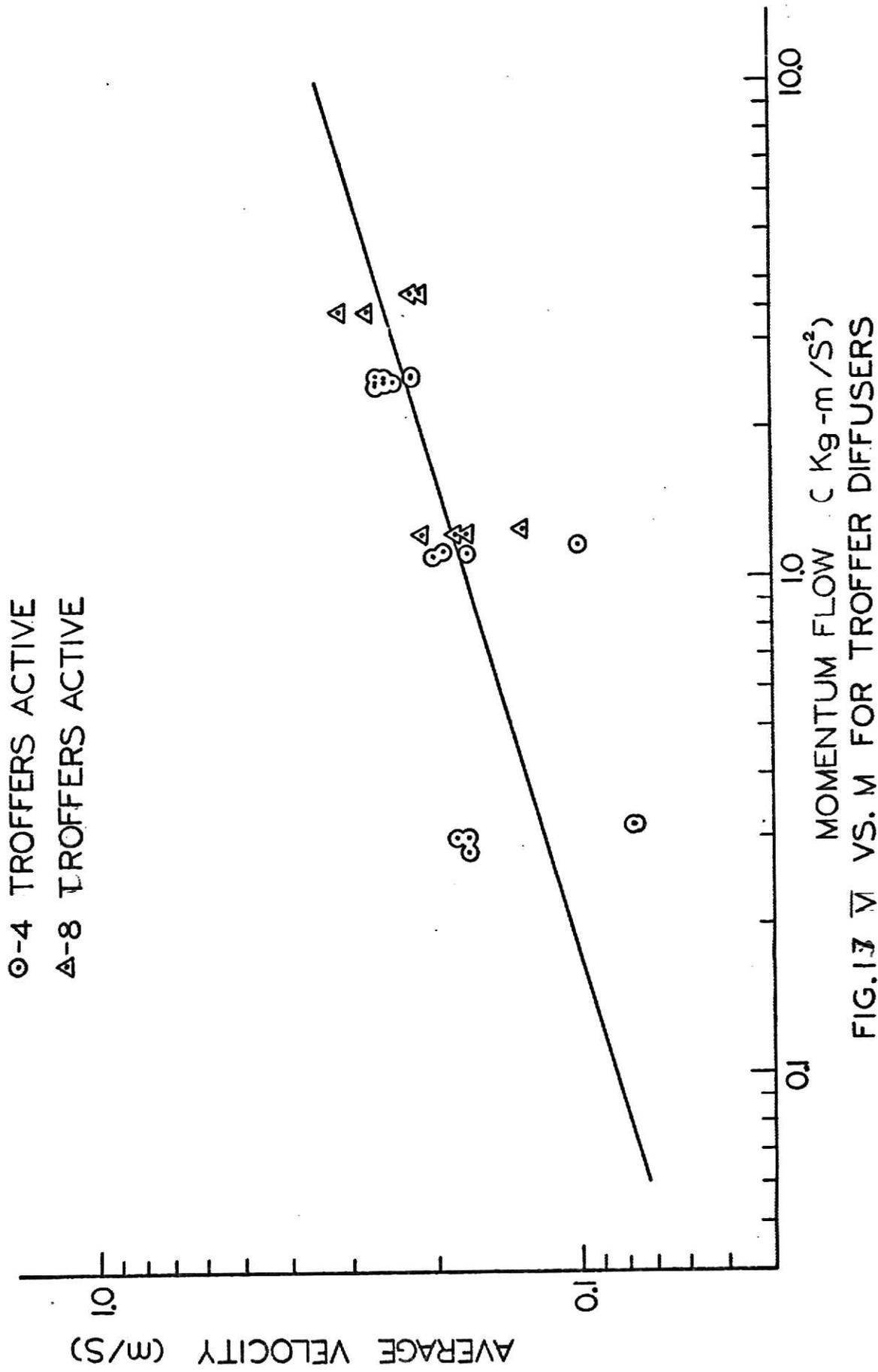FIG.13 $\bar{V}$ VS. M FOR TROFFER DIFFUSERS

Usefulness:

Combining all data and applying the least square analysis, a generalized correlation between average air velocity and supply air momentum may be obtained. The generalized equation is

$$\overline{V} = 0.22 \ M^{0.44} \quad \dots\dots\dots\dots\dots\dots(5)$$

and is plotted with the original data in Fig. 18.

This equation, having a 0.90 correlation coefficient or "goodness of fit" to the experimental data, is thus a good representation of the data. It may resonably be assumed that the average air velocity is a function of the supply air momentum, and not type of outlet device. For more accurate representation, one should use the equations given in Table 5, which apply to specific devices, and thus have higher correlation coefficients.

The broken lines in Fig. 18 shows the 95% confidence interval. At a given value of supply air momentum, there is 95% chance that the estimated average air velocity will be within this interval. As an example, assume the supply air momentum is 1.0 kg-m/sec, then the Eq. (5) above indicates the average air velocity will be 0.22 m/sec, and one may say, with 95% confidence, that the average velocity will be between 0.13 and 0.38 m/sec.

The equation of momentum flow is

$$M = mV_o = \rho Q V_o \quad \dots\dots\dots\dots\dots\dots(6)$$

and also the flow rate may be written as

$$Q = AV_o \quad \dots\dots\dots\dots\dots\dots\dots(7)$$

Substituting Eq. (7) into Eq. (6) results in

$$M = \rho A V_o^2 \quad \dots\dots\dots\dots\dots\dots\dots(8)$$

or

$$V_o = \left(\frac{M}{\rho A} \right)^{1/2} \quad \dots\dots\dots\dots\dots\dots(9)$$
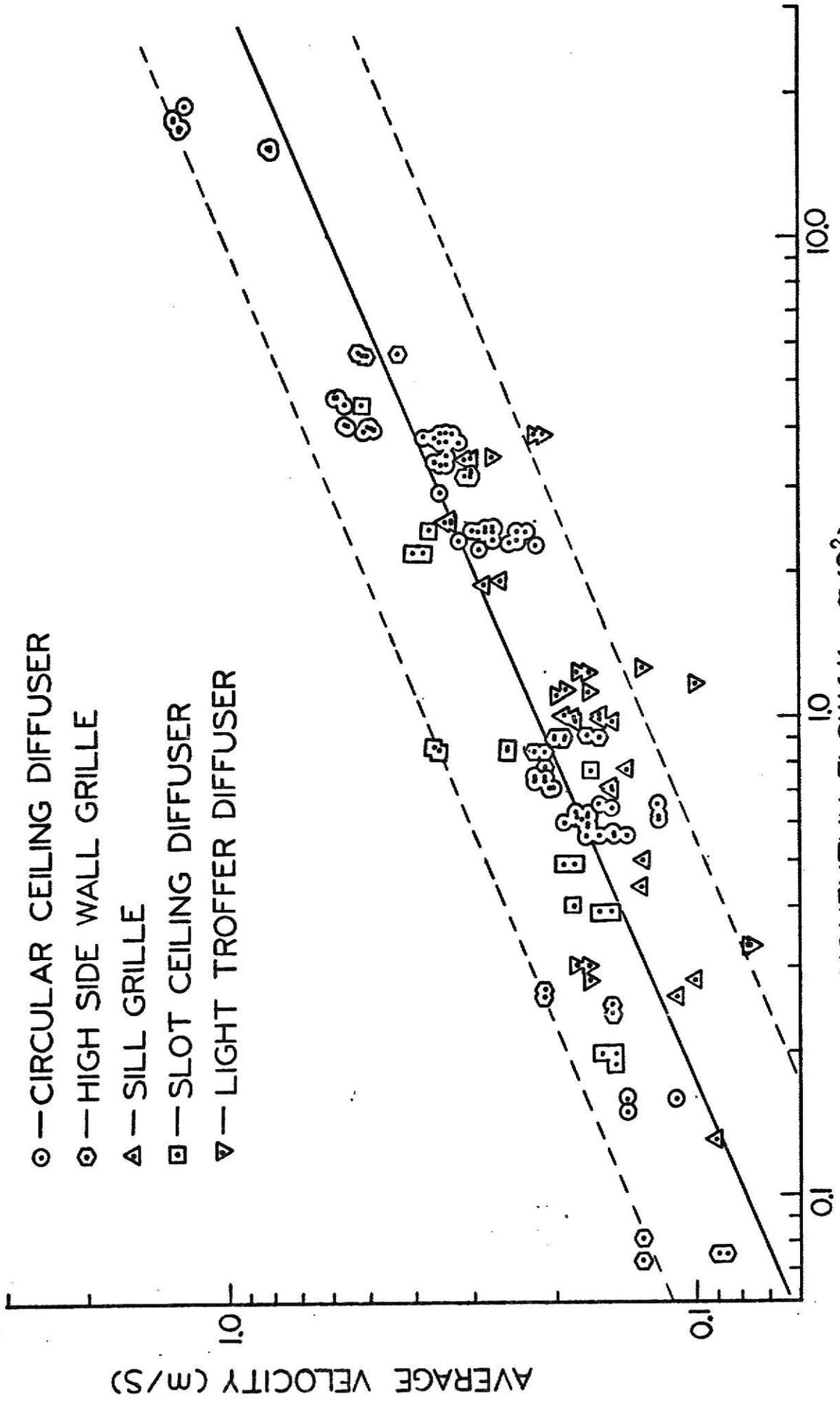
FIG. 18 AVERAGE ROOM AIR VELOCITY VS. SUPPLY AIR MOMENTUM

⊙ — CIRCULAR CEILING DIFFUSER
⊚ — HIGH SIDE WALL GRILLE
△ — SILL GRILLE
▣ — SLOT CEILING DIFFUSER
▽ — LIGHT TROFFER DIFFUSER

MOMENTUM FLOW (Kg-m/S²)

AVERAGE VELOCITY (m/S)

One might therefore assume that the room average velocity is proportional to the square root of the inlet momentum. To determine if this is a reasonable assumption, a statistical check on whether there is a significant difference between the exponents 0.50 and 0.44 (Eq.9 and Eq.5) was made. This was done by means of the " t test " applied to $|0.50 - 0.44|$.

Unfortunately the calculated " t " was greater than 95% level of significance, and it is therefore concluded that there is a significant different between $\overline{V} = k\ M^{0.44}$ and $\overline{V} = k\ M^{0.50}$.

Room Size Effect:

Jackman attempted to take the room dimensions into consideration as follows:

For circular ceiling diffusers, Jackman assumed that in the square room the air flow passed through diagonal rectangular cross sections of area $2L(\dfrac{L^2}{4} + H^2)^{0.50}$ as shown in Fig. 19.

Based on the derived equation (Table 4) $\overline{V} = 0.21\ M^{0.50}$, Jackman multiplied both sides of the equation by the area $2L(\dfrac{L^2}{4} + H^2)^{0.50}$, and substituted the length and height of the test room (where L = 4.9 m, H = 2.75 m), and obtained the following equation.

$$\overline{V} = 0.21\ M^{0.50}$$

$$= 0.21[\ 2L(\frac{L^2}{4} + H^2)^{0.50}]\ \frac{M^{0.50}}{2L(\frac{L^2}{4} + H^2)^{0.50}}$$

$$= 0.21[\ \frac{L^2}{4} + H^2]^{0.50}[\ \frac{M}{\frac{L^2}{4} + H^2}\ ]^{0.50}$$

$$\overline{V} = 0.765[\ \frac{M}{\frac{L^2}{4} + H^2}\ ]^{0.50}$$
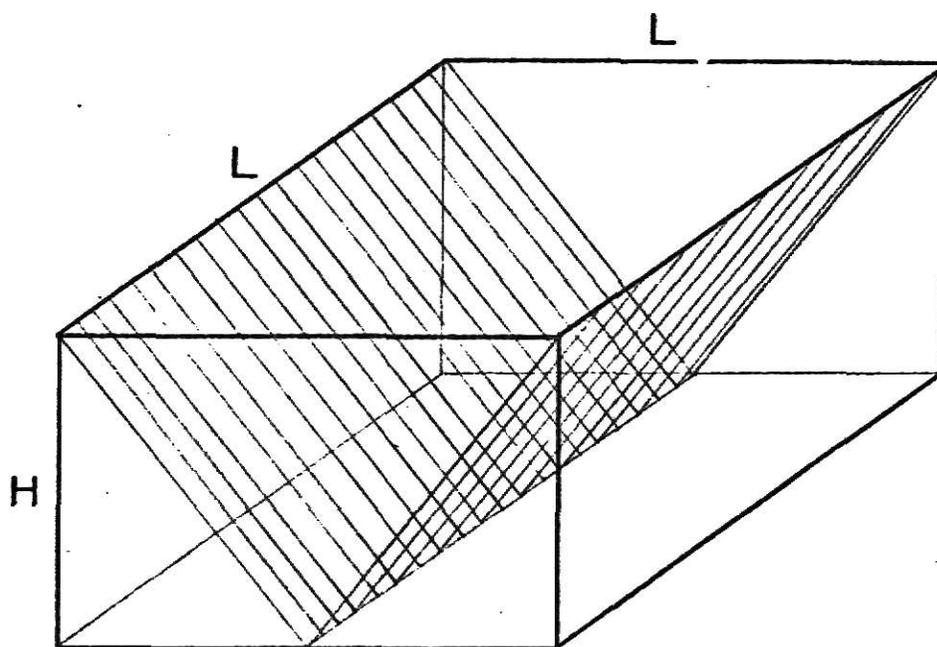
FIG. 19 ROOM SIZE EFFECT ON
THE SINGLE CIRCULAR
CEILING DIFFUSER

For high side wall grille, Jackman assumed the air flow passed through the area equivalent to $[\ B \times H\ ]^{0.50}$, as shown in Fig. 20.

By the same procedure:

$$\overline{V} = 0.23\ M^{0.50}$$

$$= 0.23[\ B \times H\ ]^{0.50}[\ \frac{M}{B \times H}\ ]^{0.50}$$
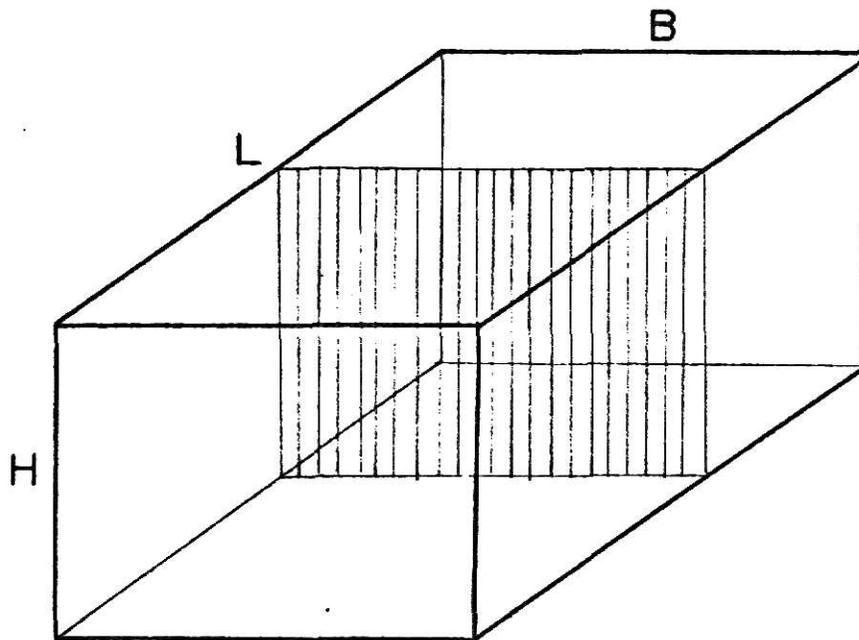
$$= 0.73[\ \frac{M}{B \times H}\ ]^{0.50}$$

FIG. 20 ROOM SIZE EFFECT ON
THE HIGH SIDE WALL GRILLE

For sill grille, he assumed the air flow passed through the area equivalent to $[\text{ B x H }]^{0.33}$. The apparent reason for choosing power of 0.33 was to match with the power of derived equation:

$$\overline{V} = 0.17 \, M^{0.33}$$

$$= 0.17[\text{ B x H }]^{0.33}[\frac{M}{B \times H}]^{0.33}$$

$$\overline{V} = 0.37[\frac{M}{B \times H}]^{0.33}$$

For linear ceiling diffusers, he assumed the air passed through the diagonal cross sectional area $B[L^2 + H^2]^{0.50}$, as shown in Fig. 21, and obtained the equation $\quad \overline{V} = 0.42 \, M^{0.50}$

$$= 0.42[B(L^2 + H^2)^{0.50}][\frac{1}{B}(\frac{M}{L^2 + H^2})^{0.50}]$$

$$\overline{V} = 1.08[\frac{M}{L^2 + H^2}]^{0.50}$$
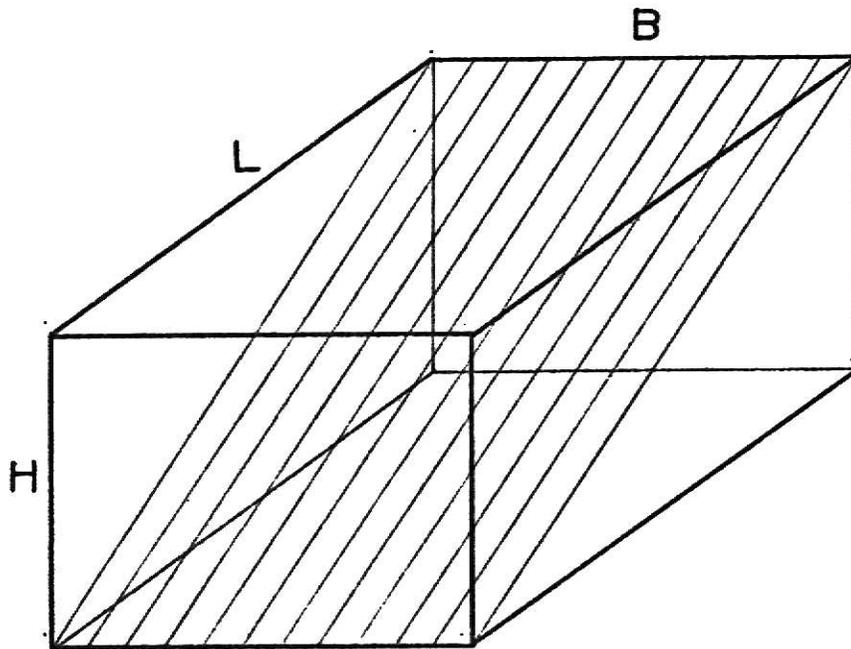
FIG. 21 ROOM EFFECT ON THE LINEAR CEILING DIFFUSER

Those derived equations related the different room shapes and sizes. It was assumed that the mean room air speed was proportional to the total supply air momentum devided by a representative room cross sectional area, and this cross sectional area was viewed as that through which the total flow passed as it recirculated.

Although above equations would seem to provide acceptable information, tests on different room shapes and sizes have not been run to prove the analysis.

Now both the equations of Miller and Jackman are tabulated in Table 6.

Table 6. Equations from Jackman's and Miller's Data

Type of Outlet	Jackman's Data			Derived from Miller's Data
	Jackman's Equations	Room Size Effect	Derived from Jackman's Data	
High Side Wall Grille	$\overline{V} = 0.23\ M^{0.50}$	$\overline{V} = 0.73(\dfrac{M}{BH})^{0.50}$	$\overline{V} = 0.23\ M^{0.22}$	$\overline{V} = 0.23\ M^{0.32}$
Sill Grille	$\overline{V} = 0.17\ M^{0.33}$	$\overline{V} = 0.37(\dfrac{M}{BH})^{0.33}$	$\overline{V} = 0.22\ M^{0.22}$	$\overline{V} = 0.19\ M^{0.46}$
Circular Ceiling Diff.	$\overline{V} = 0.21\ M^{0.50}$	$\overline{V} = 0.765(\dfrac{M}{\frac{L^2}{4}+H^2})^{0.50}$	$\overline{V} = 0.26\ M^{0.42}$	$\overline{V} = 0.21\ M^{0.51}$
Linear Ceiling Diff.	$\overline{V} = 0.42\ M^{0.50}$	$\overline{V} = 1.08(\dfrac{M}{L^2+H^2})^{0.50}$	$\overline{V} = 0.22\ M^{0.30}$	$\overline{V} = 0.25\ M^{0.35}$
Light Troffer Diff.				$\overline{V} = 0.17\ M^{0.30}$

CHAPTER V

CONCLUSION

Since the real test of success for an air diffusion system is the evaluation of the temperatures and velocities in the occupied space, the total room air motion is of major importance.

Previous attempts have been made to relate the performance of an air diffusion system to the design parameters of outlet velocity, flow rate, outlet type and placement and load on the space. However, each research group has selected a different procedure for evaluating their systems culminating in different results.

This report uses data from two research projects to arrive at a more general correlation between the average room air velocity and supply air momentum. The equation relating the two is $\overline{V} = 0.22 \ M^{0.44}$. Neither the temperature variations of the room nor the supply air affect the equation significantly. If the diffuser type is known, a more specific equation for the type of diffuser, from Table 5, can be used.

Experimental data from Jackman and from Miller indicate that the average room air velocity is probably dependent on room size as well as supply air momentum. Unfortunately, the test rooms of Jackman and Miller are so similar in size (there is only one meter difference in test room length), that it is not possible to eliminate the room size effect from the correlation. Futher experimental investigation relating to the effect of varying room dimensions on supply air momentum and average room air velocity is planned by Miller at Kansas State University.

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AN EVALUATION OF ROOM AIR
DIFFUSION PERFORMANCE

by

AN-PING HOU

B.S.; Tatung Institute of Technology, Taipei, Taiwan, 1971

AN ABSTRACT OF A MASTER'S REPORT

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Manhattan, Kansas

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ABSTRACT.

The distribution of air in an occupied space is a very important, but complex, problem. A correlation between the average room air velocity and the momentum flow of the air at outlet was found, and it gives the engineer an additional and very valuable tool to use in the design of room air diffusion systems. However, experimental data indicate the average room air velocity is probably dependent on room size as well as supply air momentum. Futher experimental investigation is planned by Miller and hopefully more data will be made available to the profession.