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MANUAL

for the

BALANCING and ADJUSTMENT

of

AIR DISTRIBUTION SYSTEMS

AIA File No. 30-D-4

SHEET METAL AND AIR CONDITIONING CONTRACTORS
NATIONAL ASSOCIATION, INC.

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BALANCING and ADJUSTMENT

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AIR DISTRIBUTION SYSTEMS

First Edition — 1967

SHEET METAL AND AIR CONDITIONING CONTRACTORS
NATIONAL ASSOCIATION, INC.

P.O. BOX 3506 WASHINGTON, D.C. 20007

"A National Association of Ventilating and Air Conditioning, Warm Air Heating, Industrial Air Handling, Architectural Sheet Metal, Roofing and Fabricating Contractors"

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Sheet Metal and Air Conditioning Contractors'
National Association, Inc.
P.O. Box 3506 Washington, D.C. 20007

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PREFACE

This Manual has been carefully compiled by members of the sheet metal and air conditioning industry. It is a part of an entire system of manuals designed to establish proper standards for the construction of air distribution systems.

Over the years, the problem of balancing and adjusting a system has frequently been neglected by both designers and installers. It is the intention of this Manual to indicate the scope of balancing and adjusting for which the contractor is responsible; a procedure to follow; methods of reporting; descriptions of instruments that may be used, and some outlines of basic knowledge necessary to the art.

The procedural and technical sections of this Manual are not intended to be a complete text book. The procedural section offers an accepted method of balancing, in outline. The technical section offers a brief review of material found in standard texts and is included as a refresher for those using the manual. An air balancing technician, using the methods and principles described in this Manual, can properly balance any system.

The committee hopes that designing engineers will use the sample specification included in this Manual. By doing so, they will aid the contractor and the local SMACNA chapter in providing a service which should be of benefit to the entire industry. The use of qualified personnel, following the recommended procedure, can provide the type of service desired by the designing engineer and the owner.

The committee wishes to acknowledge the cooperation of assigned members of the Air Diffusion Council whose technical knowledge is incorporated in several chapters of this Manual.

SHEET METAL AND AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION, INC.

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GLOSSARY OF SYMBOLS AND DEFINITIONS

AIR INLET –	An opening through which air is removed from a conditioned space (Grilles, Registers, Diffusers and Slots are used as air inlets).
AIR OUTLET –	An opening through which air is supplied into a conditioned space.
DAMPER –	An adjustable device used to vary the volume of air passing through an air outlet, inlet or run of duct.
DIFFUSER –	A circular, square or rectangular air outlet comprised of deflecting members generally located in the ceiling and designed to distribute air in varying directions and planes.
GRILLE –	A louvered or perforated covering over an air opening which can be located in the sidewall, ceiling, floor or duct.
HIGH PRESSURE UNIT –	(Sometimes referred to as Mixing Box, Dual Duct Box or Constant Volume Box). A factory made assembly consisting of either a dual duct modulating damper or a single duct pressure reducing damper and an integral volume control station together with a temperature mixing and sound attenuation system. These units are designed to deliver air when supplied from an external source with static pressures greater than .4" H_2O.
INDUCTION UNIT –	A factory made assembly consisting of a sound attenuator, a manual control damper and one or more nozzles used to develop a high jet velocity of the externally supplied primary air. The high velocity air induces secondary or room air to mix with the primary air providing a high volume of total air to be delivered from the unit. In some types the secondary air is drawn through a coil and either heated or cooled.
PRESSURE REDUCING VALVE –	A factory made assembly consisting of a manual or automatic volume control station operating at static pressures above .4" H_2O.
REGISTER –	A combination of grille and damper assembly covering an opening located in the sidewall, ceiling, floor or duct.
SLOT –	A long, narrow air outlet usually comprised of deflecting members generally located in the ceiling, high sidewall or perimeter sill and designed to distribute air in varying directions and planes. Aspect ratio of length/height is generally greater than 10/1.

CFM	–	Cubic Feet Per Minute: Volume Rate of Air Flow
fpm	–	Feet per Minute: Velocity Rate of Air Flow
TP	–	Total Pressure: inches of water
SP	–	Static Pressure: inches of water
VP	–	Velocity Pressure: inches of water
Pb	–	Barometric Pressure: inches of mercury
rpm	–	Turning speed: Revolutions Per Minute
ATC	–	Automatic Temperature Control
hp	–	Horse Power: Unit of Work
bhp	–	Brake Horse Power: Applied Unit of Work
V	–	Velocity
v	–	volts – unit of electrical pressure
$\sim$	–	(also cy) Electrical Cycles such as 60
ø	–	(also ph) Electrical Phases such as 1ø, 2ø, 3ø
A	–	(also Amps.) Amperes: unit of electrical flow
W	–	Watts: Electrical Work Measure
KW	–	1000 Watts
BTU	–	British Thermal Unit: Quantity of heat necessary to raise 1 lb. of water 1°F
A_k	–	Area Factor of an outlet or inlet measured in square feet
t	–	Throw of an outlet
V_t	–	Terminal velocity of an outlet measured in fpm
V_k	–	Discharge or intake velocity of an outlet or inlet measured in fpm
h	–	Enthalpy or total heat (in BTU per pound)

OCCUPIED ZONE – The area of a conditioned space which extends to within 6" of all room surfaces and up to a height of 6'0".

Vr	–	Room velocity in fpm determined from velocity measurements in the occupied zone.
B_p	–	Sound pressure level in decibels
Lw	–	Sound power level in decibels
DB	–	Dry Bulb temperature, °F
WB	–	Wet Bulb temperature, °F
RH	–	Relative Humidity, %
DP	–	Dew Point temperature, °F
Δ	–	(Delta) Differential

NOTE: For additional psychrometric definitions and symbols see Chapter 12, Page 31

SECTION I

SPECIFICATION

CHAPTER 1
SAMPLE SPECIFICATION, METHOD AND ACCURACY, REPORTING

Sample Specification

The contractor shall properly balance and adjust the air distribution systems as follows:

(1) Examine the air handling systems to see that they are free from obstructions. Determine that all dampers and registers are open; that moving equipment is lubricated; that filters are functioning; and perform other inspection and maintenance activities necessary for proper operation of the systems.

(2) Demonstrate that the air handling equipment performs as specified. Adjust variable type pulleys, volume and control dampers where necessary. This will not include replacing drives or motors unless so indicated.

(3) Adjust dampers and registers to distribute the air. Each register, diffuser and terminal unit shall deliver or remove the designed CFM in the proper pattern.

(4) Tabulate the results of testing on the approved forms and submit___ copies for approval and record.

(5) Perform this work in accordance with the procedures and standards described in the SMACNA Balancing and Adjusting Manual. Reports are to be made on SMACNA forms or facsimiles thereof.

Method and Accuracy

The balancing method described in Section II of the Manual, consists primarily of reading air flows in velocities or velocity pressures and controlling the flow by means of dampers or registers to achieve the desired results. The instruments to use for this purpose are described in Section IV of the Manual. Slight inaccuracies occur in such instruments as the anemometer and velometer. These inaccuracies are frequently compounded by taking readings under field conditions and by making slide rule calculations. Therefore, results at terminal units that read plus or minus 10% of design are considered acceptable.

It is not desirable to depend on using terminal unit dampers or registers for more than minor control of air flow. One obvious danger from so doing is that excessive throttling of terminal unit dampers will create high velocity noises. Therefore, it behooves the system designer to show on plans and specify sufficient manual balancing dampers at main trunk and branch intersections and scoop devices at outlet branches to provide the means to accomplish major diversions of air within the duct system.

Manufacturers' data must be available to the air balancing technician if he is to do a proper job. In many cases, such as terminal units, the manufacturer also recommends a test procedure. The recommended procedure should be followed by the balancer.

Reporting

The air balancing technician should report on his work both to show that it has been done and also to provide an exact record of system performance and design variance where such occurs. To simplify and standardize the reporting procedure, the following forms are recommended and may be procured in bulk from SMACNA. They are printed on tracing paper so that as many copies can be made as may be required for submission and for the balancers records.

The form shown on page 4 is for reporting the performance of air handling apparatus. In addition to air handling performance data such as CFM, static pressure, rpm, and brake hosepower, there is space for other useful information. Recording the shaft and keyway sizes and data on drives may have no immediate value but such information can save the engineer, owner and/or contractor considerable time and trouble in the future if changes are required in the system.

The form shown on page 5 is for reporting the performance of terminal units. A numbering system is used here to identify terminal units in addition to data as to type and size. These numbers should coincide with a numbering system used in a schematic layout described in Section II. They should also be marked on the set of as built drawings maintained by the architect or builder or on the sheet metal fabrication sketches, as may be desired.

SMACNA

APPARATUS TEST REPORT

SYSTEM NO. ____________

	DESIGN	ACTUAL
MAKE & SIZE		
R.P.M.		
S.P. TOTAL		
S.P. SUCTION		
S.P. DISCHARGE		
DISCHARGE C.F.M.		
POSITION OF VORTEX DAMPER		
SHEAVE SIZE & MAKE		
SHEAVE BORE SIZE		
SHEAVE KEYWAY SIZE		
NUMBER OF BELTS, MAKE & SIZE		
CENTERS BETWEEN PULLEYS		
MOTOR FRAME & MFG.		
MOTOR H.P.		
MOTOR VOLTAGE		
F.L. AMPS		
MOTOR R.P.M.		
SHEAVE SIZE & MAKE		
SHEAVE BORE SIZE		
SHEAVE KEYWAY SIZE		

REMARKS–

JOB NO.		JOB NAME
DATE		
TESTED BY		ADDRESS

SMACNA

OUTLET TEST REPORT

SYSTEM NO. ____________________

NO.	SIZE	FACTOR	DESIGN C.F.M.	DESIGN VEL. OR S.P.	TEST NO.1 VEL. OR S.P.	TEST NO. 2 VEL. OR S.P.	TEST NO. 3 VEL. OR S.P.	ACTUAL FINAL VEL. OR S.P.	ACTUAL FINAL C.F.M.	REMARKS
TOTAL										

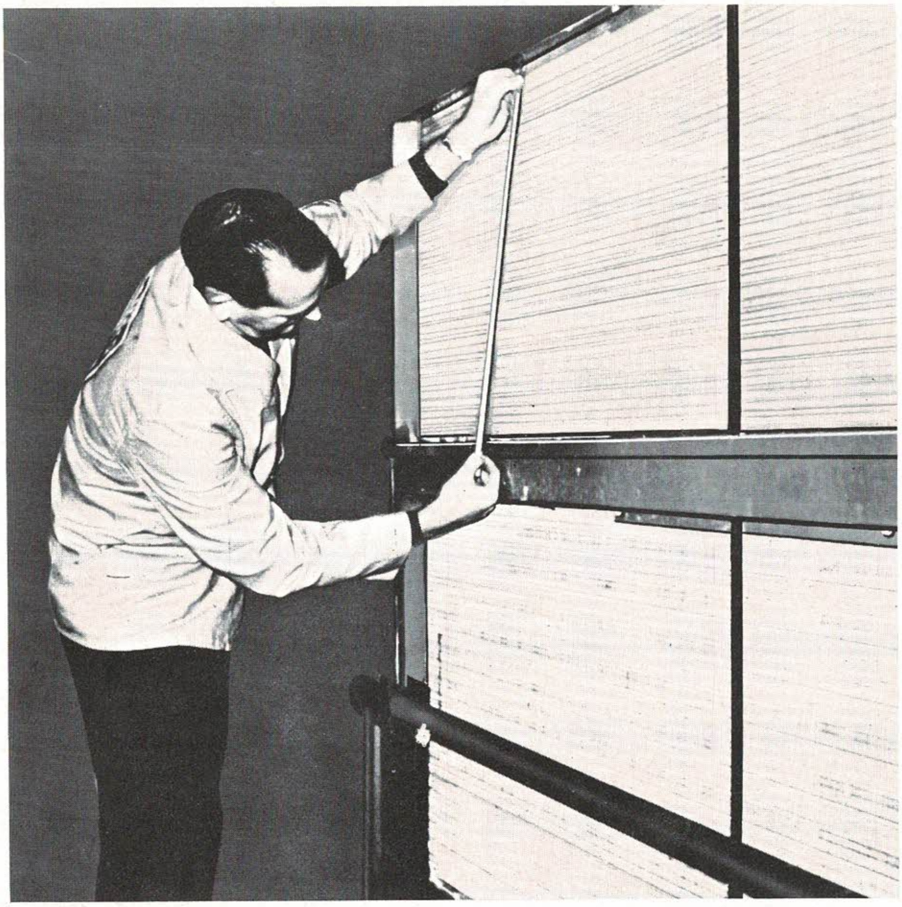
JOB NO. ____________________

DATE ____________________

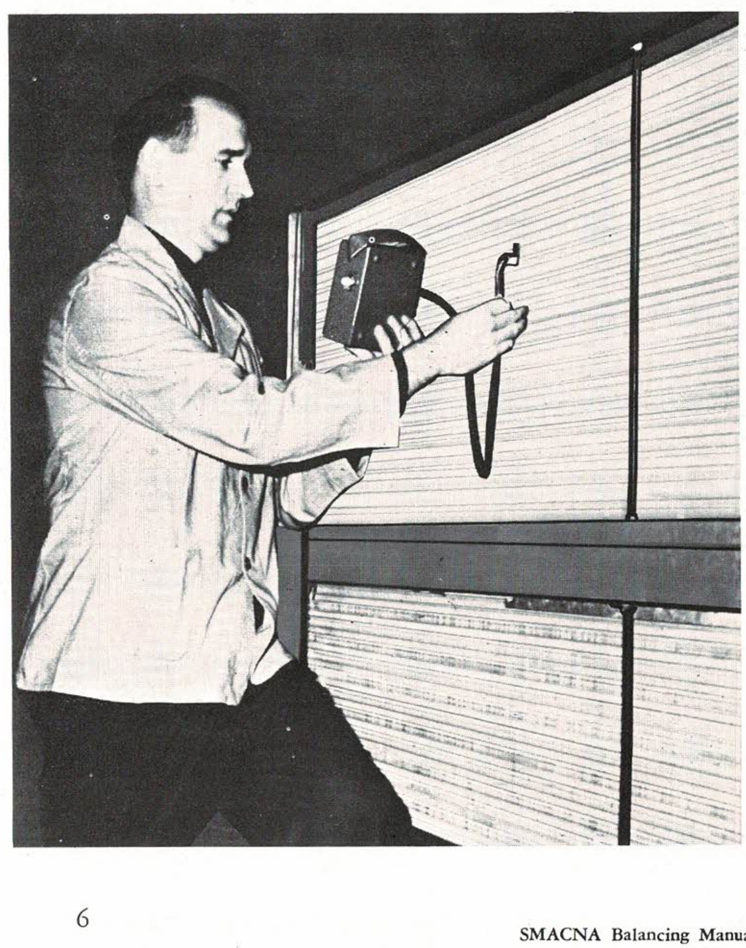
TESTED BY ____________________

JOB NAME ____________________

ADDRESS ____________________



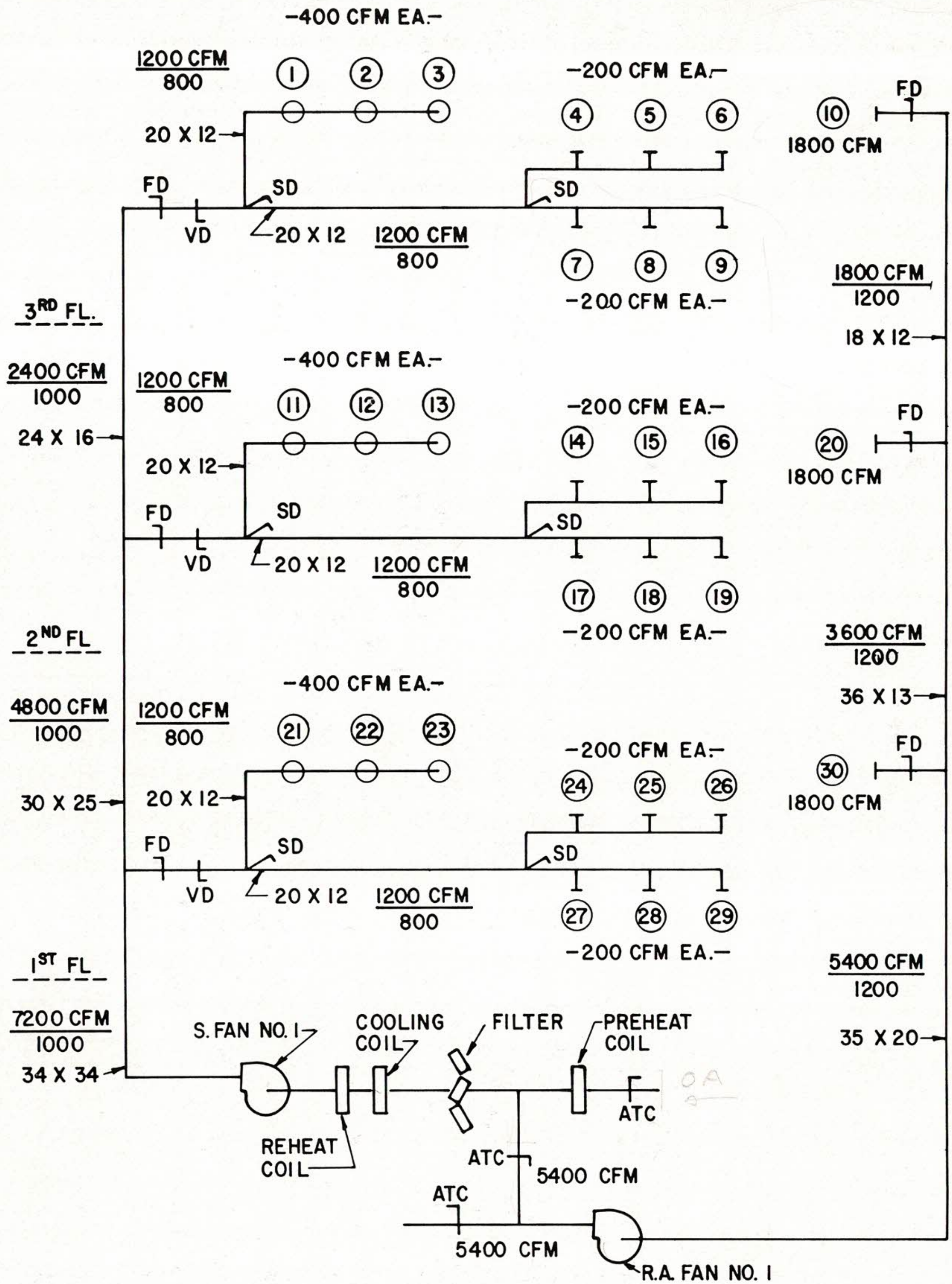
Before readings are made, the face area of the coil must be determined by measurement.



Balancer is reading velocity across the coil which, when multiplied by coil area (above) equals the air volume through the coil.

SECTION II

BALANCING and TESTING PROCEDURE



-400 CFM EA.-
1200 CFM
800
1
2
3
-200 CFM EA.-
4
5
6
10
FD
1800 CFM
20 X 12
FD
SD
SD
VD
20 X 12
1200 CFM
800
7
8
9
-200 CFM EA.-
1800 CFM
1200
3RD FL.
18 X 12
2400 CFM
1000
1200 CFM
800
-400 CFM EA.-
11
12
13
-200 CFM EA.-
14
15
16
FD
20
1800 CFM
24 X 16
20 X 12
FD
SD
SD
VD
20 X 12
1200 CFM
800
17
18
19
-200 CFM EA.-
2ND FL
3600 CFM
1200
36 X 13
-400 CFM EA.-
4800 CFM
1000
1200 CFM
800
21
22
23
-200 CFM EA.-
24
25
26
FD
30
1800 CFM
30 X 25
20 X 12
FD
SD
SD
VD
20 X 12
1200 CFM
800
27
28
29
-200 CFM EA.-
1ST FL
5400 CFM
1200
35 X 20
7200 CFM
1000
34 X 34
S. FAN NO. 1
COOLING COIL
FILTER
PREHEAT COIL
ATC
OA
REHEAT COIL
ATC
5400 CFM
ATC
5400 CFM
R.A. FAN NO. 1

FIG. 1

CHAPTER 2

PRELIMINARY PROCEDURE

1. Balancing can best be accomplished by following a systematic procedure. It is therefore recommended that a schematic layout of the system be prepared. Where there is more than one system, make a separate layout for each system. As shown on Fig. 1, all dampers, regulating devices and terminal units and outlets and inlets, should be indicated. Also show the sizes and CFM for main ducts; the sizes and CFM of outlets and inlets, and the quantities of fresh air, return air and relief air where such occur. Be sure to check shop drawings against design drawings so that changes made during construction will be shown on the schematic layout. In addition to the schematic layout of the system, the balancer should have available a set of design drawings, temperature control diagrams and equipment brochures.

2. As it is easier to work with velocities, it is recommended that all CFM figures be converted to velocities in fpm.

3. For purposes of reporting, number all outlets.

4. Fill out apparatus report sheets giving all required design and manufacturers data on the fan or fans or package equipment. (See Apparatus Test Report Form, page 10.) At this point corrections may have to be made for altitude and other special conditions. This is discussed in Section III of this Manual.

5. Fill out report sheets for all air outlets, inlets and/or terminal units of othertypes (See Outlet Test Report Form, Page 11.)

6. Where a study of the system has indicated that pitot tube traverses are desirable, prepare sheets for such traverses.

7. Study the system and determine what instruments will best do the job. Secure these instruments and check them for calibration. When possible, the same instrument should be used for the entire job because of possible errors in calibration. If more than one instrument of a similar type is used, a check should be made to compare how closely they read. The variation between instrument readings should not exceed plus or minus 5%.

It should be noted here that all this preparatory work can be done in the office or the shop of the contractor.

FIG. 2 Typical balancing instrument kit.

SMACNA FIG. 3 APPARATUS TEST REPORT

SYSTEM NO. S FAN #1

	DESIGN	ACTUAL
MAKE & SIZE	PDQ 3½	XYZ #33
R.P.M.	1172	1150
S.P. TOTAL	1¼"	1¼"
S.P. SUCTION	NOT SPEC.	¾"
S.P. DISCHARGE	NOT SPEC.	½"
DISCHARGE C.F.M.	7200	7200
POSITION OF VORTEX DAMPER	NONE	NONE
SHEAVE SIZE & MAKE	NOT SPEC.	DEF 9.4
SHEAVE BORE SIZE	1 11/16	1 11/16
SHEAVE KEYWAY SIZE	3/8 × 3/16	3/8 × 3/16
NUMBER OF BELTS, MAKE & SIZE	NOT SPEC.	2-DEF B136
CENTERS BETWEEN PULLEYS	NOT SPEC	56"
MOTOR FRAME & MFG.	NOT SPEC	#184 - SPINNER
MOTOR H.P.	2	2
MOTOR VOLTAGE	440	440
F.L. AMPS	NOT SPEC	3.3
MOTOR R.P.M.	1750	1750
SHEAVE SIZE & MAKE	NOT SPEC	DEF 6.2 ADJ.
SHEAVE BORE SIZE	7/8	7/8
SHEAVE KEYWAY SIZE	3/16 × 3/16 × 1 3/8	3/16 × 3/16 × 1 3/8

REMARKS- 1.89 BHP

JOB NO. 1234	JOB NAME BULL STORE
DATE 4/6/64	100 MAIN
TESTED BY LM	ADDRESS ANYTOWN

SMACNA

FIG. 4 OUTLET TEST REPORT

SYSTEM NO. S & R #1

NO.	SIZE	FACTOR	DESIGN C.F.M.	DESIGN VEL. OR S.P.	TEST NO. 1 VEL. OR S.P.	TEST NO. 2 VEL. OR S.P.	TEST NO. 3 VEL. OR S.P.	ACTUAL FINAL VEL. OR S.P.	ACTUAL FINAL C.F.M.	REMARKS
1	10	.478	400	837	860	830		820	392	CD
2	10	.478	400	837	835	820		820	392	CD
3	10	.478	400	837	795	820		825	393	CD
4	16x6	.44	200	455	600	450		470	207	TR-E SET
5	16x6	.44	200	455	546	400		450	198	TR-E SET
6	16x6	.44	200	455	462	380		420	185	TR-E SET
7	16x6	.44	200	455	392	500		490	215	TR-E SET
8	16x6	.44	200	455	310	450		455	200	TR-E SET
9	16x6	.44	200	455	300	450		450	198	TR-E SET
10	30x24	3.9	1800	462	325	430		436	1700	RETURN
11	10	.478	400	837	840	830		830	397	CD
12	10	.478	400	837	790	820		820	392	CD
13	10	.478	400	837	770	800		800	382	CD
14	16x6	.44	200	455	387	465		465	205	TR-E SET
15	16x6	.44	200	455	380	460		460	203	TR-E SET
16	16x6	.44	200	455	360	440		440	194	TR-E SET
17	16x6	.44	200	455	465	440		440	194	TR-E SET
18	16x6	.44	200	455	455	435		435	192	TR-E SET
19	16x6	.44	200	455	435	427		427	188	TR-E SET
20	30x24	3.9	1800	462	490	449		449	1750	RETURN
21	10	.478	400	837	856	845		845	403	CD
TOTAL	PAGE 1 OF 2									READ BY ALNOR VELOMETER

JOB NO. 1234

DATE 4/6/64

TESTED BY LM

JOB NAME BULL STORE

100 MAIN

ADDRESS ANYTOWN

CHAPTER 3

EQUIPMENT AND SYSTEM CHECK

1. Check all fans and fan components in the system or systems. Starting with the fan, make sure that the wheel is free and the housing clear of obstructions. Be sure that bearings are lubricated. Examine drives for proper tension and alignment. Vibration eliminators should be properly adjusted and be free to perform their function. Motor starters should contain the proper size overload protections and disconnect switches should be equipped with the proper sized fuses. Turning to the motor, again check lubrication. "Bump" the motor to see that the motor and fan have the correct rotation.

2. Physically examine the system or systems to see that all outlet, inlet or terminal dampers are in the desired position. All fire dampers should be examined to make sure that they are open. Filters should be in place and filters and coils should be clean. If the system has spray systems, they should be clean and functioning.

3. Check the delivered temperature conditions of the air in the system to be tested to determine if the air weight and volume (See Sec. III, Chapter 11, Psychometrics) during the test will have an appreciable difference from summer operating conditions. If necessary, adjust the calculations on the schematic testing sketch to compensate for the difference. This may vary at certain times.

4. Where required, disconnect ATC dampers from motors and set in position for summer system operation. In the case of multi-zone systems, see that all zone dampers are wide open; that the hot deck dampers are closed and that the cold deck dampers are wide open. Be extremely careful in securing dampers in position. If dampers slam shut when the fans are operating, the central apparatus could be damaged.

5. Turn on all systems in the area that affect each other, such as toilet and general exhaust. Close windows and doors and do whatever else is necessary to have the building and system under normal operating conditions.

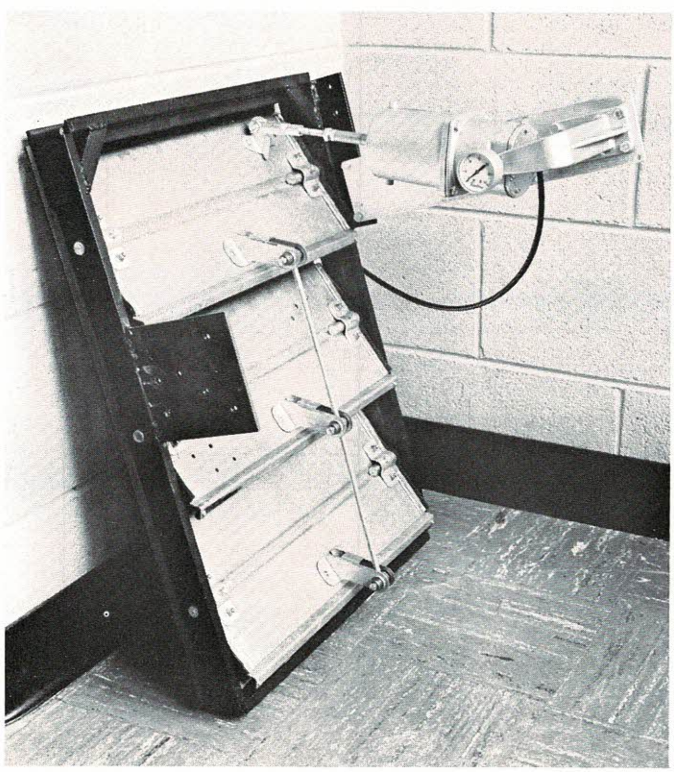


FIG. 5

Unmounted damper showing motor and linkage.

CHAPTER 4

AIR HANDLING EQUIPMENT

1. The obvious starting point for physically balancing the system or systems is the fan and the first step is to determine the air volume of the system. Check the speed of the fan or fans. (Fig. 6) Next take amperage and voltage readings to calculate the brake horse power. (Fig. 7) Follow this with a static reading across the fan.

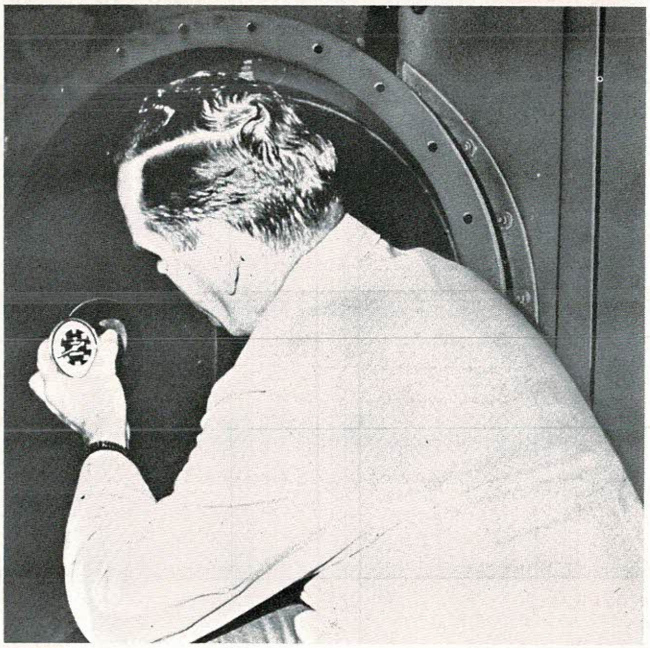
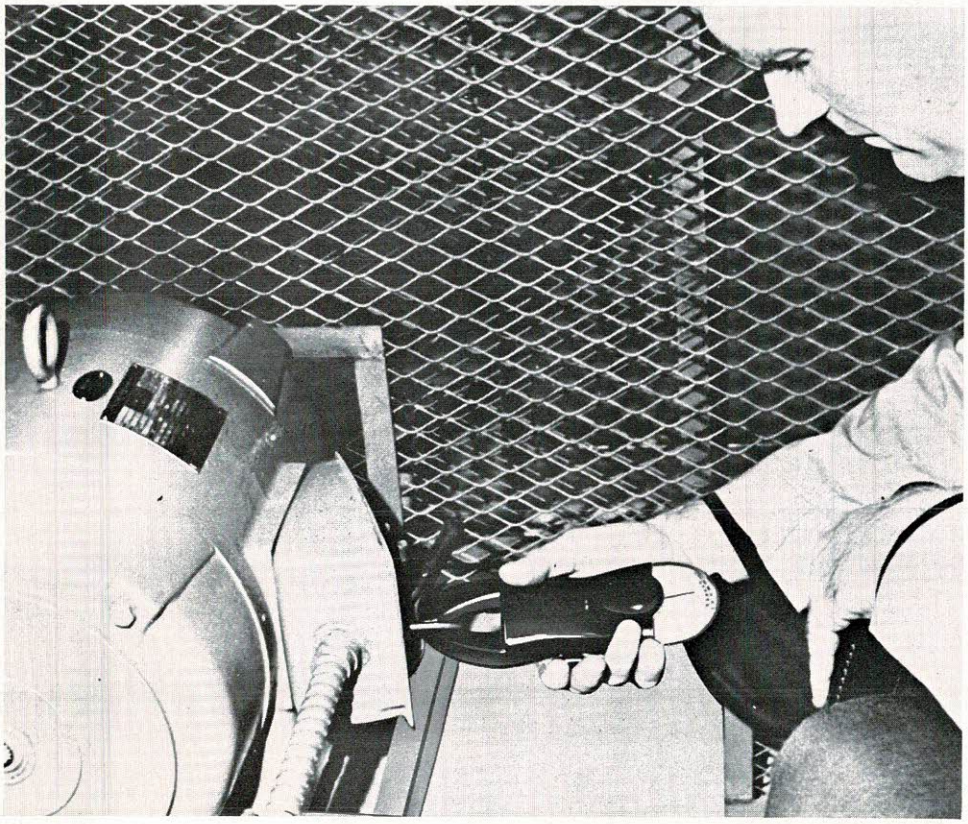


FIG. 6
The balancer is checking the rotation speed of the fan.

FIG. 7
Amperage and voltage readings are taken to calculate the brake horsepower being "pulled".



2. Using this information, consult the manufacturers performance tables or curves to determine the system CFM. A check reading to confirm the interpretation of the manufacturers tables or curves, should be made by taking an anemometer or velometer reading across the coils or filters. (Fig. 8 and Fig. 9)

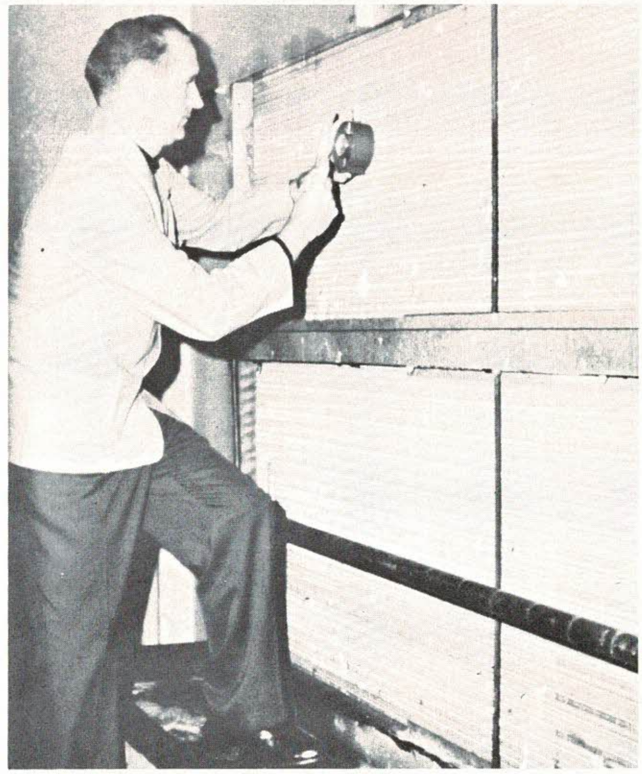


FIG. 8
The balancer is taking a reading across the coils to determine velocity using a rotating vane anemometer.

FIG. 9
The balancer is taking a reading across the filter to determine velocity using a velometer.

3. Adjust the system inlet arrangement in correct proportion to the CFM being handled by the fan. If there is a separate return air fan in the system follow steps 1 and 2 with the return air fan and adjust the return air damper. (Fig. 10)

4. Recheck the fans after each damper adjustment.

5. If fans are functioning above or below the design CFM, then adjust the speed to attempt to meet desired conditions. Check amperage carefully during this step to see that motors do not overload. (See Section III Chapter 15, Fans and Fan Laws).

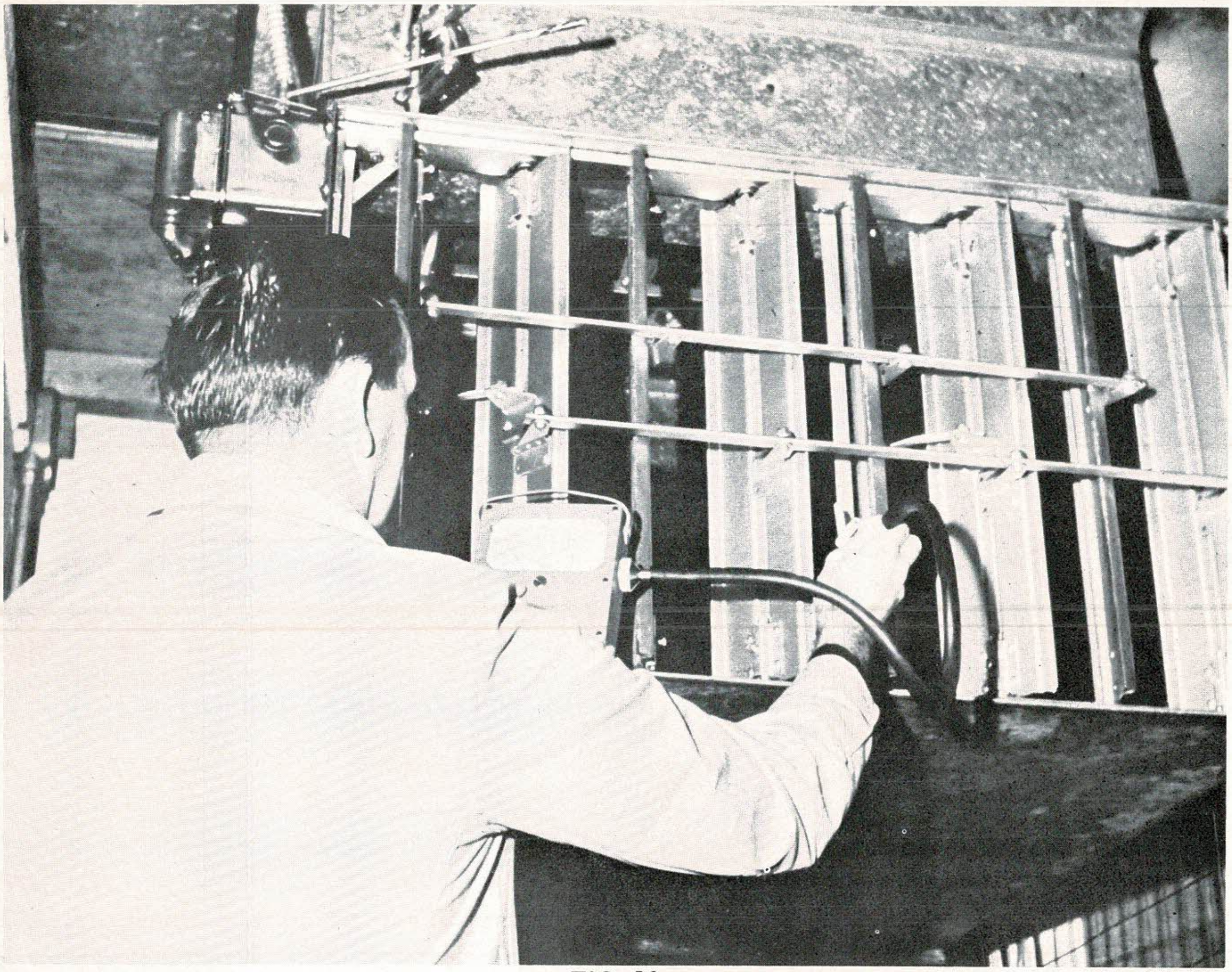


FIG. 10

In separate return air systems, the dampers should be checked to make sure the CFM is correctly proportioned.

CHAPTER 5
DUCT SYSTEM BALANCING

1. Take pitot traverse of main branch ducts. Adjust volume dampers to deliver the design CFM. Continue this process throughout the system, reading and adjusting all major branch ducts. (See Section IV, Chapter 19, Pitot Tube, for conditions under which pitot traverse is accurate.) It is not always possible to take meaningful pitot tube readings.

2. Recheck Fans to see if conditions have changed.

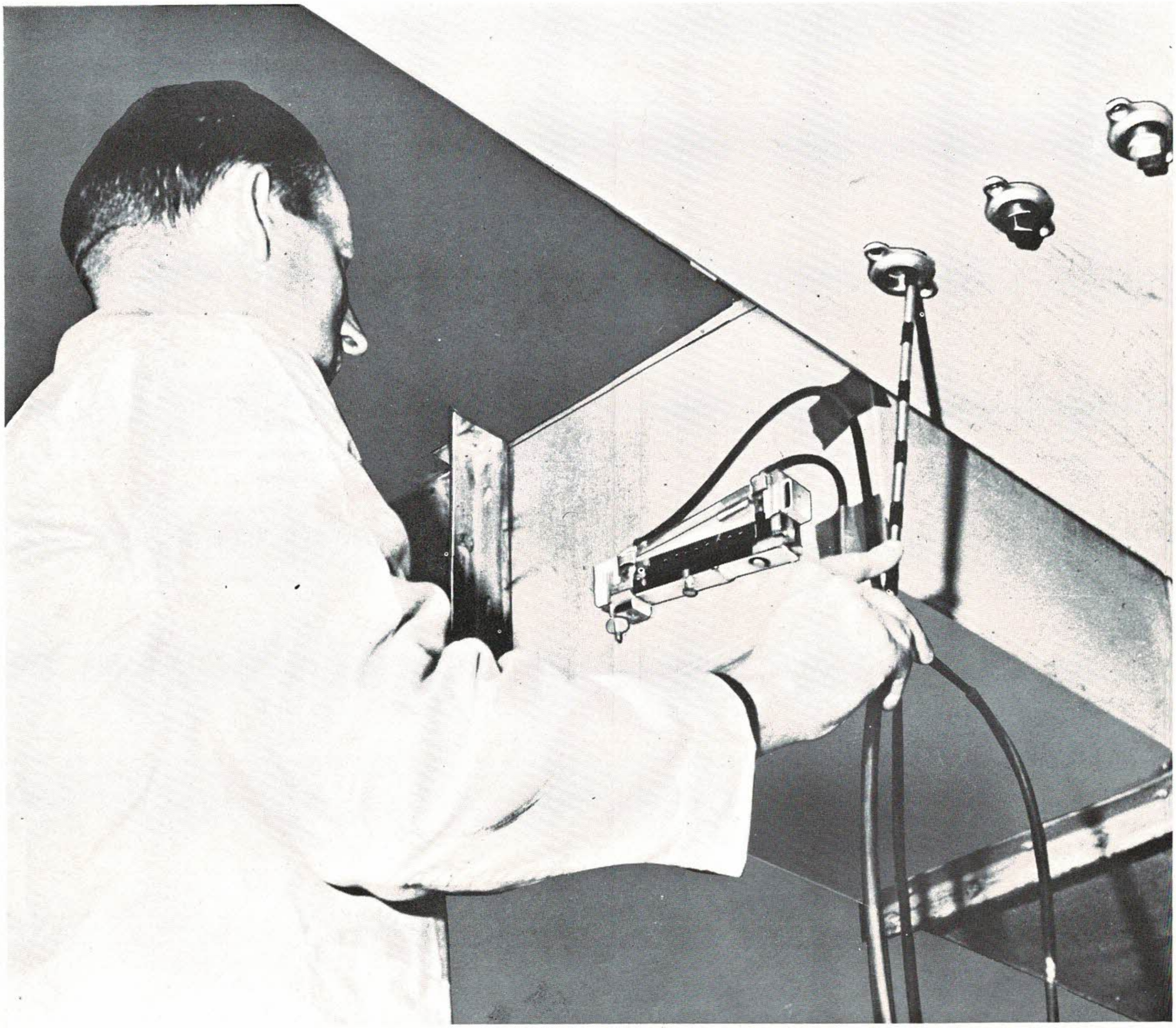


FIG. 11
Pitot tube traverses are one of the balancing technicians most important "readings". Here the dampers to major branches are being checked for design CFM.

CHAPTER 6

OUTLET, INLET AND TERMINAL UNIT BALANCING

1. At this point balance outlets. A suggested procedure is to first check the furthest outlet in each branch. If the furthest outlet velocity is below design, leave the damper full open and proceed to the next outlet upstream, adjusting as necessary until all the outlets in the branch have been adjusted. If the outlet velocity is above design, then throttle and proceed to the next outlet.

2. It is important that the balancer refers to the manufacturers' data for the proper "K" factor to use in conjunction with the instrument to be used. As will be seen in Section IV, Chapter 21, these factors will vary with the use of different instruments. It is also important to note that the manufacturer will designate various locations for taking diffuser readings on different models of diffusers.

3. Since the adjustment of one outlet does affect the others, it may be necessary to repeat this one or more times making finer adjustments each time.

It should be noted here that the adjustment of scoop devices is preferred for modifying outlet capacity rather than the adjustment of outlet dampers. If the throttling process involves closing of terminal dampers to a degree that generates noise, then re-examine the branch duct capacity for design CFM.

Now that the supply and return fans have been checked and adjusted and the supply outlets have been balanced, it is necessary to follow the same procedure with the return outlets, while rechecking fan performance.

Set the central apparatus dampers for winter operation. In cooperation with the automatic temperature control installer, set the damper operators so that they will not open beyond the set points. (See Section II, Chapter 10)

Prepare all reports and submit as required.

FIG. 12

The velometer is being used to take a reading at a light troffer. Note location of the probe.

FIG. 13

The readings taken at diffusers must be made in accordance with the manufacturer's recommendations as to location of the probe.

FIG. 14
This sidewall grille is being checked for CFM delivery with an anemometer.

FIG. 15
The velometer can be used to check side wall grilles.

Other methods have been developed to accomplish this final balancing process. The "ratio method" outlined in the ASHRAE Guide is as follows:

All steps up to the outlet balancing are the same as previously outlined. Then begin balancing the supply system at the last outlet on the branch farthest from the fan (branch number 1). This is outlet number 1, number 2 is the next to last outlet.

Measure the air flow at outlet number 1(Q_m) and compare with the design air flow for that outlet (Q_d), record the ratio, (Q_m/Q_d).

Measure the air flow at outlet number 2 and determine the ratio $(Q_m/Q_d)_2$. Compare $(Q_m/Q_d)_2$ and $(Q_m/Q_d)_1$.

If these ratios are not within 10 percent of each other, adjust outlet number 2 to bring ratios into closer agreement . . . DO NOT ADJUST OUTLET NUMBER 1.

Measure and again determine (Q_m/Q_d) and $(Q_m/Q_d)_1$ and compare. If these are within 10 percent of each other, no further adjustment is necessary. Proceed to outlet number 3.

Determine $(Q_m/Q_d)_3$ and compare with $(Q_m/Q_d)_2$. If If necessary, adjust number 3 so that $(Q_m/Q_d)_3$ and $(Q_m/Q_d)_2$ do not vary by more than 10 percent. DO NOT ADJUST OUTLETS 1 or 2. (Adjustment of outlet 3 automatically changes the Q_m/Q_d ratios of outlets 2 and 1. The ratios for all these outlets approach the same values. For this reason, once the outlet has been adjusted, it does not require further adjustment.)

Proceed to outlet number 4 and adjust to obtain agreement between $(Q_m/Q_d)_4$ and $(Q_m/Q_d)_3$.

After all outlets on branch number 1 are proportionately balanced to each other, proceed to branch number 2, etc.

Upon completion of proportionate balancing of all outlets the branches should be proportionately balanced.

Select typical outlets in branches 1 and 2 . . . Adjust number 2 branch damper to obtain agreement of the Q_m/Q_d ratios for the two branches. Proceed in like manner to obtain agreement between branches 2 and 3, 3 and 4, etc.

Upon completion of proportionate balancing, recheck the fan capacity. Adjust the fan speed to obtain a Q_m/Q_d ratio of 1.0 at the fan. Since the system has been proportionately adjusted, the Q_m/Q_d ratio throughout the system will be approximately 1.0 and the flow from each outlet will be the design air flow rate.

CHAPTER 7

VARIATIONS

The basic steps previously outlined form the foundation for balancing any system. There are, however, a number of variations to the conventional system that warrant review here.

Dual Duct Systems –

Most dual duct systems are designed to handle 100% of the total system supply through the cold duct, and from 75% to 100% through the hot duct. Balancing should be accomplished as follows:

a. Set all room control thermostats for maximum cooling, thus opening the cold valves to full open.

b. Repeat steps in Chapter 4 and 5, to check the apparatus and main trunk capacities.

c. Proceed to the extreme end of the system and check with an inclined gage, the static pressure immediately ahead of the last units. The static pressure of these points should be equal to, or in excess of, the minimum static pressures recommended by the dual duct box manufacturer. Normally, we would be looking for approximately .2" S.P. for a dual duct pneumatic constant volume box, and in the range of .75" S.P. for the mechanical volume regulator.

d. For those units that are not factory preset, the next step is to dial the proper setting on the CFM calibrated scale on the mechanical volume regulator type; or with the pneumatic volume regulator the pressure differential across the balancing orifice must be read and with the manufacturer's calibration curve, the proper air delivery can be set. (Fig. 16)

e. Balance diffusers or grilles on the low pressure side of the box as previously described.

f. Change the control setting to full heating, to make certain that the controls and dual duct box function properly, by re-checking air flow at one diffuser.

Induction Systems

In Chapter 17 a general description of induction systems may be found. Most induction systems use high velocity air distribution. Balancing should be accomplished as follows:

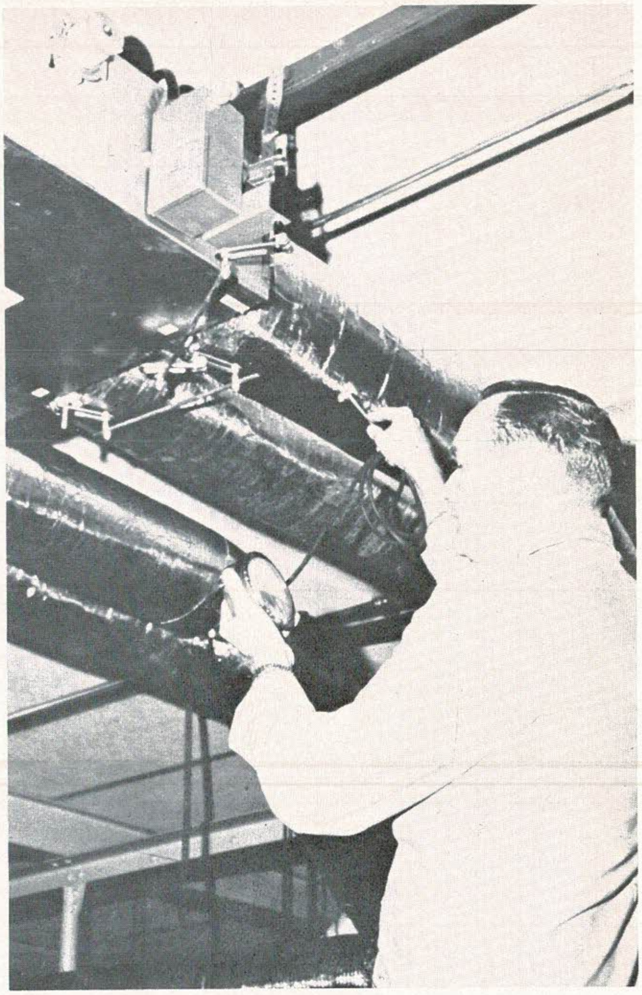


FIG. 16

Static pressure check before the mixing box.

1. Perform steps outlined in Chapters 4 and 5 to check the apparatus and main trunk capacities.

2. Primary air flow from each unit can be determined by reading the unit plenum pressure with a portable dry type of draft gage or Magnehelic gage. The unit manufacturer will furnish curves or charts of CFM vs. static pressure.

3. Make a spot check of the air distribution by reading the first and last unit on each riser. Do not reset units. Study these results and then adjust riser dampers to regulate proper flow in each riser. Normally the high risers will be cut back to improve the air flow in the rest of the system.

4. Start the first pass around the system, reading and

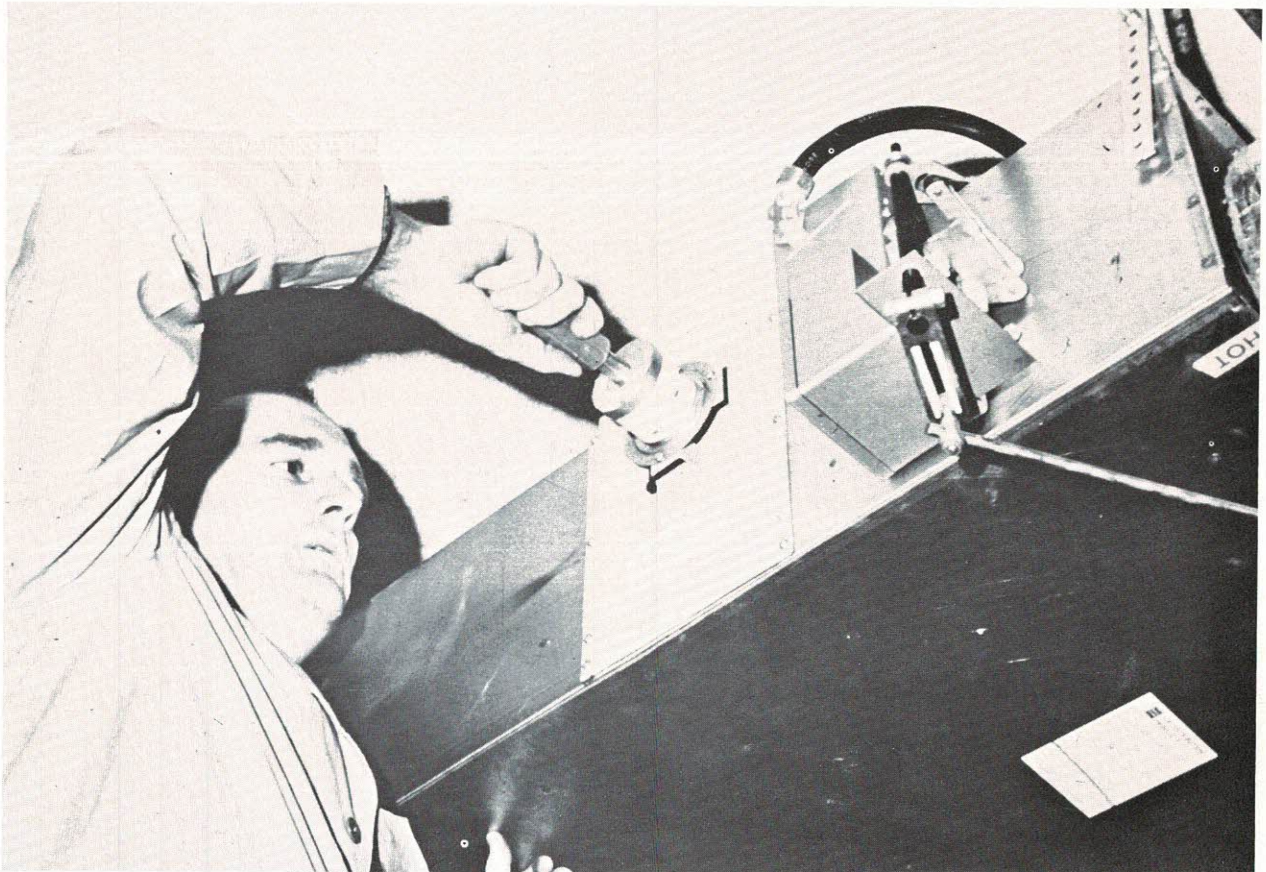


FIG. 17 – Adjusting air volume control on mixing box

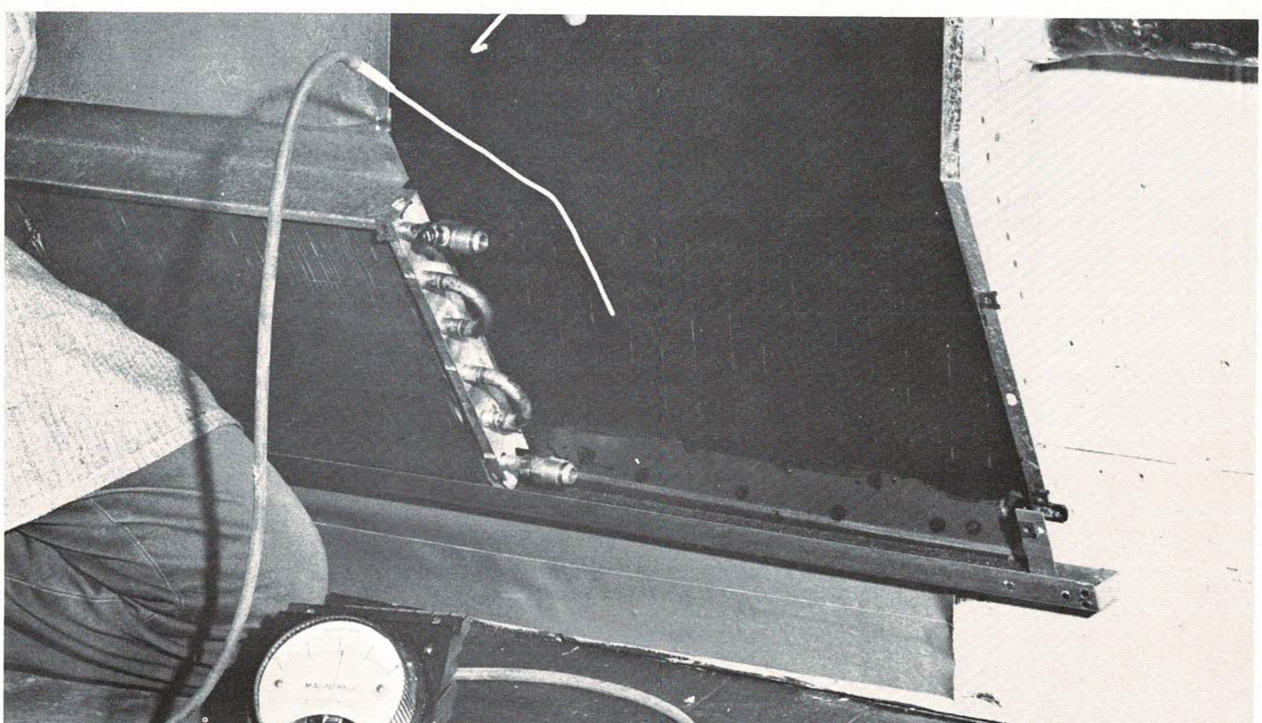


FIG. 18 – Reading nozzle pressure with probe inserted in a nozzle of induction unit.

adjusting as you go. Start on the floor nearest the supply header. If all unit dampers were open, set the units on the floor nearest the header about 10% under design.

As the units further down the system are throttled back to design, the initial units will increase in flow.

5. Normally about three complete passes around the entire system are required for proper adjustment.

6. If unit dampers are throttled excessively, noise will result. Use branch and riser dampers so that this will not result.

7. Normally the flow of water is automatically controlled to adjust room temperature. Some systems use the primary air source to power the controls and move a secondary air damper for adjusting room temperature. Here it is extremely important that the manufacturer's minimum static pressure in the plenum of each unit be maintained.

Ceiling Inductor –

The system employing the single inlet ceiling inductor is balanced in a manner similar to the system outlined in Step 2. However, it is very important to follow the manufacturer's recommendation for setting the units.

Terminal Reheat Systems –

Low and medium pressure terminal units, including reheat types, should be treated the same as any other low pressure outlet except that readings should be taken as directed by the manufacturer and without heat in the reheat coils.

Ceiling Plenums –

Perforated ceiling type installations are usually designed to maintain a given pressure above the ceiling for correct penetrations and distribution of the air. It is important, before attempting differential pressure readings, to check for leakage in the perimeter of the plenum. Leakage between the tiles is also critical as it makes it difficult to determine the pressure differential between the plenum and the room. Finally, read the pressure differential with a micro-manometer and consult the manufacturers tables to determine the CFM.

Systems With Hoods –

Systems with hoods are found in various types of ventilating systems. Kitchen hoods for restaurants and institutions are the type most frequently encountered. Most kitchen hoods are designed for a face velocity of about 100 feet per minute. Proper flow is necessary to insure entrapment of steam and grease laden vapors. Some municipalities have ordinances setting minimum requirements for face velocities at kitchen hoods and further require that a city inspector be present at the time of balancing.

Kitchen make up air systems must be in operation when the balancing takes place. Sometimes make up is by means of relief grilles from other areas. The "Anemotherm" is a good instrument for measuring these low face velocities. Some swinging vane Anemoneters (Velometer) can be used at velocities under 100 FPM. A Pitot tube used with a Micromanometer can also be used. When taking a Pitot tube traverse of the duct from the hood, be sure to correct for density, if required, due to temperature. If a fan serves only one hood, totals can often be taken conveniently at the fan.

Fume hoods are encountered in laboratory buildings. Here experiments are carried on in enclosed hooded areas that are designed to prevent the escape of toxic or noxious fumes. Balancing should be done with the fume hood door set in the normal operating position. Some hoods have a built in make up air system to minimize the loss of conditioned air from the laboratory. This must also be accurately balanced. When toxic experiments are to be performed, a smoke candle test should be made to ensure that vapors do not escape.

Factory exhaust systems with hoods fall into two categories. One group is used in conjunction with dip tanks and plating tanks and are similar in many respects to laboratory fume hoods. Hoods are often placed at the end of the tank and make up hoods are placed at the opposite end. This permits vapors to be swept from the tank surface but still leave the top open for overhead handling equipment. Balancing is the same as for fume hoods.

Factory exhaust systems are also used to remove and convey materials or waste products. Sawdust, wood chips, paper trimmings, etc. are transported through exhaust systems. These systems must be balanced so that velocities do not fall below predetermined transport velocities, at which point the materials will drop out. Balancing of these systems is done with blast gates that are installed instead of dampers and are used to temporarily shut off unused branches. In addition to velocity readings, static pressure readings should be recorded at each hood or intake device. This will permit easy future checks designed to spot any deviations in exhaust volumes from original values.

CHAPTER 8

LEAK TESTING

High velocity ductwork is normally tested for leaks because efficient and satisfactory performance is affected by leaks to a greater extent than in low pressure systems as follows:

1. More air is lost through a given sized leak opening as the pressure difference increases.

2. Insulation applied over leaking high pressure ducts will often balloon or fail in other ways when the vapor barrier is subjected to a build up that approaches duct operating pressure.

3. Large leaks in a high pressure system will generate noise.

4. In high pressure induction systems using secondary water cooling coils, the cooling effect lost at the unit is greater than the cooling lost by the absence of some of the cold primary air. This is caused by reduction of induced secondary air over coil when primary air is decreased.

Allowable Leakage

A top quality high pressure duct system is not necessarily hermetically tight. Therefore, tests are not made by pressurizing a system, valving it off and observing pressure drop as would be done in a piping system.

It is desirable to test high pressure ductwork at 25% above its operating pressure. At this test pressure, a total system leakage of 1% of the total system CFM is reasonable. When a system is broken into convenient segments, to simplify testing, allowable leakage must be apportioned to each section.

To facilitate the testing of segments of large systems, the allowable leakage can be expressed in CFM as a percentage of the volume of the segment in cubic feet. A CFM leakage equal to 10% of the volume of a system is normally satisfactory. Thus, a duct segment 24" x 24" that is 100 feet long would have a volume of 400 cubic feet and would be allowed a leakage of 40 CFM.

In any event, all audible leaks must be repaired no matter how small the leakage.

Test Procedure

A complete system or a part of a system such as, a riser, is completely sealed off to prepare it for testing. Openings, branch take offs, etc., are sealed off using metal heads, plugs, balloons, or any other convenient device. A portable high pressure blower is used to build up the pressure to 25% above operating pressure (discharge pressure not fan total pressure). A special orifice meter is placed between the blower and the system to measure the amount of air entering the system to maintain the test pressure. The air entering the system is equal to the leakage out. The air leakage is measured by a manometer across the orifice. The system pressure is measured by another manometer.

Construction of Test Apparatus

A typical test apparatus for an average size system is shown in Fig. 19.

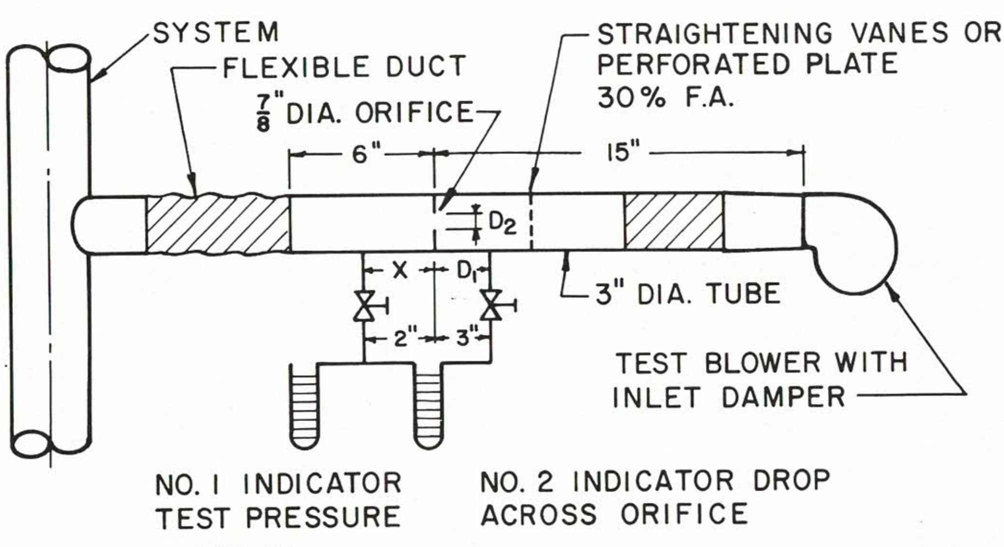


FIG. 19 Typical test rig for an average sized system.

The upstream pressure tap should be located a distance from the orifice equal to the tube diameter. The downstream pressure tap should be X times the tube diameter from the orifice as shown in the following table.

Diameter of Orifice / Diameter of tube	X
0.2	.74
0.3	.71
0.4	.66
0.5	.60
0.6	.53
0.7	.45
0.8	.36

The flow through such a test rig is determined as follows:

Flow across a sharp edge orifice is calculated using the following formula:

$$Q = 4005\ CA\ \sqrt{P}$$

Q = CFM
C = Orifice coefficient* (See Table below)
A = Area of orifice in square feet
P = Pressure difference in inches of water

Area of Orifice / Area of Tube		.1	.2	.3	.4	.5	.6	.7
3" ø tube	C	.619	.631	.653	.684	.728	.788	.880
12" ø tube	C	.610	.620	.637	.663	.700	.756	.846

LEAK TEST CURVE

$\frac{7}{8}$" DIA. ORIFICE 3" DIA. TUBE

C.F.M. = $10\sqrt{P}$

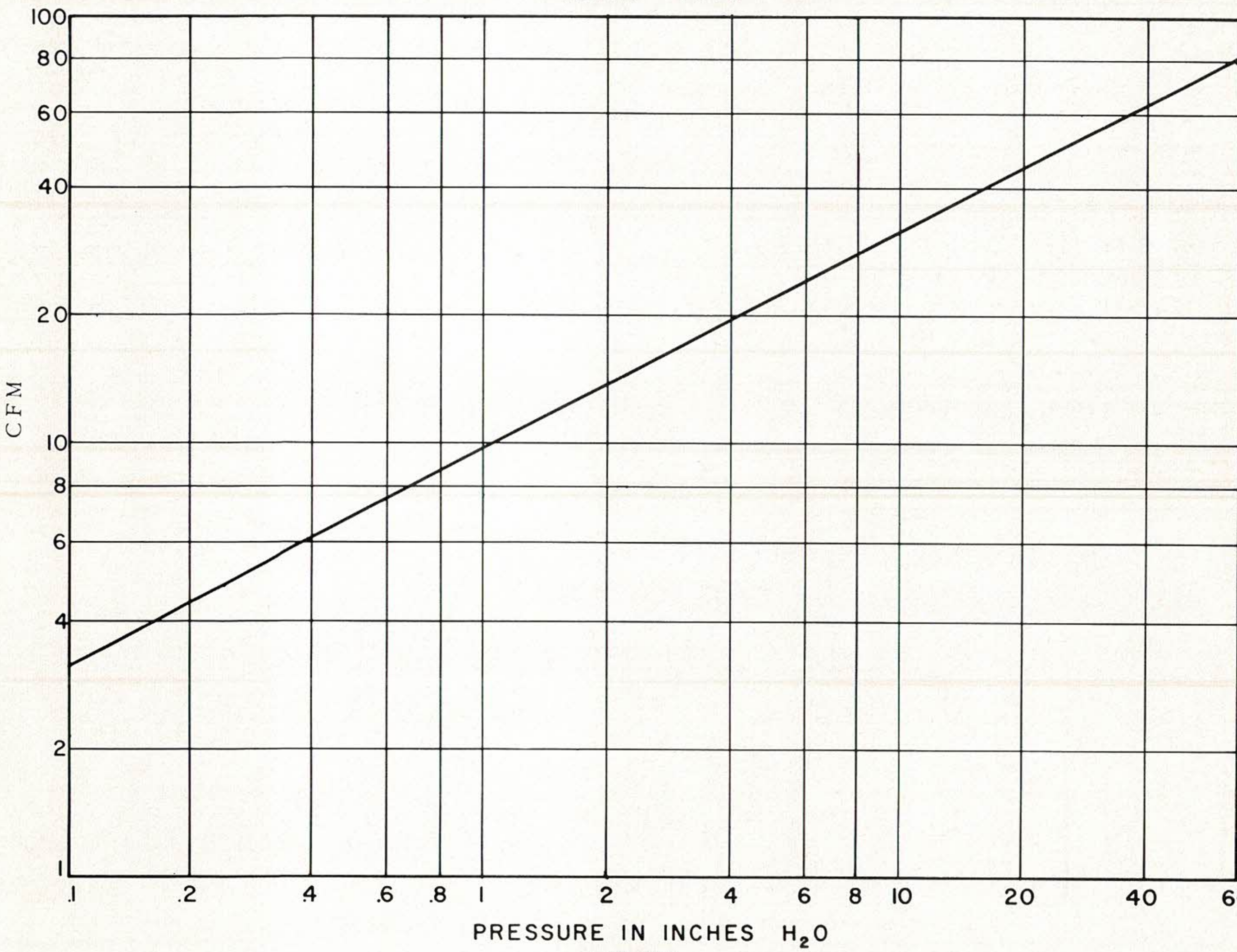


FIG. 20

Leak test curve shows graphically the formula - $Q = 4005\ CA\sqrt{P}$

For the Fig. 19 test rig, the formula works out to be:

$$CFM = 10 \sqrt{P}$$

This is more easily used when shown graphically on the preceeding page, Fig. 20.

Test apparatus of the foregoing size is satisfactory for leakage rates up to approximately 20 CFM, which is equal to 4 inches of pressure difference across the orifice.

For large systems a larger test apparatus using a larger tube and a larger orifice should be built. The curve can be calculated as shown above. Be sure to obtain a test blower capable of delivering a CFM equal to the maximum leak rate when operating at a static pres- of approximately 30 inches to allow for the system test pressure plus the drop within the test apparatus.

Detection and Repairs of Leaks

After the system is sealed off, use the test apparatus to build up the system to test pressure. Read the leakage on the manometer and refer to curve. Most leaks can be found audibly. Sometimes it will be necessary to use soap suds to find leaks. Leaks should be marked and then repaired after shutting down the blower. Do not retest until the sealer has set.

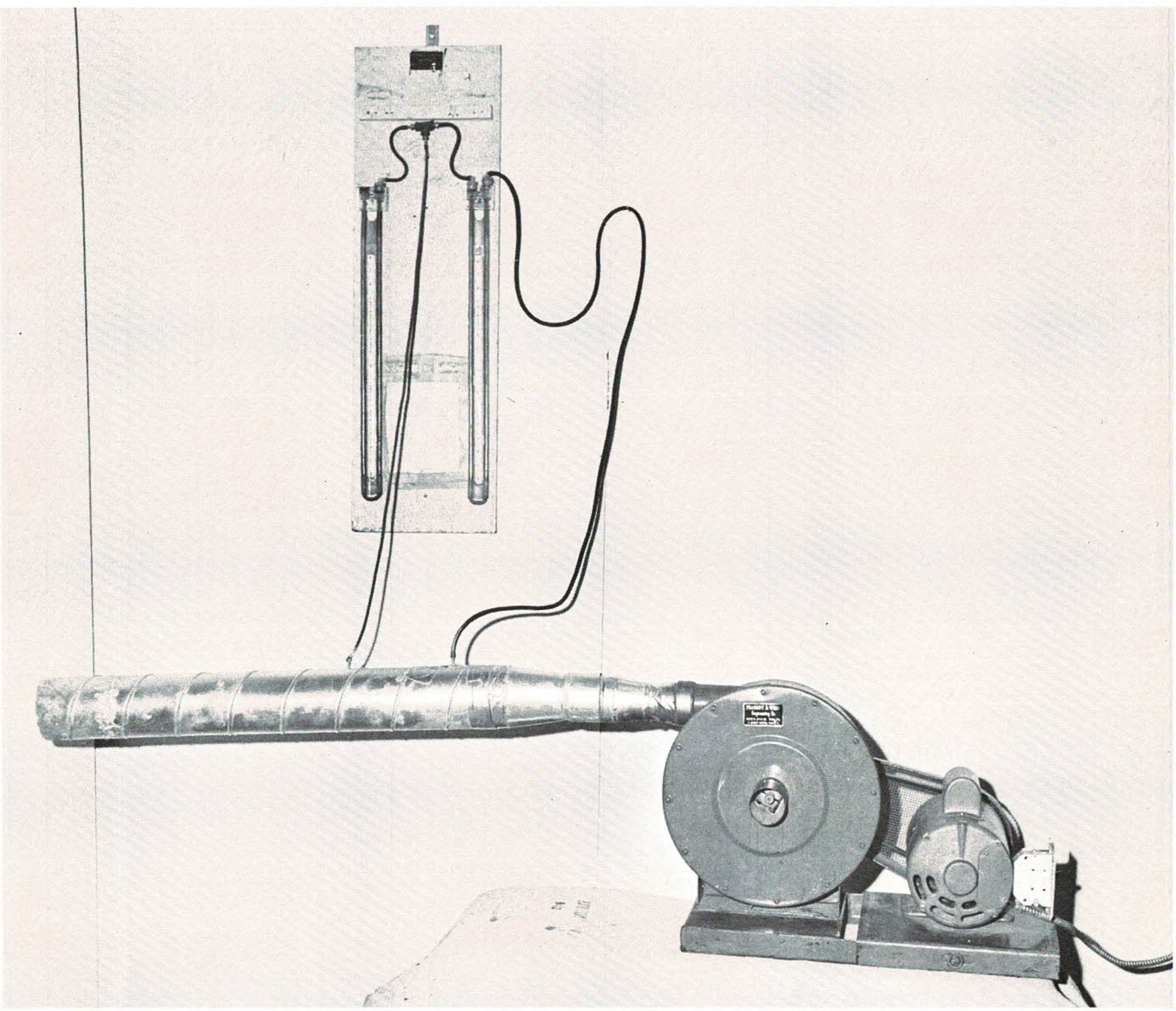


FIG. 21

Typical high pressure testing rig.

CHAPTER 9

WATER BALANCE

Although this manual stresses air balancing, it must be recognized that the chilled water, hot water and dual temperature water systems, when they exist, must also be balanced by the piping contractor. If the water temperature and flow quantities are not correct, the air temperatures leaving the apparatus containing coils will not be as designed. If the air system is completely balanced and temperatures do not meet design conditions, then the temperatures of the air leaving the coils must be checked against design.

Improper leaving conditions can be a result of incorrect water temperature or differences in flow from design. These situations can be readily checked by reading inlet and outlet water temperatures. These temperatures can be read by reading system thermometers, if the system is so equipped. Water pressure drop across coils can also be checked to determine proper water flow.

Other methods can be employed, such as using contact thermometers or bleeding water from the inlet and outlet drain headers over the thermometer bulb.

The air side performance of a cooling coil can be determined from the following formula:

$$\text{Btu per hr.} = 4.5\ \text{CFM}\ (h_{in} - h_{out})$$

To determine the enthalpy (h) readings, take the wet-bulb temperature of the entering air; find it on the dew point scale of the psychrometric chart and project along the wet bulb temperature lines for the proper enthalpy value.

Repeat the process for the leaving air. Enthalpies can be determined from Table 2 (Total Heat Content of Air at Various Wet Bulb Temperature) (page 26).

The formula to be used when a heating coil is involved is:

$$\text{Btu per hr.} = 1.08\ \text{CFM}\ (T_{out} - T_{in})$$

To determine the required water flow in the coil, the following formula may be used:

$$\text{GPM} = \frac{\text{Btu per hr.}}{500\ (T_{out} - T_{in})}$$

If these sample calculations indicate a water balance problem, the piping contractor should be notified.

TABLE 2

Total Heat Content of Air at Various Wet Bulb Temperatures

Interpolated to Tenths of Degrees, 40 WB – 79.9 WB

No. 1

WET BULB TEMP.	BTU PER POUND
40	15.23
41	15.70
42	16.17
43	16.66
44	17.15
45	17.65
46	18.16
47	18.68
48	19.21
49	19.75
50	20.30
51	20.86
52	21.44
53	22.02
54	22.62
55	23.22
56	23.84
57	24.48
58	25.12
59	25.78
60	26.46
61	27.15
62	27.85
63	28.57
64	29.31
65	30.06
66	30.83
67	31.62
68	32.42
69	33.25
70	34.09
71	34.95
72	35.83
73	36.74
74	37.66
75	38.61
76	39.57
77	40.57
78	41.58
79	42.62
80	43.69
81	44.78
82	45.90
83	47.04
84	48.22
85	49.43
86	50.66
87	51.93
88	53.23
89	54.56
90	55.93

No. 2

WET BULB TEMP.	BTU PER POUND	WET BULB TEMP.	BTU PER POUND	WET BULB TEMP.	BTU PER POUND	WET BULB TEMP.	BTU PER POUND	WET BULB TEMP.	BTU PER POUND	WET BULB TEMP.	BTU PER POUND	WET BULB TEMP.	BTU PER POUND	WET BULB TEMP.	BTU PER POUND
40.0	15.23	45.0	17.65	50.0	20.30	55.0	23.22	60.0	26.46	65.0	30.06	70.0	34.09	75.0	38.61
.1	15.28	.1	17.70	.1	20.36	.1	23.28	.1	26.53	.1	30.14	.1	34.18	.1	38.71
.2	15.32	.2	17.75	.2	20.41	.2	23.34	.2	26.60	.2	30.21	.2	34.26	.2	38.80
.3	15.37	.3	17.80	.3	20.47	.3	23.41	.3	26.67	.3	30.29	.3	34.35	.3	38.90
.4	15.42	.4	17.85	.4	20.52	.4	23.47	.4	26.74	.4	30.37	.4	34.43	.4	38.99
40.5	15.47	45.5	17.91	50.5	20.58	55.5	23.53	60.5	26.81	65.5	30.45	70.5	34.52	75.5	39.09
.6	15.51	.6	17.96	.6	20.64	.6	23.59	.6	26.87	.6	30.52	.6	34.61	.6	39.19
.7	15.56	.7	18.01	.7	20.69	.7	23.65	.7	26.94	.7	30.60	.7	34.69	.7	39.28
.8	15.61	.8	18.06	.8	20.75	.8	23.72	.8	27.01	.8	30.68	.8	34.78	.8	39.38
.9	15.65	.9	18.11	.9	20.80	.9	23.78	.9	27.08	.9	30.75	.9	34.86	.9	39.47
41.0	15.70	46.0	18.16	51.0	20.86	56.0	23.84	61.0	27.15	66.0	30.83	71.0	34.95	76.0	39.57
.1	15.75	.1	18.21	.1	20.92	.1	23.90	.1	27.22	.1	30.91	.1	35.04	.1	39.67
.2	15.80	.2	18.26	.2	20.97	.2	23.97	.2	27.29	.2	30.99	.2	35.13	.2	39.77
.3	15.84	.3	18.32	.3	21.03	.3	24.03	.3	27.36	.3	31.07	.3	35.21	.3	39.87
.4	15.89	.4	18.37	.4	21.09	.4	24.10	.4	27.43	.4	31.15	.4	35.30	.4	39.97
41.5	15.94	46.5	18.42	51.5	21.15	56.5	24.16	61.5	27.50	66.5	31.23	71.5	35.39	76.5	40.07
.6	15.99	.6	18.47	.6	21.20	.6	24.22	.6	27.57	.6	31.30	.6	35.48	.6	40.17
.7	16.04	.7	18.52	.7	21.26	.7	24.29	.7	27.64	.7	31.38	.7	35.57	.7	40.27
.8	16.08	.8	18.58	.8	21.32	.8	24.35	.8	27.71	.8	31.46	.8	35.65	.8	40.37
.9	16.13	.9	18.63	.9	21.38	.9	24.42	.9	27.78	.9	31.54	.9	35.74	.9	40.47
42.0	16.17	47.0	18.68	52.0	21.44	57.0	24.48	62.0	27.85	67.0	31.62	72.0	35.83	77.0	40.57
.1	16.22	.1	18.73	.1	21.49	.1	24.54	.1	27.92	.1	31.70	.1	35.92	.1	40.67
.2	16.27	.2	18.79	.2	21.55	.2	24.61	.2	27.99	.2	31.78	.2	36.01	.2	40.77
.3	16.32	.3	18.84	.3	21.61	.3	24.67	.3	28.07	.3	31.86	.3	36.10	.3	40.87
.4	16.37	.4	18.89	.4	21.67	.4	24.74	.4	28.14	.4	31.94	.4	36.19	.4	40.97
42.5	16.42	47.5	18.95	52.5	21.73	57.5	24.80	62.5	28.21	67.5	32.02	72.5	36.29	77.5	41.08
.6	16.46	.6	19.00	.6	21.78	.6	24.86	.6	28.28	.6	32.10	.6	36.38	.6	41.18
.7	16.51	.7	19.05	.7	21.84	.7	24.93	.7	28.35	.7	32.18	.7	36.47	.7	41.28
.8	16.56	.8	19.10	.8	21.90	.8	24.99	.8	28.43	.8	32.26	.8	36.56	.8	41.38
.9	16.61	.9	19.16	.9	21.96	.9	25.06	.9	28.50	.9	32.34	.9	36.65	.9	41.48
43.0	16.66	48.0	19.21	53.0	22.02	58.0	25.12	63.0	28.57	68.0	32.42	73.0	36.74	78.0	41.58
.1	16.71	.1	19.26	.1	22.08	.1	25.19	.1	28.64	.1	32.50	.1	36.83	.1	41.68
.2	16.76	.2	19.32	.2	22.14	.2	25.25	.2	28.72	.2	32.59	.2	36.92	.2	41.79
.3	16.81	.3	19.37	.3	22.20	.3	25.32	.3	28.79	.3	32.67	.3	37.02	.3	41.89
.4	16.86	.4	19.43	.4	22.26	.4	25.38	.4	28.87	.4	32.75	.4	37.11	.4	42.00
43.5	16.91	48.5	19.48	53.5	22.32	58.5	25.45	63.5	28.94	68.5	32.84	73.5	37.20	78.5	42.10
.6	16.95	.6	19.53	.6	22.38	.6	25.52	.6	29.01	.6	32.92	.6	37.29	.6	42.20
.7	17.00	.7	19.59	.7	22.44	.7	25.58	.7	29.09	.7	33.00	.7	37.38	.7	42.31
.8	17.05	.8	19.64	.8	22.50	.8	25.65	.8	29.16	.8	33.08	.8	37.48	.8	42.41
.9	17.10	.9	19.70	.9	22.56	.9	25.71	.9	29.24	.9	33.17	.9	37.57	.9	42.52
44.0	17.15	49.0	19.75	54.0	22.62	59.0	25.78	64.0	29.31	69.0	33.25	74.0	37.66	79.0	42.62
.1	17.20	.1	19.81	.1	22.68	.1	25.85	.1	29.39	.1	33.33	.1	37.76	.1	42.73
.2	17.25	.2	19.86	.2	22.74	.2	25.92	.2	29.46	.2	33.42	.2	37.85	.2	42.83
.3	17.30	.3	19.92	.3	22.80	.3	25.98	.3	29.54	.3	33.50	.3	37.95	.3	42.94
.4	17.35	.4	19.97	.4	22.86	.4	26.05	.4	29.61	.4	33.59	.4	38.04	.4	43.05
44.5	17.40	49.5	20.03	54.5	22.92	59.5	26.12	64.5	29.69	69.5	33.67	74.5	38.14	79.5	43.16
.6	17.45	.6	20.08	.6	22.98	.6	26.19	.6	29.76	.6	33.75	.6	38.23	.6	43.26
.7	17.50	.7	20.14	.7	23.04	.7	26.26	.7	29.84	.7	33.84	.7	38.33	.7	43.37
.8	17.55	.8	20.19	.8	23.10	.8	26.32	.8	29.91	.8	33.92	.8	38.42	.8	43.48
.9	17.60	.9	20.25	.9	23.16	.9	26.39	.9	29.99	.9	34.01	.9	38.52	.9	43.58

CHAPTER 10
AUTOMATIC TEMPERATURE CONTROL

To properly balance and adjust an air system, a thorough knowledge of the temperature control system is required. Fresh air and exhaust air dampers are usually electrically interlocked with the supply fan to open to a fixed minimum when fan is started, manually, or by a time clock, or other equipment interlock. A mixed air stat may then control the fresh air, return air, and exhaust dampers to maintain a set mixed air temperature. At a pre-set temperature or at high humidities of outside air, the outside air damper will often be returned to a minimum to decrease the cooling load of the outside air. In case of power or compressed air service failure, the outside air damper usually closes automatically. A freeze stat can also stop the fan and close the outside air dampers.

A discharge stat will often control a heating or cooling coil valve, face and by-pass dampers or mixing dampers. A room stat can control a hot water booster coil, a steam booster coil, chilled water booster coil, an electric booster coil, or hot and cold mixing dampers. A humidistat can control a humidifier or a cooling coil for dehumidification. Controls can be direct acting, reverse acting, modulating or snap acting, stepped, master, sub-master, series, parallel, and can control dampers, valves, relays, start, step, stop motors, fans, equipment and be controlled by time clocks, time delay relays, static pressure controllers, air switches, flow switches, level controllers, fire and smoke detectors and be connected to alarm systems, and be controlled, readjusted, indicated from or at remote control panels with multi-points and be very simple or very complex.

Temperature control systems use line voltage electricity, low voltage electricity, electronics, pneumatic air, or fluid power to control dampers, valves, switches, motors, etc. by modulation or snap action.

Probably the most important effect on air balancing and adjusting is the setting of the fresh air, return air, and exhaust dampers. After fan rpm and capacity has been checked out, set fresh air dampers for minimum outside air. Use thermometers or thermocouple at fresh air, return air and mixed air to measure temperature. Use the mixed air temperature formula that follows in Equation (1) to determine amount of outside air. Work with temperature control contractor to set the minimum outside air condition. Mark dampers for minimum and check CFM again.

If an economizer cycle is used, next check for 100% outside air and again test CFM. Follow procedure for 25%, 50%, 75% outside air working closely with temperature control contractor. The mixing of the return air with outside air to give a known mixed air temperature can be shown by Equation (1).

$$100\ T_m = X_o T_o + X_r T_r \tag{1}$$

T_m = temperature of the mixture of return and outdoor air, F.

X_o and X_r = percentage of outdoor and return air respectively

T_o = temperature of outdoor air, F.

T_r = temperature of return air, F.

The following equations are used for determining percentages of outside air. For this work, more convenient forms of expressing Equation (1) are given in Equations (2a) and (2b).

$$X_o = 100\,\frac{(T_r - T_m)}{(T_r - T_o)} \tag{2a}$$

$$X_r = 100\,\frac{(T_m - T_o)}{(T_r - T_o)} \tag{2b}$$

An example of this is as follows:

If 75° return air is mixed with 25° outside air and the mixed air temperature is 55° -- what is the percent of outside air?

$$X_o = \frac{(T_r - T_m)}{(T_r - T_o)}$$

$T_m = 55°$

$T_r = 75°$

$T_o = 25°$

$$X_o = \frac{(75-55)}{(75-25)} = 100\,\frac{(20)}{(50)} = 40\%$$

Similarly, if we want to set dampers for 15% minimum outside air when return air is 75°F. and outside air is −5°F. what will the mixed air temperature be?

$$100\ T_m = X_m T_o + X_r T_r$$

$$T_o = -5°F;\ T_r = 75°F;\ X_o = 15°;$$
$$X_r = 85°$$

$$100\ T_m = (15)\ (-5) + (85)\ (75)$$

$$100\ T_m = -75 + 6375$$

$$100\ T_m = 6300$$

$$T_m = 63°$$

Then proceed to work with temperature control contractor so that he sets minimum outside air dampers to give you 63°F. mixed air.

Have temperature control contractor adjust damper linkage until proper reading is made. Test fan CFM. In like manner, test fan CFM after dampers are adjusted by control contractor at 25%, 50%, 75% and 100% outside air. It is most important that traverses be made of mixed air temperatures to prevent stratification. Baffles, deflectors, perforated plates, etc. are sometimes required to prevent this stratification of air under certain conditions.

If a return exhaust fan is used, the adjustment of the manual return air damper is most important.

Sometimes the measurement of air temperature change, water flow through coils and water temperature change can be used to determine CFM if certified coil performance is available from the coil manufacturer.

No amount of adjustment of air volume will correct for a bad mixing damper linkage or a leaky valve that is supplying 60° supply air to space when 55° was design temperature. With a knowledge of the temperature control system, use thermometers and thermocouples to check complaints as well as your other air measuring instruments.

It is suggested that the balancer work with the temperature control contractor when setting damper linkages.

SECTION III

THEORY AND EQUIPMENT FUNDAMENTALS

CHAPTER 11
PSYCHROMETRICS

Introduction

The equipment designer must be well versed in psychrometrics to do his job. The system designer must know the psychrometric chart if he is to do his load calculations, select equipment and lay out the systems. The air balancing technician need not be experienced in psychrometrics, but a basic knowledge of the fundamentals will frequently be useful. A little grounding may help him in spotting equipment not operating properly. When the CFM in a given branch can not be measured, the figure can sometimes be calculated by analyzing the results of mixing. Serious leaks can be uncovered by the same method. When ducts are sweating or when windows are sweating, the reason can be learned from psychrometrics and sometimes corrections can be made. The purpose of this Section is to give the technician a brief grounding in the fundamentals of psychrometrics.

Properties of Air

Dry air is normally a mixture of fixed percentages of gases, primarily nitrogen and oxygen. Air in the atmosphere contains a small amount of water vapor in varying amounts. The amount of water vapor in the air normally represents less than 1% of the weight of the moist air mixture. Normally atmospheric air is only partially saturated with water vapor, and thus the vapor in the air is in a superheated state. Unsaturated air will readily pick up more water vapor if any is available.

The water vapor and the air each behave as if the other were not present. The pressure exerted by the water vapor is independent of the pressure exerted by the dry air. The total pressure corresponding to the to the barometric pressure is the sum of the two pressures, the partial pressure of the water vapor and the partial pressure of the air.

Air and Vapor Relationship

Throughout the normal ranges of atmospheric pressures and temperatures, both air and water vapor behave as perfect gases. That is the relationship between pressure, temperature, and volume as defined by the formula:

$$\frac{PV}{T} = R \qquad (1)$$

where: P = pressure (lb/sq. ft.)

V = volume of 1 lb (cu. ft.)

T = absolute temperature (459.6 + t°F)

$$R = \frac{1545}{\text{Molecular weight}}$$

Using the above equation we can calculate changes in air density or the volume for 1 pound of air as follows:

$$\frac{P_1 V_1}{T_1} = R = \frac{P_2 V_2}{T_2} \qquad (2)$$

Assuming constant atmospheric pressure, then:

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$

$$V_2 = \frac{V_1 T_2}{T_1} \qquad (3)$$

Now starting with "Standard Air" which is the fixed reference for air conditioning calculations, we have the following properties:

T = 70°F

P = 29.92 inch of mercury

V = 13.35 cu. ft. occupied by 1 lb. of air, or the density is .075 lb/cu. ft.

Example - -

What is the volume of one pound of air after it has been heated to 100°F? Using Equation No. 3 we have:

$$V_2 = \frac{V_1 T_2}{T_2} = 13.35 \frac{460 + 100}{460 + 70} = 13.35 \frac{560}{530}$$

V_2 = 14.1 cu. ft.

This would indicate that when the air temperature deviates greatly from "Standard Air", and volume is being measured, temperature corrections should be made.

Since the volume of a pound of air varies inversely with atmospheric pressure it would also be necessary to make corrections to measurements taken at high altitudes. Corrections of this type are usually ignored at elevations below 2000 feet above sea level. At 5000 feet altitude, however, the atmospheric pressure is 24.89 inches of mercury pressure. Our standard pound of 70 degree air at that altitude would have a volume of $\frac{29.92}{24.89} \times 13.35 = 16.05$ cu. ft. This is approximately 20% lighter and a cubic foot of this less dense air would have less heat carrying capacity.

CHAPTER 12

THE PSYCHROMETRIC CHART

On the following page is a reproduction of the American Society of Heating Refrigerating and Air Conditioning Engineers psychrometric chart. The skeleton chart (Fig. 22) below will show the arrangement of the various coordinates.

(1) Saturation temperature
(2) Dew-Point temperature
(3) Enthalpy
(4) Relative humidity
(5) Humidity ratio
(6) Wet-Bulb temperature
(7) Volume of mixture
(8) Dry-Bulb temperature

This chart is for standard barometric pressure of 29.92 inches of mercury. Similar psychrometric charts are available from other sources. Special psychrometric charts can be prepared for high altitudes such as would be incurred at Denver, Colorado, or Mexico City, Mexico.

Rather than go through calculations every time we need some information about properties of air, psychrometric charts have been devised that graphically represent the values of the various psychrometric properties. Following are definitions of the various terms and symbols used in psychrometrics:

Dry-Bulb Temperature (DB) The temperature of the air as registered by an ordinary thermometer. (t)

Wet-Bulb Temperature (WB) The temperature registered by a thermometer whose bulb is covered by a wetted wick and exposed to a current of rapidly moving air. (t')

Dew-Point Temperature (DP) The temperature to which air must be reduced in order to produce condensation of the moisture contained therein. (t_{dp})

Wet-Bulb Depression The difference between dry- and wet-bulb temperatures. ($t - t'$)

Dew-Point Depression. The difference between dry-bulb and dew-point temperatures. ($t - t_{dp}$)

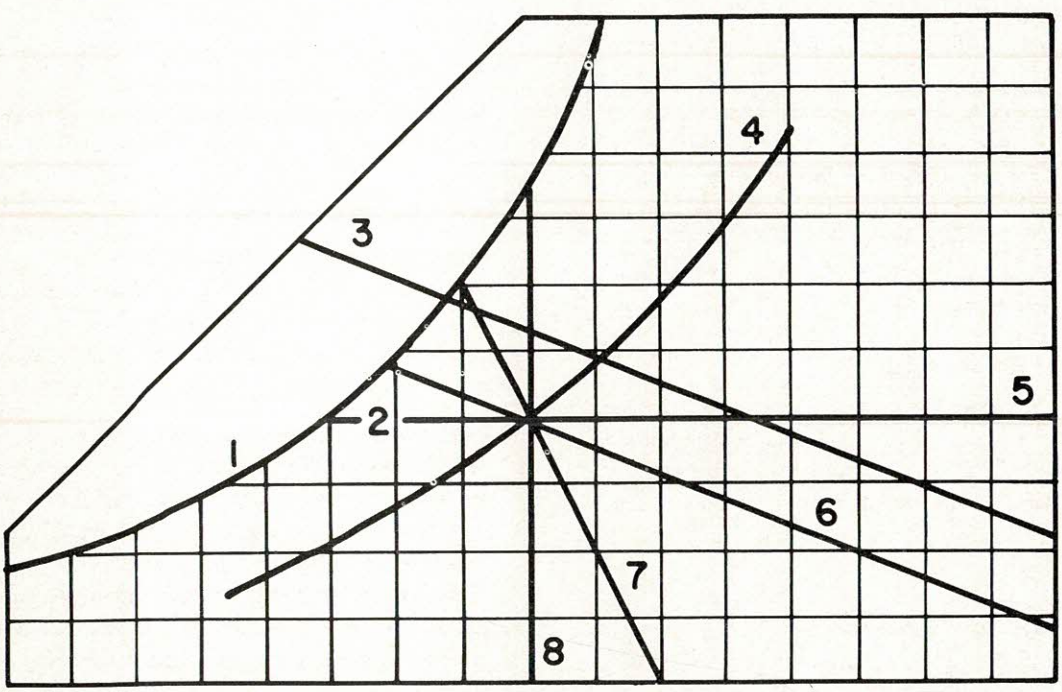


FIG. 22

Skeleton psychrometric chart.

Vapor Pressure The partial pressure exerted by the water vapor contained in the air, measured in inches of mercury.

Humidity The condition of the air with respect to the moisture contained in it in the form of vapor.

Absolute Humidity (vapor density) The weight of water vapor per unit volume of air, expressed in grains per cubic foot of air.

Specific Humidity The weight of water vapor per unit weight of air, expressed in grains per pound of dry air.

Relative Humidity (RH) The ratio expressed in per cent of the actual vapor pressure in the air to the vapor pressure of saturated air at the same temperature.

Specific Volume Volume (of the mixture) in cubic feet per pound

Volume (as used in psychrometry) Cubic feet per pound of dry air in the mixture. (V)

Specific Heat The quantity of heat, BTU, required to raise the temperature of 1 lb. of a substance 1°F. Specific heat at constant pressure is denoted by C_p. For air, $C_p = 0.24$. For water vapor (saturated or superheated steam at ordinary room temperatures, $C_p = 0.45$. For liquid water, $C_p = 1.0$.

Sensible Heat Heat that changes the temperature of a substance when added to or abstracted from it.

Latent Heat Heat that does not affect the temperature but changes the state of a substance when added to or abstracted from it. Specifically, in psychrometry, the latent heat of fusion, water to ice, $h_f = 144$ BTU/lb. The latent heat of evaporation, water to steam, h_{fg}, can be found in standard steam tables.

Total Heat (enthalpy) The summation of sensible and latent heat, in BTU per pound of a substance, between an arbitrary datum point and the temperature and state under consideration (h).

ASHRAE PSYCHROMETRIC CHART NO. 1

NORMAL TEMPERATURE

BAROMETRIC PRESSURE 29.921 INCHES OF MERCURY

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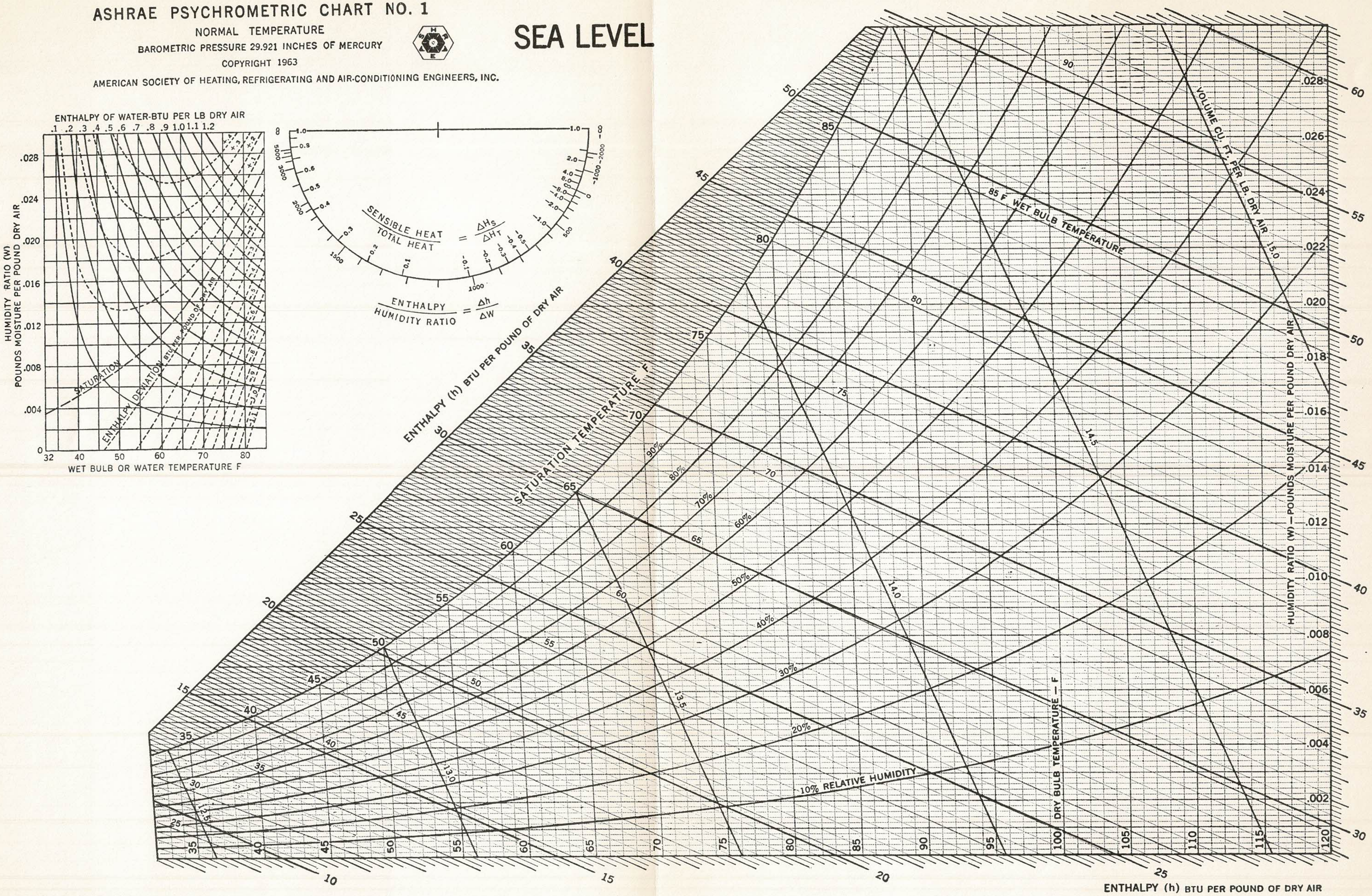
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

SEA LEVEL



CHAPTER 13
ELEMENTARY DUCT DESIGN

Theory of Airflow

Gases (such as air) may be considered equivalent to a "fluid" in which intermolecular space is of such magnitude that cohesive forces in the gas are negligible. Equal volumes of gases at the same temperature and pressure have the same number of molecules; therefore the volume occupied by the molecular weight of all gases is identical. The gaseous state of matter is characterized by indefinite volume and shape. Matter in the gaseous state expands to fill all spaces available. No resistance is offered by the gas itself to change of shape and very little resistance is offered to change in volume. Both gases and vapors tend to expand sufficiently to exert a pressure equilibrium equivalent to its individual molecular or kinetic energy. This kinetic energy is proportional to the absolute temperature; therefore the volume of a gas is constant for a given temperature. If a temperature is maintained constant but the volume is varied the pressure of the gas will be proportional to its density. If the volume of a gas is kept constant and its temperature changed the pressure is proportional to the absolute temperature.

The measurement of quantity of flow of air is generally accomplished either by making observations on the change of state due to flow system configuration, such as the pressure drop across a metering orifice, or by the displacement of some device such as a rotating vane anemometer. The selection of the metering system is usually determined by the type of fluid, by the precision of measurement desired, by the cost of equipment and installation, by the range of flow quantity, by the ease of maintenance, by the method of observation and recording.

Instruments to measure the flow and velocity of air are covered in Section IV of this manual.

Static, Velocity, Total Pressures

Static pressure is the compressive pressure existing in a fluid, such as air and is a measure of its potential energy. Static pressure may exist in a fluid at rest or in motion and is virtually the means of producing flow and maintaining flow against resistance. Static pressure is not sufficient to change materially the volume of air at the ordinary pressures encountered in fan systems. For example, one inch water pressure corresponds to a change of less then one quarter of one percent in the volume of the air.

Velocity pressure is the pressure corresponding to the velocity of flow and is a measure of the kinetic energy in the fluid. Assuming frictionless flow, velocity pressure is equivalent to the pressure producing the flow or the pressure that may be produced by the stoppage of flow.

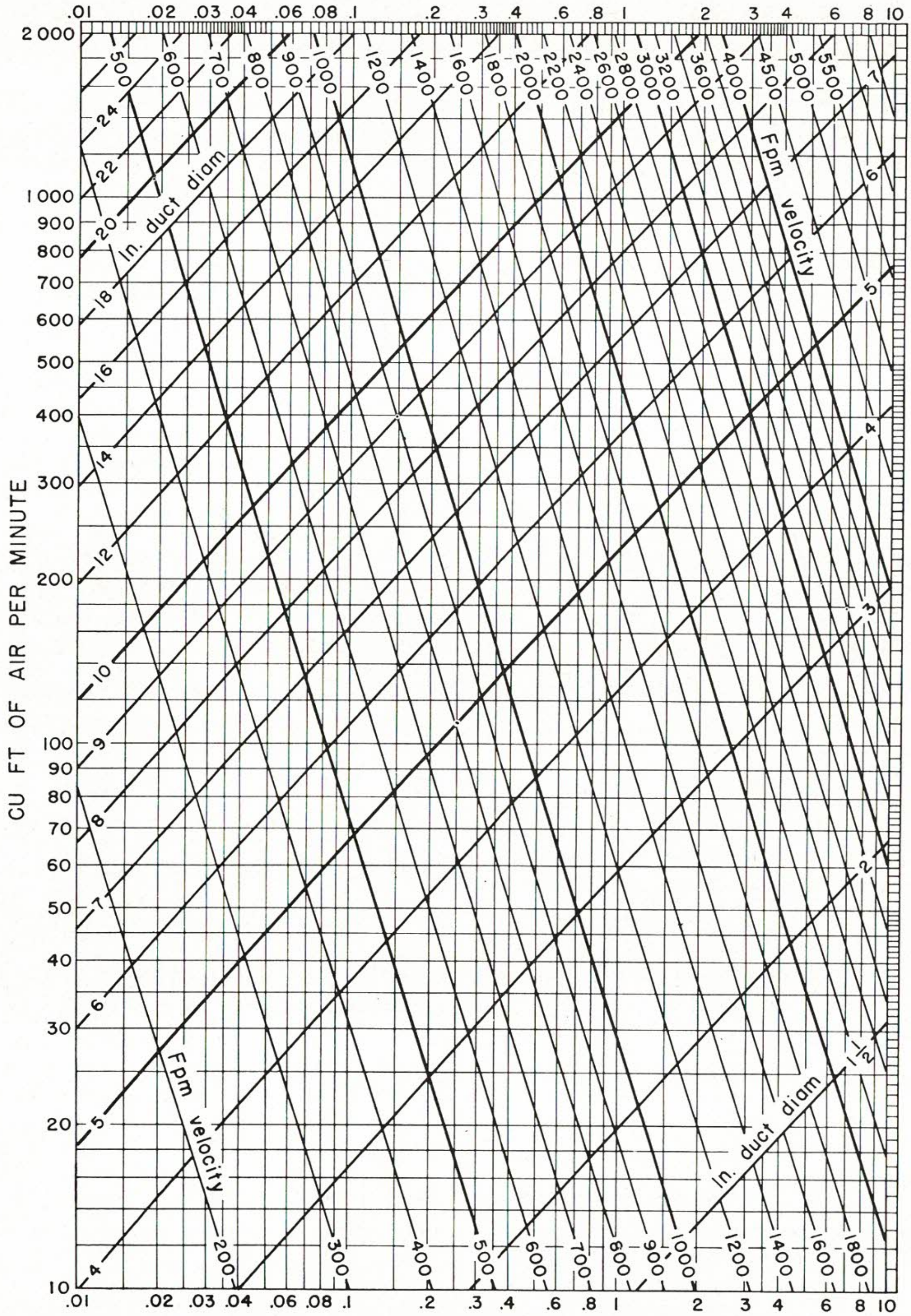
Total pressure is the sum of the static and velocity pressures and is a measure of the total energy in the fluid. Frequently in fan work some of the pressures are negative or less than that of the atmosphere, in which case confusion may be avoided by amending the above to read that the absolute total pressure is the sum of the absolute static pressure and the velocity pressure.

Static pressure may be transformed into velocity pressure and vice versa though always with some loss of efficiency depending on conditions. Thus, the static pressure in a plenum chamber is partly converted into velocity pressure as the air enters the discharge ducts. Again, if the pipe section is gradually enlarged the velocity pressure will be reduced and the static pressure increased. This is generally termed static regain.

Laminar and Turbulent Flow

When air flows through a pipe at low velocities the particles follow predictable paths free from eddys or swirls. The flow is then said to be laminar. As the velocity increases the character of flow changes; eddys form, and the paths of the fluid become sinuous or swirling. This type of flow is known as turbulent flow. Each type of flow has its own laws or resistance to motion. Turbulent flow is not necessarily caused by poor design or workmanship of the duct system but it is a condition always existing where the velocity exceeds certain critical values even though the pipe be of utmost smoothness. When swirls develop we have what is known as "higher" critical velocity. At "lower" critical velocity the eddys die out and the flow becomes laminar.

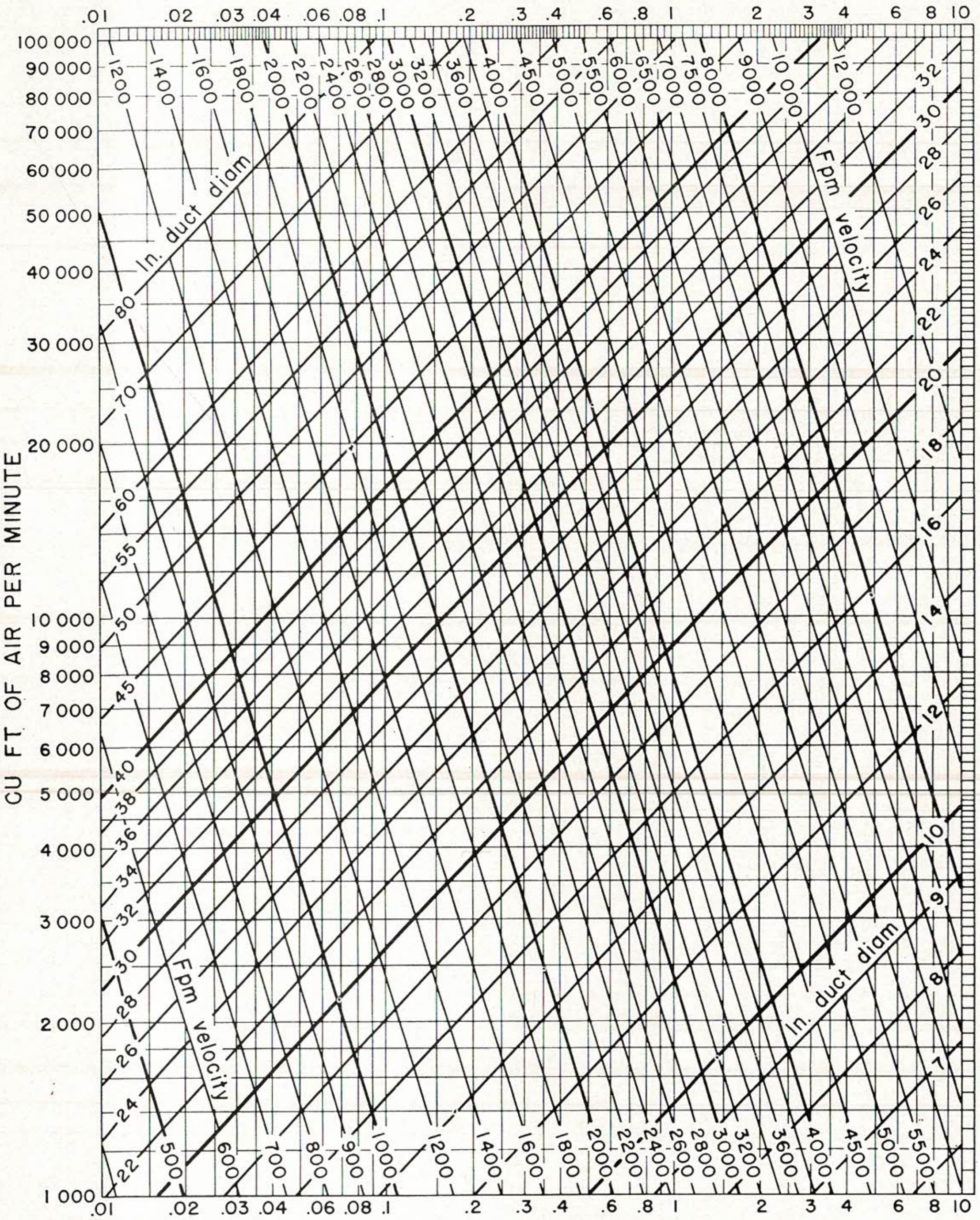
While laminar flow develops less resistance than turbulent flow it can sometimes cause problems of stratification or a failure of different temperatures of air to mix so that air taken from one side of a duct is of a different temperature than that taken from the other. This may be corrected by the installation of mixing baffles.



(Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.) Caution: Do not extrapolate below chart.

FIG. 23

Friction of air in straight ducts for volumes of 10 to 2000 CFM.



(Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.)

FIG. 24

Friction of air in straight ducts for volumes of 1000 to 100,000 CFM.

Side Rectangular Duct	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0
3.0	3.8	4.0	4.2	4.4	4.6	4.8	4.9	5.1	5.2	5.5	5.7	6.0	6.2	6.4	6.6	6.8	7.0
3.5	4.1	4.3	4.6	4.8	5.0	5.2	5.3	5.5	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.4	7.6
4.0	4.4	4.6	4.9	5.1	5.3	5.5	5.7	5.9	6.1	6.4	6.8	7.1	7.3	7.6	7.8	8.1	8.3
4.5	4.6	4.9	5.2	5.4	5.6	5.9	6.1	6.3	6.5	6.9	7.2	7.5	7.8	8.1	8.4	8.6	8.9
5.0	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.7	6.9	7.3	7.6	8.0	8.3	8.6	8.9	9.1	9.4
5.5	5.1	5.4	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.6	8.0	8.4	8.7	9.0	9.4	9.6	9.8

Side Rectangular Duct	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	22	24	26	28	30	Side Rectangular Duct
6	6.6																				6
7	7.1	7.7																			7
8	7.5	8.2	8.8																		8
9	8.0	8.6	9.3	9.9																	9
10	8.4	9.1	9.8	10.4	10.9																10
11	8.8	9.5	10.2	10.8	11.4	12.0															11
12	9.1	9.9	10.7	11.3	11.9	12.5	13.1														12
13	9.5	10.3	11.1	11.8	12.4	13.0	13.6	14.2													13
14	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3												14
15	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4											15
16	10.4	11.4	12.2	13.0	13.7	14.4	15.1	15.7	16.3	16.9	17.5										16
17	10.7	11.7	12.5	13.4	14.1	14.9	15.5	16.1	16.8	17.4	18.0	18.6									17
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.6	17.3	17.9	18.5	19.1	19.7								18
19	11.2	12.2	13.2	14.1	14.9	15.6	16.4	17.1	17.8	18.4	19.0	19.6	20.2	20.8							19
20	11.5	12.5	13.5	14.4	15.2	15.9	16.8	17.5	18.2	18.8	19.5	20.1	20.7	21.3	21.9						20
22	12.0	13.1	14.1	15.0	15.9	16.7	17.6	18.3	19.1	19.7	20.4	21.0	21.7	22.3	22.9	24.1					22
24	12.4	13.6	14.6	15.6	16.6	17.5	18.3	19.1	19.8	20.6	21.3	21.9	22.6	23.2	23.9	25.1	26.2				24
26	12.8	14.1	15.2	16.2	17.2	18.1	19.0	19.8	20.6	21.4	22.1	22.8	23.5	24.1	24.8	26.1	27.2	28.4			26
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.6	24.4	25.0	25.7	27.1	28.2	29.5	30.6		28
30	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9	26.7	28.0	29.3	30.5	31.6	32.8	30
32	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.6	24.4	25.2	26.0	26.7	27.5	28.9	30.1	31.4	32.6	33.8	32
34	14.4	15.7	17.0	18.2	19.3	20.4	21.4	22.4	23.3	24.2	25.1	25.9	26.7	27.5	28.3	29.7	31.0	32.3	33.6	34.8	34
36	14.7	16.1	17.4	18.6	19.8	20.9	21.9	23.0	23.9	24.8	25.8	26.6	27.4	28.3	29.0	30.5	32.0	33.0	34.6	35.8	36
38	15.0	16.4	17.8	19.0	20.3	21.4	22.5	23.5	24.5	25.4	26.4	27.3	28.1	29.0	29.8	31.4	32.8	34.2	35.5	36.7	38
40	15.3	16.8	18.2	19.4	20.7	21.9	23.0	24.0	25.1	26.0	27.0	27.9	28.8	29.7	30.5	32.1	33.6	35.1	36.4	37.6	40
42	15.6	17.1	18.5	19.8	21.1	22.3	23.4	24.5	25.6	26.6	27.6	28.5	29.4	30.4	31.2	32.8	34.4	35.9	37.3	38.6	42
44	15.9	17.5	18.9	20.2	21.5	22.7	23.9	25.0	26.1	27.2	28.2	29.1	30.0	31.0	31.9	33.5	35.2	36.7	38.1	39.5	44
46	16.2	17.8	19.2	20.6	21.9	23.2	24.3	25.5	26.7	27.7	28.7	29.7	30.6	31.6	32.5	34.2	35.9	37.4	38.9	40.3	46
48	16.5	18.1	19.6	20.9	22.3	23.6	24.8	26.0	27.2	28.2	29.2	30.2	31.2	32.2	33.1	34.9	36.6	38.2	39.7	41.2	48
50	16.8	18.4	19.9	21.3	22.7	24.0	25.2	26.4	27.6	28.7	29.8	30.8	31.8	32.8	33.7	35.5	37.3	38.9	40.4	42.0	50
52	17.0	18.7	20.2	21.6	23.1	24.4	25.6	26.8	28.1	29.2	30.3	31.4	32.4	33.4	34.3	36.2	38.0	39.6	41.2	42.8	52
54	17.3	19.0	20.5	22.0	23.4	24.8	26.1	27.3	28.5	29.7	30.8	31.9	32.9	33.9	34.9	36.8	38.7	40.3	42.0	43.6	54
56	17.6	19.3	20.9	22.4	23.8	25.2	26.5	27.7	28.9	30.1	31.2	32.4	33.4	34.5	35.5	37.4	39.3	41.0	42.7	44.3	56
58	17.8	19.5	21.1	22.7	24.2	25.5	26.9	28.2	29.3	30.5	31.7	32.9	33.9	35.0	36.0	38.0	39.8	41.7	43.4	45.0	58
60	18.1	19.8	21.4	23.0	24.5	25.8	27.3	28.7	29.8	31.0	32.2	33.4	34.5	35.5	36.5	38.6	40.4	42.3	44.0	45.8	60
62	18.3	20.1	21.7	23.3	24.8	26.2	27.6	29.0	30.2	31.4	32.6	33.8	35.0	36.0	37.1	39.2	41.0	42.9	44.7	46.5	62
64	18.6	20.3	22.0	23.6	25.2	26.5	27.9	29.3	30.6	31.8	33.1	34.2	35.5	36.5	37.6	39.7	41.6	43.5	45.4	47.2	64
66	18.8	20.6	22.3	23.9	25.5	26.9	28.3	29.7	31.0	32.2	33.5	34.7	35.9	37.0	38.1	40.2	42.2	44.1	46.0	47.8	66
68	19.0	20.8	22.5	24.2	25.8	27.3	28.7	30.1	31.4	32.6	33.9	35.1	36.3	37.5	38.6	40.7	42.8	44.7	46.6	48.4	68
70	19.2	21.	22.8	24.5	26.1	27.6	29.1	30.4	31.8	33.1	34.3	35.6	36.8	37.9	39.1	41.3	43.3	45.3	47.2	49.0	70
72															39.6	41.8	43.8	45.9	47.8	49.7	72
74															40.0	42.3	44.4	46.4	48.4	50.3	74
76															40.5	42.8	44.9	47.0	49.0	50.8	76
78															40.9	43.3	45.5	47.5	49.5	51.5	78
80															41.3	43.8	46.0	48.0	50.1	52.0	80
82															41.8	44.2	46.4	48.6	50.6	52.6	82
84															42.2	44.6	46.9	49.2	51.1	53.2	84
86															42.6	45.0	47.4	49.6	51.6	53.7	86
88															43.0	45.4	47.9	50.1	52.2	54.3	88
90															43.4	45.9	48.3	50.6	52.8	54.8	90
92															43.8	46.3	48.7	51.1	53.4	55.4	92
96															44.6	47.2	49.5	52.0	54.4	56.3	96

Equation for Circular Equivalent of a Rectangular Duct:[5]

$$d_c = 1.30 \frac{(ab)^{0.625}}{(a+b)^{0.250}} = 1.30 \sqrt[8]{\frac{(ab)^5}{(a+b)^2}}$$

where

a = length of one side of rectangular duct, inches.

b = length of adjacent side of rectangular duct, inches.

d_c = circular equivalent of a rectangular duct for equal friction and capacity, inches.

FIG. 25

Circular equivalents of rectangular ducts for equal friction and capacity.

Air Carrying Capacity

The basic formula for the carrying capacity of a duct is:

CFM equals area (in square feet) times velocity (in feet per minute). However, since ducts are frequently sized in terms of inches rather than in feet the formula can be converted to --

CFM equals (area in square inches) over 144 times feet per minute.

$$CFM = \frac{A}{144} \times fpm$$

This leads to three basic formulas of air flow and capacity.

CFM equals area (in square inches) times feet per minute over 144.

$$CFM = \frac{A \times fpm}{144}$$

Feet per minute equals 144 times CFM over area (in square inches).

$$fpm = \frac{144 \times CFM}{A}$$

Area (in square inches) equals 144 times CFM over fpm.

$$A = \frac{CFM \times 144}{fpm}$$

Pressure drop (meaning frictional loss) in a straight duct is caused by surface friction and this friction loss is most readily calculated by means of the air friction charts which appear in the ASHRAE Guide and Data Book. One chart shows friction in straight ducts for volumes of ten to two-thousand CFM (Fig. 23). The other chart shows friction in straight ducts for volumes of one-thousand to one-hundred-thousand CFM (Fig. 24).

Methods of Duct Sizing

In the design of air ducts for ventilation and air conditioning, three methods are usually favored by the engineer-designer. These three methods are: The Velocity Reduction Method; the Equal Friction Method; and the Static Regain System. These three methods vary in complexity and offer different levels of accuracy so the designer usually favors the simplest system which will meet the requirements of the installation.

Velocity Reduction Method

Under this method the engineer selects the velocity at the fan discharge and then designs his duct system for progressively lower velocities in the main at each branch duct. When the selected velocities and the amount of CFM are known, the duct diameters (for conversion to rectangular ducts) can be selected directly from the two friction loss charts of ASHRAE (Figs. 23 and 24) included in the manual. The engineer normally determines the main and branch having the greatest length and therefore the highest resistance when the straight pipe, elbows, and all transitions are included. This longest system, or system with the highest resistance, determines the fan's static pressure required for the supply duct system. The return air system is sized similarly starting with the lowest velocities at the return air intakes and increasing the velocities progressively in the direction of the fan inlet. Dampers are depended upon to balance this system.

Engineers consider the merits of the velocity reduction method to be: (1) duct sizes are determined easily from the friction charts; (2) velocities can be limited to velocities known to be safe from causing noise. The weaknesses of this system are: (1) the proper selection of velocities requires experience and judgment on on the part of the engineer; (2) it is not always possible for the designer to determine by inspection which run probably has the highest resistance.

Equal Friction Method

The idea behind this method is to make the pressure loss per foot of length the same for the entire system. The practice is to select the velocity in the main duct near the fan from the standpoint of noise for the particular application. Since the air flow rate (in CFM) is known, this establishes a value of friction loss per 100 feet of duct which can be determined directly from the two friction charts (Figs. 23 and 24) of ASHRAE. Engineers consider one feature of the equal friction method to be that the method automatically reduces the duct velocities in the direction of air flow and this, in turn, pretty well insures that noise will not be a problem. If there is a limitation to the equal friction method it probably is that such a system is difficult to balance since the equal friction method makes no provision for equalizing pressure drops in branches or for providing the same static pressure behind each air terminal. In the sizing of ducts and branches under this system only the CFM carried determines the duct size.

If the pressure available for the duct work is known, this pressure can be divided by the total equivalent length of the run which apparently has the highest resistance and this, in turn, will determine the design friction loss value per foot. This design friction loss value is then used in connection with the two charts of friction loss (Figs. 23 and 24) of ASHRAE. It is of course necessary to calculate the resistances of all fittings in terms of equivalent length of straight duct. In some instances it may be difficult to determine the actual static resistance of fittings and transitions; hence, some care must be used in estimating the frictional loss of fittings by visual examination.

Static Regain Method

The principle behind the static regain method is to size a duct run so that the increase in static pressure (regain) at each takeoff junction just offsets the pressure loss of the succeeding section of the run. As velocities are reduced a conversion of velocity pressure into static pressure occurs. Many engineers feel that the static regain method provides a means of designing long runs of duct work having numerous takeoffs so that essentially the same static pressure exists at the entrance to each branch run. Also, if supply outlets are connected directly to the run instead of branch ducts then essentially the same static pressure will exist behind each outlet. This, in turn, simplifies the selection of the outlet and the balancing of the system. The initial velocity in the main duct is selected on consideration for noise and pressure loss factors and the branch ducts are sized by the modified-equal-friction method.

Design of High Velocity Ducts

The transmission of air at high velocity is becoming a common practice and high velocity systems are becoming popular. The design of a high velocity duct system involves a compromise between reduction of duct size and the consequent necessity for higher fan horsepower. The same general rules which apply to conventional or low velocity duct systems also

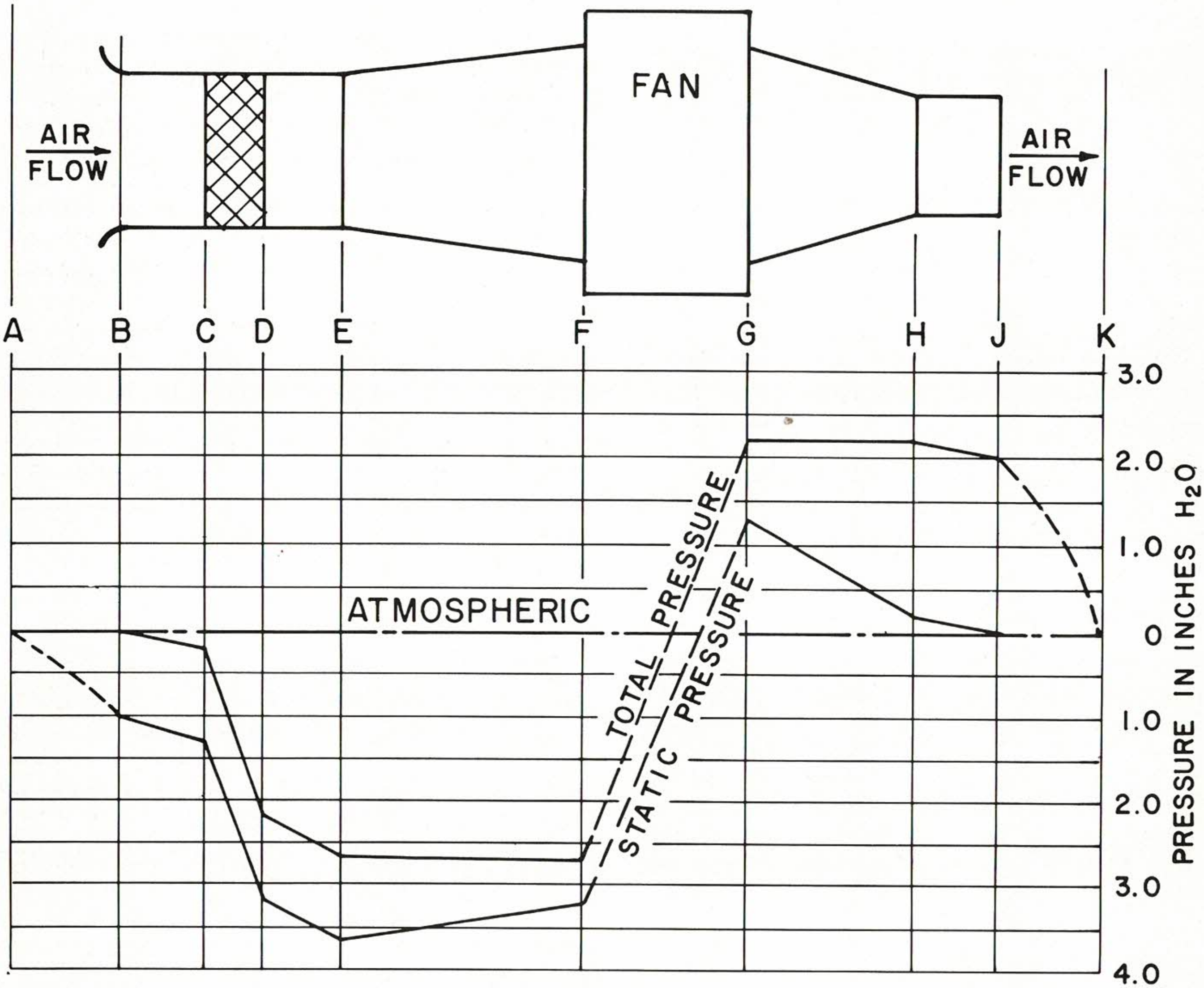


FIG. 26
Diagram of system pressure losses.

apply to high velocity duct systems. In designing high velocity systems attention must be paid to static regain. Latest engineering tests indicate that approximately 85% of the original velocity pressure can be regained. This might easily account for as much as one inch of water static pressure and, accordingly, a method which does not take into consideration static regain in a high velocity system will probably oversize the fans and motors.

Careful consideration must be given to the control of noise. Often attenuators are installed after the fan and acoustical terminal devices, or acoustically lined duct work are called for in several parts of a high velocity system. In present day practice the air velocities in main ducts may vary from 2,500 fpm to over 6,000 fpm and the corresponding maximum branch duct velocities will vary from 2,000 fpm to 4,500 fpm.

Many designers attempt to maintain velocities approximately constant in the main risers and trunks until the friction loss reaches a level of one inch per 100 feet of duct work. As further reductions in duct sizes occur the velocities are selected to maintain a constant friction loss of one inch per 100 feet.

Round ducts are used to a great extent in high velocity systems because they are easier to seal and do not require bracing or stiffening. Where rectangular duct work is used in high velocity systems the duct aspect ratio should be kept within the range of 4.1. The duct must be rigidly braced to withstand the static pressure in the system. SMACNA has published a complete manual of duct construction for high velocity systems and this manual can be used as a guide for adequate reinforcing, bracing, sealing, etc. The SMACNA High Velocity Duct Manual also details fittings and appurtenances such as turning vanes, takeoffs, etc. In high velocity systems operating with variable air flow, some control of static pressures may be required in order to prevent static pressure unbalance, which is designated as a large deviation from design static pressures at the inlet of a terminal.

In single duct high velocity systems operating with changing air volumes the variation in static pressures can be limited by: (1) static pressure controllers operating dampers in the air distribution system; (2) static pressure controllers operating inlet vane dampers on the fan; (3) zoning and changing air supply temperature in response to static pressure changes.

In dual duct systems the variations in heating and cooling loads produce constantly changing demands for cold or warm air and this, in turn, causes a wide variance in the flow and in the duct static pressures. It is usually necessary therefore to control the total fan delivery and in some cases the duct static pressure to limit pressures at the terminals. The three methods commonly used to control static pressures in dual duct systems are: (1) by the use of dampers operated by static pressure regulators located at critical points in the air distributing system; (2) by static pressure controllers regulating cold and warm air temperatures in order to limit the variations in the airflow in individual ducts; (3) by volume regulators in each individual air mixing valve or acoustic terminal device.

CHAPTER 14

ELECTRICITY

The air balancer requires a sufficient knowledge of electricity and electrical circuitry to determine that the brake horsepower being applied to air handling equipment and that the motor is properly connected and protected.

A few simple formulas should be kept in mind when dealing with electricity. The first is a derivative of Ohm's law. "The current in a circuit is equal to the electro-motive-force activity in the circuit divided by the resistance in the circuit." By formula this is:

$$\text{AMPERES} = \frac{\text{VOLTS}}{\text{OHMS}}$$

The second formula involves power, stated in Watts This is:

$$\text{WATTS} = \text{AMPERES} \times \text{VOLTS}$$

Of course, the kilowatt is 1000 watts and horsepower measured electrically is 746 watts. One watt-hour equals 3.412 BTU.

In the United States, we have a variety of electrical services. Some direct current is still produced as are various two phase systems and alternating current systems varying from 20 to 60 cycles. However, with the interlocking of networks and as a result of manufacturers' pressure to standardize, most power companies are converting their services to 3 phase, 60 cycle, alternating current. We will, therefore, concern ourselves only with single phase and three phase, 60 cycle systems.

The word "phase" comes from the fact that such current is produced by a generator having more than one set of armature coils; thus producing more than one electro-motive-force. In a 3 phase system there are 3 sets of such armatures. Cycles represent a pulse count of the electrical "waves" within a phase. 60 cycles implies that there are 60 such "waves" a second. Cycles are a factor in determining the speeds of induction motors which are in increments of 60, depending on the number of poles, less a "slip" factor.

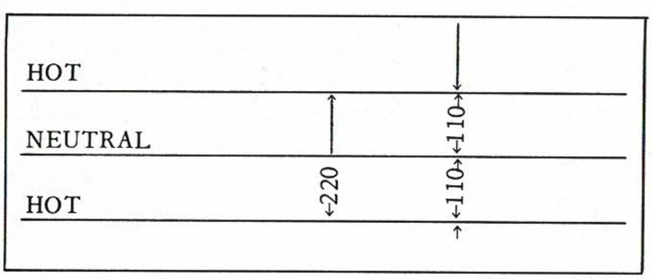


FIG. 27

Voltage relationship for single phase, three wire system.

The Edison single phase, 3 wire system is commonly used to bring power to most homes and small places of business. This consists of two "hot" wires and a neutral. The voltage relationship is shown in Fig. 27. The voltage across the "hot" lines is 220 to 440 while the voltage from "hot" to neutral is 110 to 220.

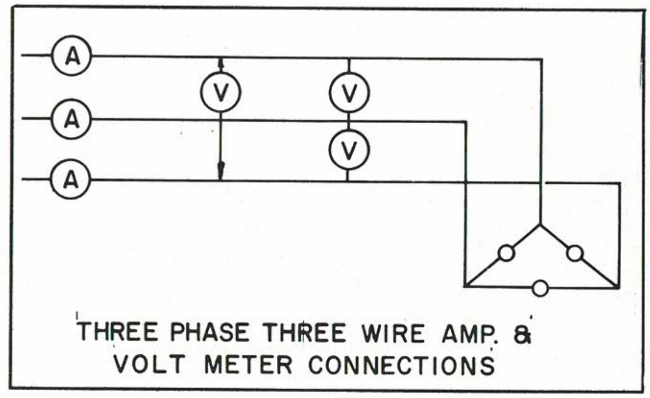


FIG. 28

Wiring diagram for three phase, three wire connections.

The three phase, 3 wire system (Fig. 28) differs from the single phase system in that each line is a "hot" line and the voltages across any two lines are the same.

In either single phase or three phase applications the amperage can be read by installing an ammeter in series in each load connection. Or, a clamp around ammeter may be used on each load connection. For practical purposes, the two or three readings may be averaged. Voltage readings are made by connecting a voltmeter across the lines parallel to the load.

Voltmeters and ammeters are further described in Section IV of this manual.

Since the rotor of an induction motor follows a rotating field, the motor will reverse if the field is made to rotate in the opposite direction. When starting systems, it is advisable to "bump" (apply power for a brief time) the motor to observe rotation. If it is not in the proper rotation, interchange any two leads of the stator winding of a three phase motor.

If a single phase motor is not in proper rotation, it is advisable to study the wiring connection directions that are either adhered to the junction box cover or are loosely inserted in the junction box, or are attached to the motor like an additional nameplate. Frequently there is more than one wire at the motor to be connected to each of the power leads. The directions generally advise which connections to modify to change rotation.

Every electric motor generates heat as well as power. The more inefficient the motor, the greater the amount of heat produced. Unless this heat is dissipated, the temperature within the motor will rise until the insulation is destroyed.

The amount of heat produced in the motor also depends upon the load. Therefore, most motors are rated on the basis of a certain temperature rise when running fully loaded. Open motors are generally designed to run at a temperature rise of 40 degrees C (72 degrees F) above the surrounding (ambient) air temperature. In other words, an open motor when running at full nameplate conditions, should be running at a temperature of 142 degrees if the surrounding air is 70F. Totally enclosed motors are generally rated at 55 degrees C (99 degrees F) temperature rise. Under the latter conditions, a totally enclosed motor would run at 169F.

Protection is given to motors in many motor starting devices by "heaters" or overload protectors. These operate somewhat as fuses by interrupting the circuit if their rated amperage is exceeded over a predetermined period. Consult the motor control manufacturer for "heater" selections for each motor and see that the proper "heaters" are installed in all controllers. If a motor stops because of overload it usually is indicative of an improperly sized heater element; a faulty motor condition; or improperly operating or designed system. Single phase, manual switches usually have one such "heater." Three phase starters usually have two or three "heaters."

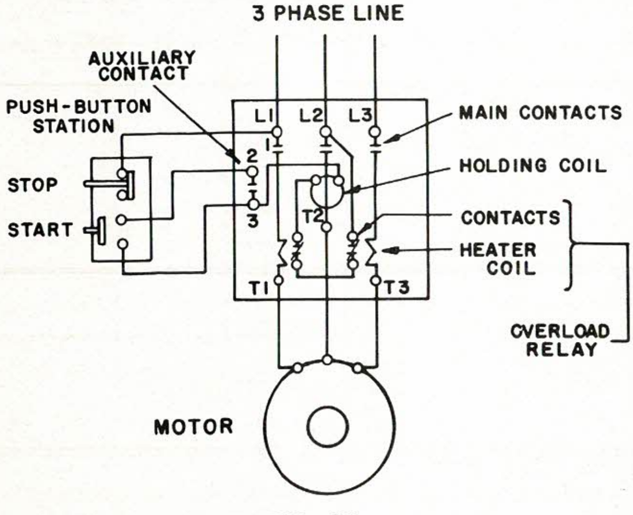


FIG. 29
Three-wire magnetic starter control.

The manual starter is a simple device that operates similar to a toggle switch and may contain one, two or three poles. Unless relays are used in this electrical circuit, this starter does not lend itself to automatic operation.

A magnetic starter consists of contacts for single or polyphase operation that electrically "make" a circuit when pulled into place by an energized electric magnet. This magnet or holding coil may be energized by means of a pushbutton on the face of the starter; by remote pushbuttons; by pneumatic electric switches in the temperature control system; by auxiliary contacts in another motor controller or by a variety of other means. (See Fig. 29) The holding coil normally gets its current from connections from the line side of the starter. The coil is usually rated at the same voltage as the current being applied to the motor. In this case all remote control circuitry must be of the same voltage as the motors. If it is necessary or desirable to operate controls at a lesser or different voltage than the motors, then the holding coil must be selected for such control voltage. The different voltage must come from a separate source or from a transformer built into the motor controller.

There is another safety device put into circuits to protect motors. This is a low voltage protector. Undervoltage will cause an increase in amperage which in turn can damage the motor and cause trouble in the electrical system. Variations of over or under 10% of rated voltage should be promptly called to the attention of the design engineer through the proper channels.

Electrical systems are protected by fuses or circuit breakers. These are usually combined with manual disconnecting devices and may be in separate enclosures or within the same enclosure as the motor control. Do not operate motors by "jumping" fuses.

As pointed out in the preliminary procedure, the balancer should check all motor starting equipment before starting a system. When the system is operating, following the balancing procedure, the balancer should take amperage and voltage readings to determine the brake horsepower of the motor. The following formula applies:

$$\text{bhp} = \frac{\sqrt{3}\ (\text{Volts x Amps x Power Factor x } E_{ff}}{746}$$

However, as we can usually only guess at the power factor and efficiency it may be desirable to calculate the bhp by plotting. This can be done as follows:

(2) Prepare a chart (Fig. 30) with horsepower at about 125% of rated horsepower, in even increments in the horizontal base line. On the vertical base line, plot the amps in even increments to about 110 to 120% of the rated full load amps.

(b) Disconnect the motor from the fan and read the amps when running at no load. Mark this reading at point "a" on the vertical base line. Divide this no load amperage reading by two and mark at point "b" on the vertical base line.

(c) Read the motor nameplate for the full load amps and project this from the vertical base line across the chart. Read the rated horsepower and project this from the horizontal base line across the chart until it intersects the full load amp line. Call this point "c". Draw a line from point "b" to point "c".

(d) Divide the rated horsepower by two and project this value from the horizontal base line until it intersects line "b" to "c". Call this intersection point "d". Draw a smooth curve through points "a", "d", and "e".

(e) Determine the fan hp by plotting the running amps to points of intersection on the curve "a-d-c".

This method is more accurate and useful than the popular method of dividing the running amps by the rated full load amps and multiplying the result by the rated horsepower.

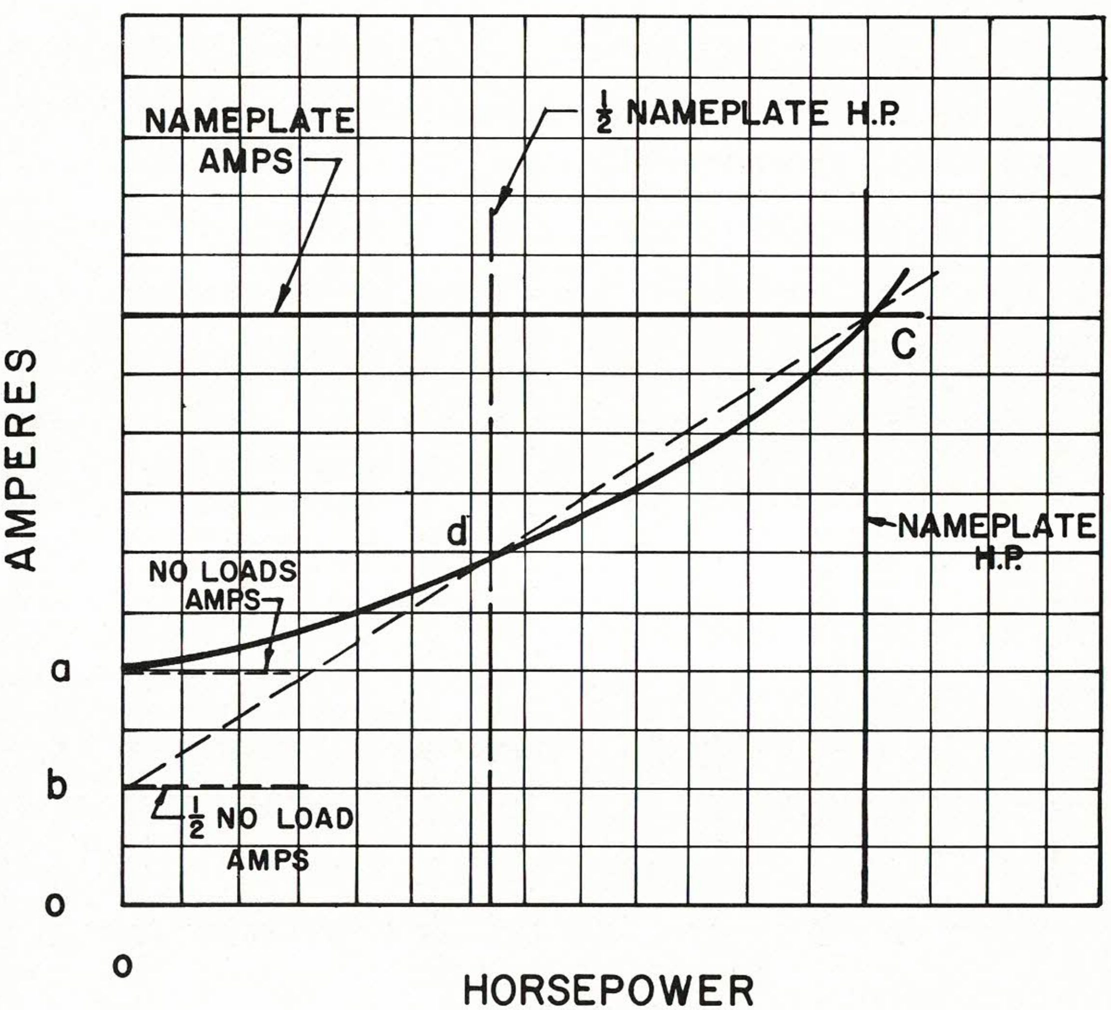


FIG. 30

Plotted chart to determine the brake horsepower of the motor.

CHAPTER 15

FAN AND FAN LAWS

Fans used in heating, ventilating and air conditioning work are divided into two classes:

1. Axial flow fans which consist of tubeaxial, vane axial, and propeller fans.

2. Centrifugal fans:

Forward Curved Blade Centrifugal Fans:

This fan operates at relatively low wheel peripheral velocities. The speed of the air leaving the wheel is greater than the wheel peripheral speed. Forward-curved fans are very dependent on the discharge duct from the fan for static pressure recovery. It requires a long straight duct to slow down the air and build static pressure, so the duct in reality is an extension of the fan casing. The fan blades are sloped in the direction of rotation which gives this fan its designation.

Radial Blade Centrifugal Fans:

This fan provides air speeds about equal to wheel peripheral velocities. This fan has a higher peak efficiency than the forward curved fan and it is not as dependent on the discharge duct for static regain.

Backward Curved Blade Centrifugal Fans:

This fan provides air speeds lower than the wheel peripheral velocity. The efficiency of this type fan is the highest and it does not depend on the discharge duct for static recovery. The backwardly inclined fan has a steeper pressure curve and variations in system pressure will result in a smaller variation in air volume. In its best operating area, to the right of the pressure peak, the horsepower curve is almost flat thus giving a non-overloading effect. (See Fan Curves for Typical Forward and Backward Curved Fans – Fig. 31 and Fig. 32)

A properly designed "Air-Foil" fan adds two advantages to the conventional backward curved fan. It is generally more quiet and efficient in operation.

The following table illustrates the relative character-

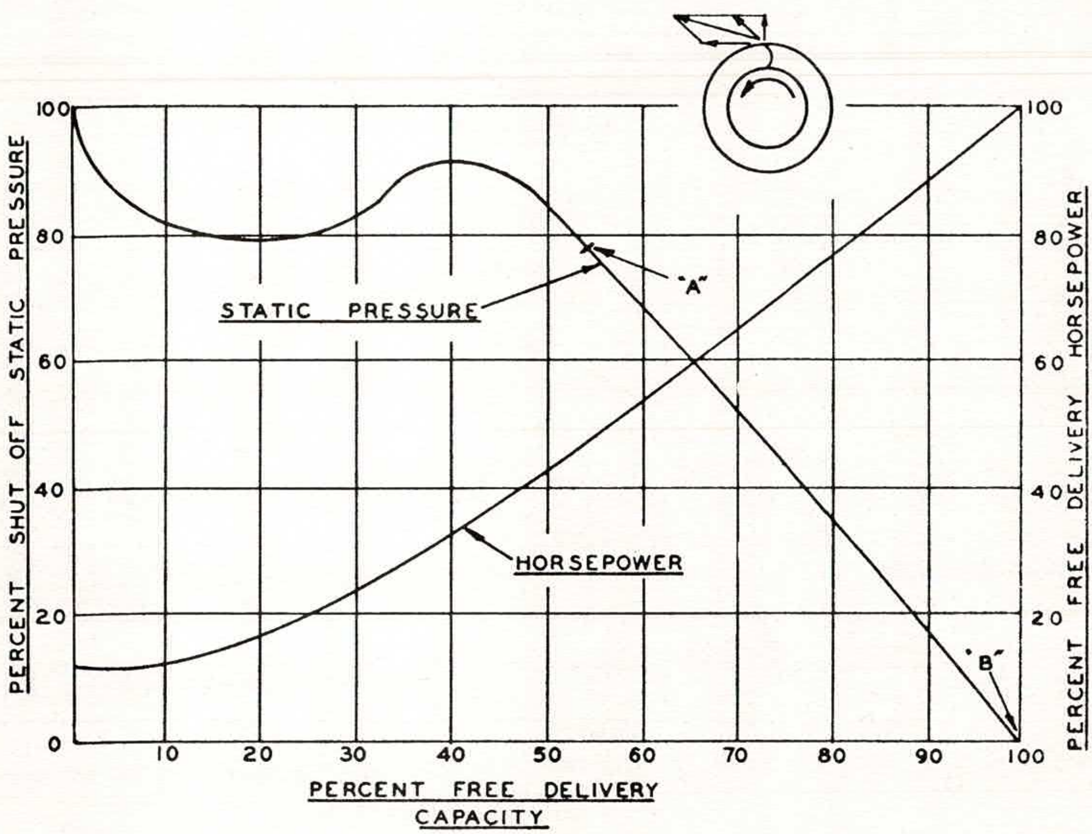


FIG. 31

Performance curve for a forward curved fan blade.

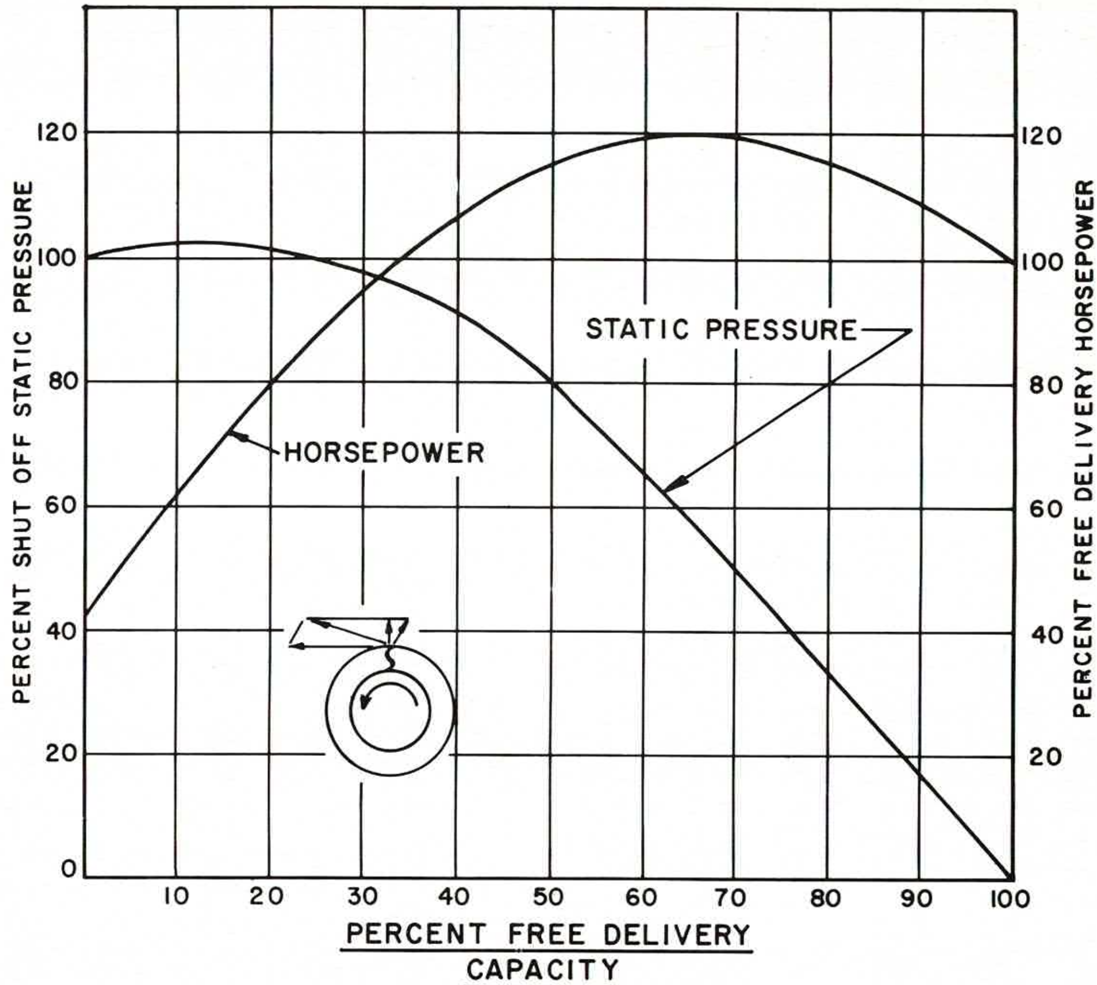


FIG. 32

Performance curve for a backward curved fan blade.

istics of the above mentioned fan types to produce a given CFM against a given resistance. Pressures are measured at fan discharge:

	Velocity Pressure	Static Pressure	Wheel Speed
Forward Curved	80%	20%	Lowest
Radial Blade	55%	45%	Medium
Backward Curved	30%	70%	Highest

Fan Law No. 1 – Relation of Volume To Speed:

"The amount of air delivered by a fan will vary in direct proportion to the fan speed."

(a) $\text{CFM desired} = \dfrac{\text{rpm desired} \times \text{CFM measured}}{\text{rpm measured}}$

(b) $\text{rpm desired} = \dfrac{\text{CFM desired} \times \text{rpm measured}}{\text{CFM measured}}$

To determine pulley sizes if fan speed must be changed, use the following ratio:

$$\frac{\text{rpm fan}}{\text{rpm motor}} = \frac{\text{Pitch Diameter of Motor Pulley}}{\text{Pitch Diameter of Fan Pulley}}$$

Fan Law No. 2 – Relation of Pressure Required To Speed:

"The resistance of an air handling system varies as as the square of the fan speed."

$$SP_{Required} = SP_{Measured} \times \left(\frac{\text{rpm Required}}{\text{rpm Measured}}\right)^2$$

Fan Law No. 3 – Relation of Horsepower To Speed:

"The power required to operate a fan varies as the cube of the fan speed."

$$hp_{Required} = hp_{Measured} \times \left(\frac{\text{rpm Required}}{\text{rpm Measured}}\right)^3$$

Since static pressure is a result of the air volume being pushed through the system, it is extremely important to study the effect of system design on the fan. Fan performance can be radically influenced by air load distribution due to inlet design and by inlet and outlet dampers among many other factors.

When field testing fans it probably will be found that the CFM indicated by pitot tube tests will not match the CFM determined by bhp. Nor will either be the same as that shown by static pressure. If these are close, they probably only indicate test inaccuracy and may be averaged to determine performance. If the difference is great, then there may be a system design factor affecting the fan performance and this will have to be analyzed in order to make a correction.

Drives:

Regardless of whether drives consist of stock or special items there are certain primary conditions to consider with respect to the design of satisfactory drives. Those most commonly encountered are:

(a) Drives should be installed with provisions for center distance adjustment. This is essential, as all belts stretch.

(b) Centers should not exceed 2½ to 3 times the sum of the sheave diameters nor be less than the diameter of the larger sheave.

(c) Arc of contact on the smaller sheave should not be less than 120°.

(d) Ratios should not exceed 8:1.

(e) Belt speed preferably should not exceed 5000 ft./min., nor be less than 1000 ft./min. Best practice is about 4000 ft./min.

(f) Sheaves should be dynamically balanced for speeds in excess of 5000 ft./min. rim speed.

When installing drives these points should be particularly watched:

(a) Be sure that shafts are parallel and sheaves are in proper alignment. Check again after a few hours of operation.

(b) Do not drive sheaves on or off shafts. Wipe shaft, key and bore clean with oil. Tighten screws carefully. Recheck and re-tighten after a few hours of operation.

(c) Belts should never be forced over sheaves.

(d) In mounting belts be sure the slack in each belt is on the same side of the drive. This should be the slack side of the drive.

(e) Belt tension should be reasonable. When in operation, the tight side of the belts should be in a straight line from sheave to sheave and with a slight bow on the slack side. All drives should be inspected periodically to be sure belts are under proper tension and are not slipping.

(f) When making replacements of belts on a drive, be sure to replace the entire set with a new set of matched belts.

CHAPTER 16

GRILLES AND REGISTERS*

Room Air Distribution

Since the human body loses heat to the environment by radiation, conduction, convection, and evaporation, the space factors of mean radiant temperature, dry bulb temperature, relative humidity and air movement are important to the occupants' comfort. The function of an air conditioning system, primarily through its room air supply and air distribution system, is to control the above factors so that the occupants' heat loss is maintained at a comfortable rate. (Fig. 33)

It is known that the body heat loss by radiation can be partially regulated by control of the dry-bulb temperature. Body radiation losses may also be regulated by controlling the mean radiant temperature with heated and cooled panels.

Considering that a "draft" is any local feeling of coolness, it has been determined that a velocity change of 15 fpm produces about the same effect on comfort as a one degree temperature change. Eighty per cent of the people will accept a local air velocity of 80 fpm in the neck region when the local air temperature equals the ambient temperature. However, when the local air temperature is 2 degrees below the ambient, these same people prefer the air velocity to be 40 fpm or less. In addition, people will tolerate air temperatures about 4 degrees lower at the ankle region than at the neck region.

In order to satisfy 80 percent of the occupants, the room air distribution system should provide local room air velocities less than 40 to 80 fpm in the occupied area. In addition, the local air temperatures should be less than 2 degrees below the ambient and the air temperatures at the ankle region should be less than 4 degrees below the air temperatures at the neck region.

During heating, local air velocities generally are below 40 fpm, and therefore, the air distribution should be planned to limit the temperature gradient from the ankle to the neck region to less than 4 degrees.

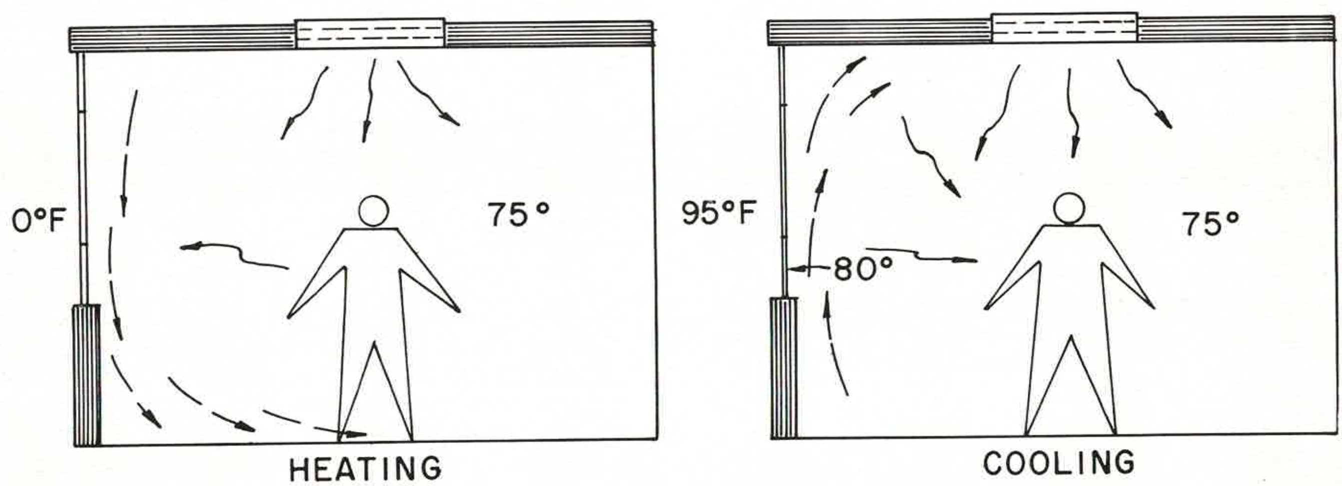


FIG. 33

Some elements affecting the body heat loss.

During cooling, local air temperatures generally are not more than 1 to 2 degrees below the ambient, and therefore the air distribution should be planned to limit the local air velocities within the range of 50 to 80 fpm.

General Principles of Room Air Distribution

The natural convection currents, down the glass during heating and up the glass during cooling, as shown in Fig. 33, are a major influence on the air distribution in the perimeter zones of a building.

During heating, these currents carry cool air down to the floor level causing a stratification of air in layers of increasing temperatures from the floor to the ceiling. The severity of the temperature gradient depends on the outdoor temperature, the construction, and the air distribution. It is easily understood that warm

**Reprinted by permission – Refrigeration Service Engineers Society* supply air introduced at the base of the wall would tend to counteract these currents and reduce or eliminate the stratification. As a matter of fact, optimum air distribution in perimeter zones requires perimeter introduction of air or supplementary radiation at the perimeter.

During cooling, the currents carry the warm air up the wall to the ceiling level. Stratification then forms from the ceiling down. To eliminate this stratification, cool air should be projected into this region near the ceiling. To do this most effectively, the supply air outlets should be located high in the wall or in the wall or in the ceiling. (See Fig. 34)

Performance of the Supply Air Outlet

Extensive studies of supply performance have shown that throw from free round openings, grilles, perforated panels and ceiling diffusers, are related to the average velocity at the face of the supply outlet or opening.

A jet discharged from a free opening has four zones of expansion and the centerline velocity of the jet in any zone is related to the initial velocity as shown in Fig. 35. Regardless of the type of outlet, the air stream will tend to assume a circular shape in free space. The effects of surfaces will be considered later.

Zone III is the most important as far as room air distribution is concerned and the relation between initial velocity and jet centerline velocity is given by the equation:

$$V_x = V_o K \frac{\sqrt{A_o}}{X} = \frac{KQ_o}{X \sqrt{A_o}} \qquad (1)$$

where:

V_x = Centerline velocity, fpm (Also designated terminal velocity)

V_o = Initial velocity, fpm

A = Area at which initial velocity was measured, sq. ft.

X = Distance from outlet to point of measurement of V_x, ft.

K = Proportionality constant.

Q_o = Supply Outlet Flow Rate, CFM

This equation refers strictly to isothermal free jets. However, with the proper A and K values, it also defines the throw for any type of outlet (Isothermal refers to the supply air being at the same temperature as the room air.) Values of K vary from 6, for free openings to 5, for grilles and 4, for perforated panels. The important point is that the performance of any supply outlet is related to the initial velocity and initial area as shown by the equation and Fig. 35.

Beyond the second zone, the jet is a mixture of supply and room air. The jet expands because of induction of room air. The ratio of the total volume of the jet, to the initial volume of the jet at any distance from the point of origin depends mainly on the ratio of the initial velocity (V_o) to the terminal velocity (V_x). For example, doubling the initial velocity for the same terminal velocity doubles the induction ratio and also the throw as shown by Equation 1.

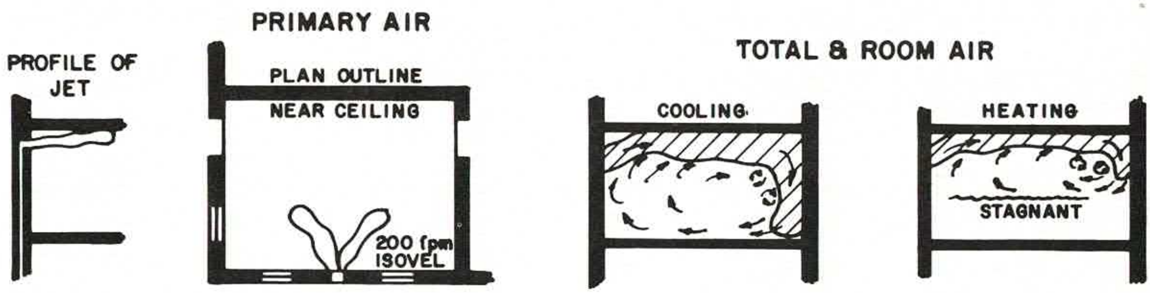
In Zone IV where the terminal velocity is relatively low and specifically for terminal velocities of 50 fpm, the throw, as shown by Equation 1, should be reduced 20%.

The buoyant forces with non-isothermal jets cause the jet to rise during heating and drop during cooling. These conditions result in shorter throws than shown by Equation 1 when the throw is referred to a terminal velocity less than 150 fpm.

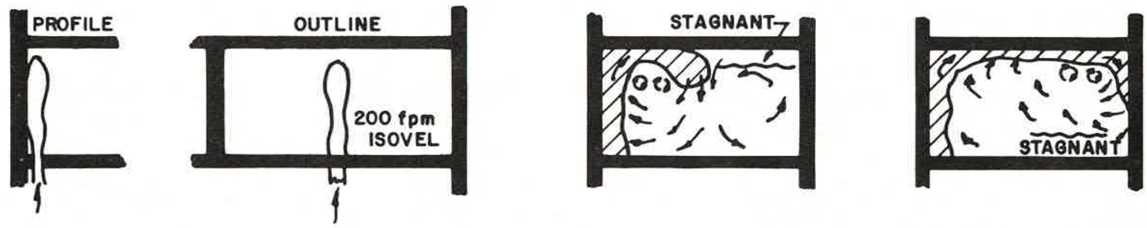
The discussion of throw and drop has been limited to free space applications. If the jet is projected parallel to and within a few inches of a surface, the jet performance will be affected by the surface, which limits the induction on the surface side of the jet. This creates a low pressure region between the jet and the surface which draws the jet toward the surface. In fact, this effect will prevail if the angle of discharge between the jet and the surface is less than 40°. The surface effect will draw the jet from a sidewall outlet to the ceiling if the outlet is within one foot of the ceiling. The jet from a floor outlet will be drawn to the wall and the jet from a ceiling outlet will be drawn to the wall and the jet from a ceiling outlet will be drawn to the ceiling. Surface effect increases the throw for all types of outlets and decreases the drop for horizontally projected air streams. (See Fig. 34)

The typical performance data for sidewall grilles given in Fig. 36 and 37 show most of the factors which affect the airstream from a supply outlet. Fig. 36 shows typical spread characteristics with three deflection settings. With a 0° deflection, the air stream spreads at nearly at 22° angle to boundaries of low velocities, less than 50 fpm. A core of higher velocities, however, does not display this same spread characteristic.

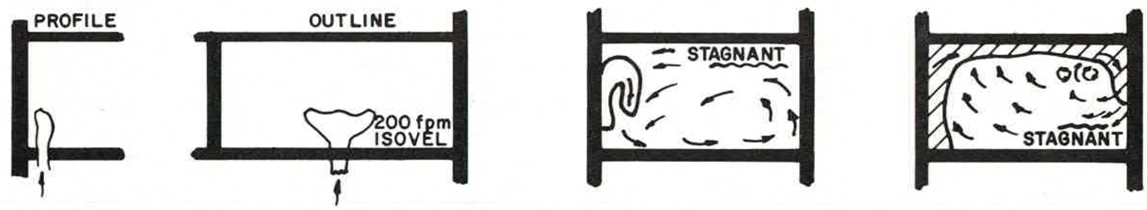
As might be expected, the 22-½° and 45° deflection settings show a greater spread with the higher velocity core remaining close to the low velocity boundaries. With most modern grilles it is possible to adjust to any pattern between the 0° and 45° deflection setting.



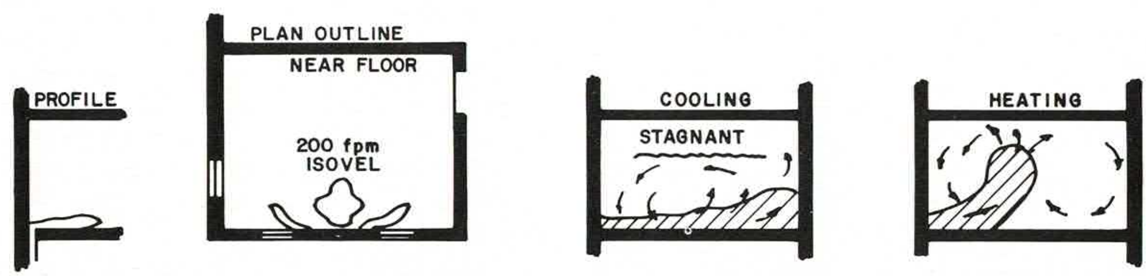
GROUP A, OUTLET NEAR CEILING, HORIZONTAL DISCHARGE, HIGH



GROUP B, OUTLET IN OR NEAR FLOOR, NON-SPREADING VERTICAL JET



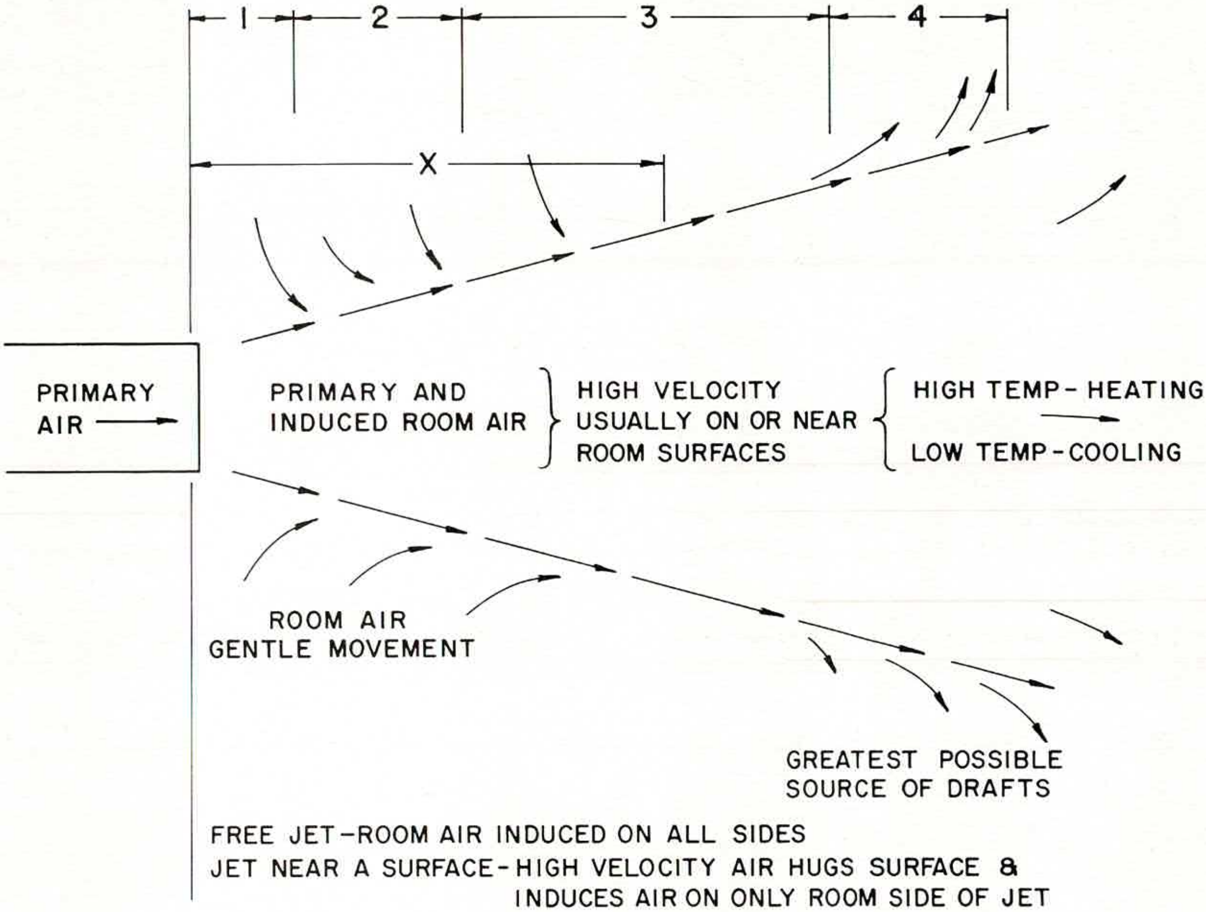
GROUP C, OUTLET IN OR NEAR FLOOR, SPREADING VERTICAL JET



GROUP D, OUTLET NEAR FLOOR, HORIZONTAL DISCHARGE, LOW

FIG. 34

Air motion characteristics of four groups of outlets.



ZONE	SUPPLY VELOCITY, V_O JET CENTERLINE VELOCITY, V_X
1	$V_X = V_O$
2	$V_X \approx V_O / \sqrt{X}$
3	$V_X \approx V_O / x$
4	V_X APPROACHES ROOM VELOCITY

FIG. 35

Four zones in jet expansion.

Fig. 37 shows the relation of the throw and drop for various deflection settings and applications with a 20° F cooling temperature differential. On each chart the intersection of the same airflow (CFM) and outlet velocity (fpm) indicates the 50 fpm terminal point of the air stream from the same size outlet with the various applications and deflection settings.

Comparison of the circled terminal end points with the outlet handling 300 CFM and an outlet velocity of 600 fpm shows:

1. Surface effect, due to the ceiling, increases the throw and reduces the drop over the free space conditions.

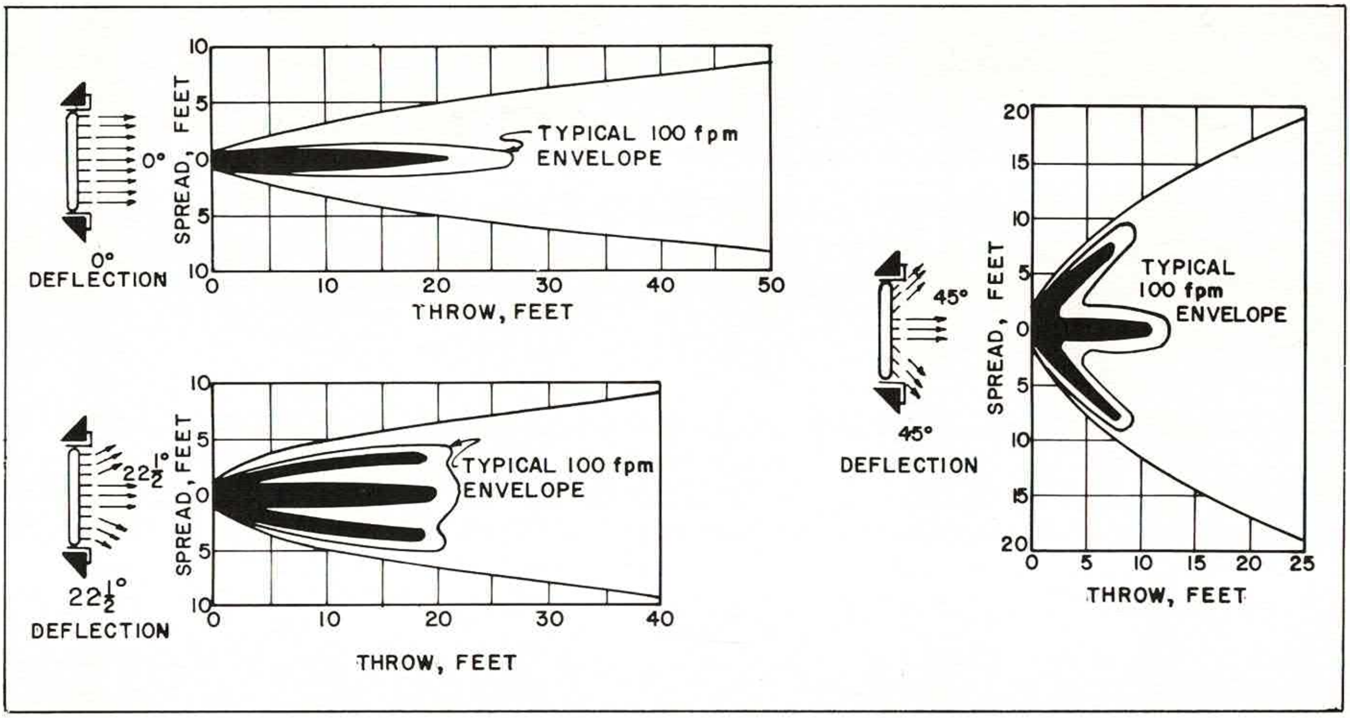


FIG. 36

Typical spread characteristics for sidewall grilles with various deflection settings.

2. More surface effect can be obtained by mounting the outlet below the ceiling and directing the air stream up to the ceiling. When this is done, the air stream spreads out over the ceiling after the impact.

3. A similar increased ceiling effect is obtained by spreading the air as with the 45° horizontal setting.

4. Spreading the air stream reduces the throw and the drop.

5. Drop depends primarily on the quantity of air and only partially on the outlet size or velocity. When drop is a factor, more outlets with less air per outlet are in order.

6. A short throw and minimum drop can be obtained only by using small quantities of air and by spreading the air.

7. The air stream from the outlet tends to "hug" the surface. As a matter of fact this characteristic is almost essential for good comfort air conditioning. Therefore, rather than trying to direct the air away from surfaces, the surfaces should be used intentionally. Note that where the surfaces are used most effectively, the high velocity portions of the air stream have less tendency to enter the occupied zone of the space.

How Room Air Motion Is Related To Outlet Performance

The room air near the supply air stream is entrained by the air stream and, in turn, is replaced by other room air. The room air always moves toward the supply air. Studies in test rooms with fixed dimensions have shown that it is possible to relate average velocity at selected locations within the occupied zone to the number of air changes per hour. However, these relations may not hold for other room configurations. Where average room air motion of less than 50 fpm is required, natural convection and buoyancy effects will predominate over the motion produced by the supply outlet. The only general statement that can be made regarding room air motion and the number of air changes is that 8 to 10 air changes per hour are required to prevent formation of stagnant regions (air motion less than 15 fpm). This is not to imply that stagnant regions represent a serious condition. After all, reducing the temperature one degree is equivalent to a 15 fpm change.

In special rooms, such as clean rooms for precision assembly, an average air motion of 100 fpm may be maintained by supplying air at one side of the room and exhausting it at the other side. The room cross-section then becomes a cross-section of a duct. Here

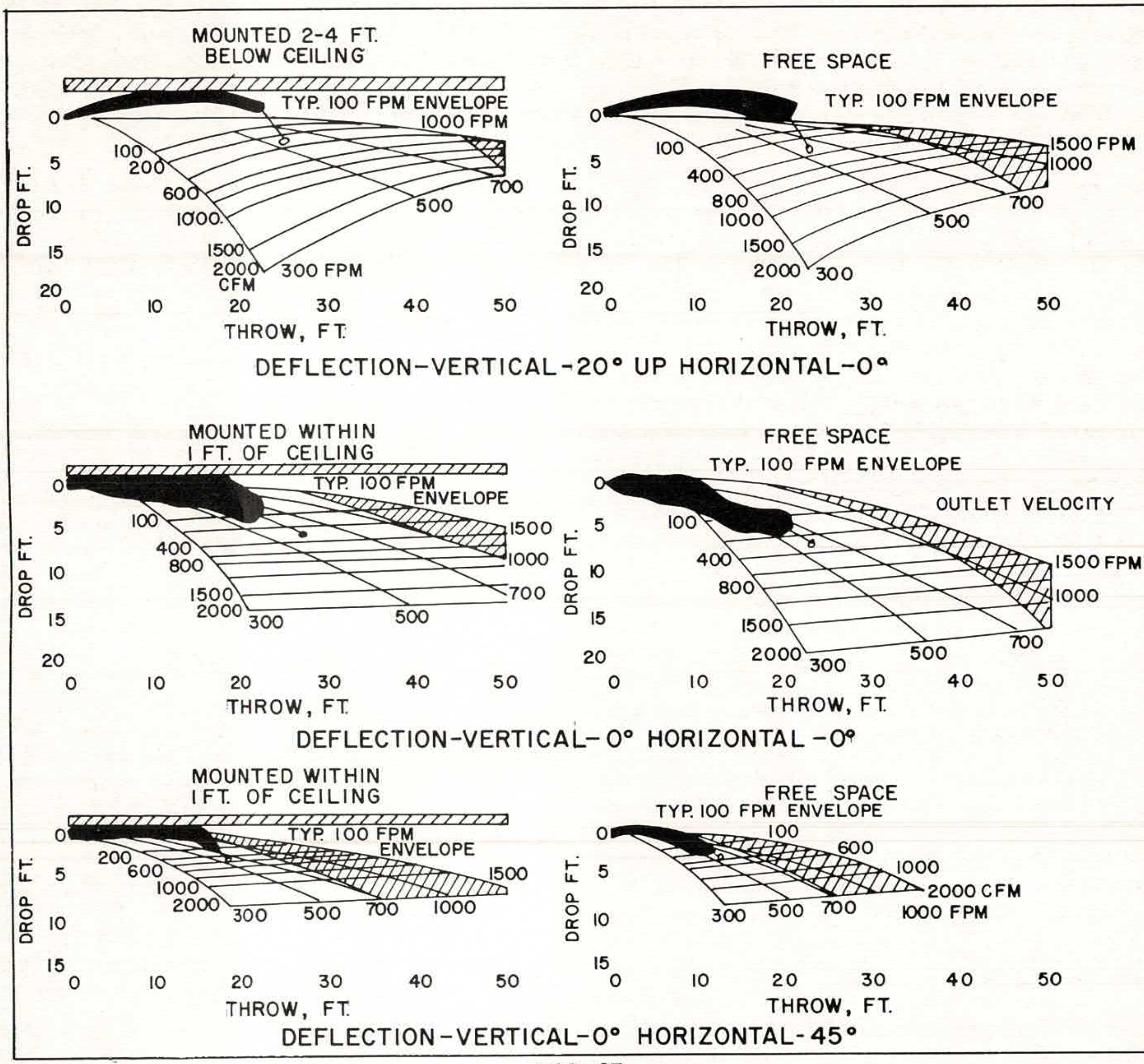


FIG. 37

NOTES:

1. The data above shows the characteristics at a terminal velocity of 50 Fpm for cooling with a 20° differential between the primary air and room air temperatures. During heating, drop is of little concern. In fact, with a free space application, the primary air rises.

2. The characteristics of a single size outlet are shown at the same flow rate and velocity in each chart since all supply velocities are referred to a zero degree deflection setting.

3. The far right section on each chart represents conditions which result in excessive noise levels for application in homes, apartments and churches. Conditions in these regions should only be used in relatively noisy atmospheres, such as restaurants, coliseums, and factories.

4. The core area of a grille selected from above is: CORE AREA, Sq. Ft. = $\frac{\text{Cfm}}{0.82 \text{ fpm}}$

the objective is to provide ultra-clean air and the air change rate may be well over 150 air changes per hour. The high air flow rates assure that the supply air temperature will be very nearly equal to the room air temperature and thus there are no natural convection effects in the space.

It has been shown that supplying cooling from a completely perforated over-head panel with low velocity air will result in buoyancy effects taking over and causing drafts within the occupied zone. The low velocity air collects in a layer at the ceiling, until a breakthrough of cool air at one side of the room occurs and the cool air descends in a stream into the occupied zone. Once the breakthrough has been established, the cool air continues to flow across the ceiling to the break-through point and into the occupied zone. For this reason it is necessary that only part of the ceiling be perforated. Preferably this should be accomplished with alternated perforated and non-perforated panels.

This approach of making the room essentially a cross-section of a duct is impracticable for most air distribution applications. The ducting is prohibitive in cost and, more important, for average velocities below 50 fpm, buoyancy effects will prevent maintenance of a constant velocity throughout the space.

For most applications, a better approach is to supply air in such a way that the high velocity air stream from the outlet does not enter the occupied zone. It is practical to consider the region within 1 ft. of the walls as outside the occupied zone as well as the region above the heads of the occupants. If the limits of the throw are outside the occupied zone, the average motion within the occupied zone will necessarily be less than the terminal velocity at the end of the throw. The supply air should be spread in a thin layer over the surfaces, to surround the occupied zone with conditioned air. The air within the occupied zone will then move toward the total air stream, the mixture of primary and room air. The room air carries the load with it into the air stream and the room conditions are maintained by constant mixing of the room and supply air.

Referring again to Fig. 37 and specifically the upper left-hand diagram, the method of introducing the supply air in such a way that the high velocity air stream does not enter the occupied zone can be considered further.

If the marked terminal point for 200 CFM and 600 fpm is considered, the net drop of the air stream below the outlet is 2 ft. Thus to prevent the high velocity air stream from entering the occupied zone (floor to 6 ft. above the floor) the outlet should be mounted 8 ft. above the floor and the ceiling height should be at least 10 ft. Also the space from the outlet to the opposite wall should be at least 26 ft. Since it is not necessary to throw the air over the entire space, the space could be considerably larger, such as 30 ft. to 40 ft.

More often than not the space is smaller than the throw and the high velocity air stream hits and follows down the opposite wall. When this is the case, the air stream remains close to the opposite wall and does not enter the occupied zone, if the space 1 ft. from the wall is not included as part of the occupied zone.

The room characteristics are shown in Fig. 37A for the same conditions referred to above, with 300 CFM and 600 fpm outlet velocity, except that the space is only 20 ft. The following terminal velocities and corresponding throws were calculated from Equation 1 with $K = 5$ and $A_o = 0.5$ sq. ft. For a terminal velocity of 50 fpm the throw was reduced 20% as recommended:

TERMINAL VELOCITY, fpm	150	100	50
AT THROW, FT. FROM OUTLET	14	21	34

The high velocities on the opposite wall vary from maximums of 50 fpm at the floor to 100 fpm near the ceiling. In this terminal region the air stream temperatures are usually within one degree of the room temperatures.

Where throws to various terminal velocities are known for outlets, it is possible to estimate the velocities near walls, or due to overlapping air streams from two outlets, as done above. The air stream is assumed to follow the surfaces horizontally and vertically a total distance equal to the throw.

Outlet Selection

Present commercial selection data on outlets show drop or recommended ceiling heights to avoid the high velocity air stream entering the occupied zone. Minimum and maximum throw or radius of diffusion are usually indications of the space limitations. When the space equals or is larger than the maximum throw little or no overblow will occur but when the space equals the minimum throw, overblow, such as covered before, exists.

Although specific information is not given here for all outlets, Table 1 does contain throw coefficients for some general classes of outlets. Regardless of heating or cooling, the throw to a terminal velocity of 150 fpm will be the same and could be used to size outlets to satisfy the limits which follow.

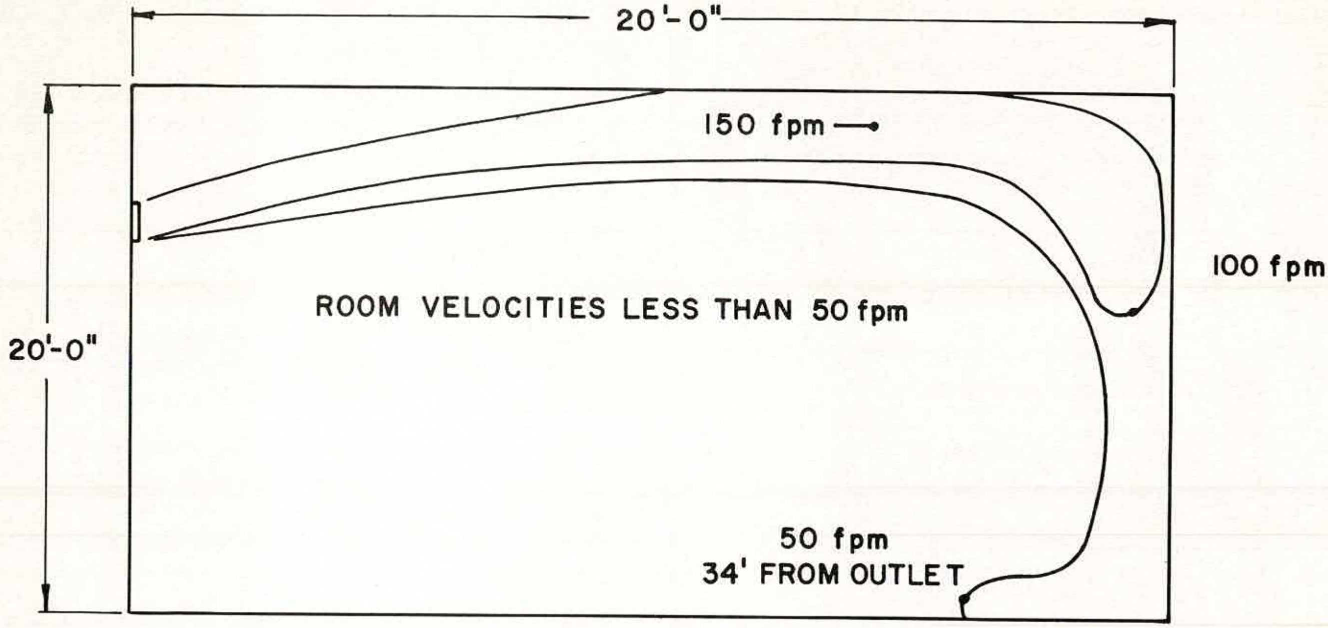


FIG. 37A

Room Air Characteristics With The Outlet Throw Greater Than The Space

TABLE 1

OUTLET TYPE	DISCHARGE PATTERN	AREA (A_o)	COEFFICIENT
High Sidewall-Grilles	0° Deflection	Core Area	5.0
	45° Deflection	Core Area	3.4
Linear	Core Less Than 4" High	Core Area	4.0
	Core More Than 4" High	Core Area	4.5
Low Sidewall	Up and On Wall No Spread	Free Area	4.5
	Wide Spread	Free Area	3.0
Baseboard	Up and On Wall No Spread	Free Area	4.0
	Wide Spread	Free Area	2.0
Floor	No Spread	Free Area	4.7
	Wide Spread	Free Area	1.6
Ceiling-Circular	360° Horizontal	Neck Area	1.0
Pattern	4 Way - Little Spread	Neck Area	2.7
	Free Area - 0.5 Neck Area		
Linear	One Way - Horizontal Along Ceiling	Free Area	5.5

For selection considerations, all outlets can be placed in four groups, as shown in Fig. 34, according to the method of introducing the air into the room. In the diagrams of Fig. 34 the primary air pattern is shown by the envelope of air with a constant velocity of about 200 fpm. The total air, mixture of primary and room air, is shown in the diagonally lined envelope of air with a velocity of about 50 fpm. Air velocities inside the envelopes are higher than 50 fpm.

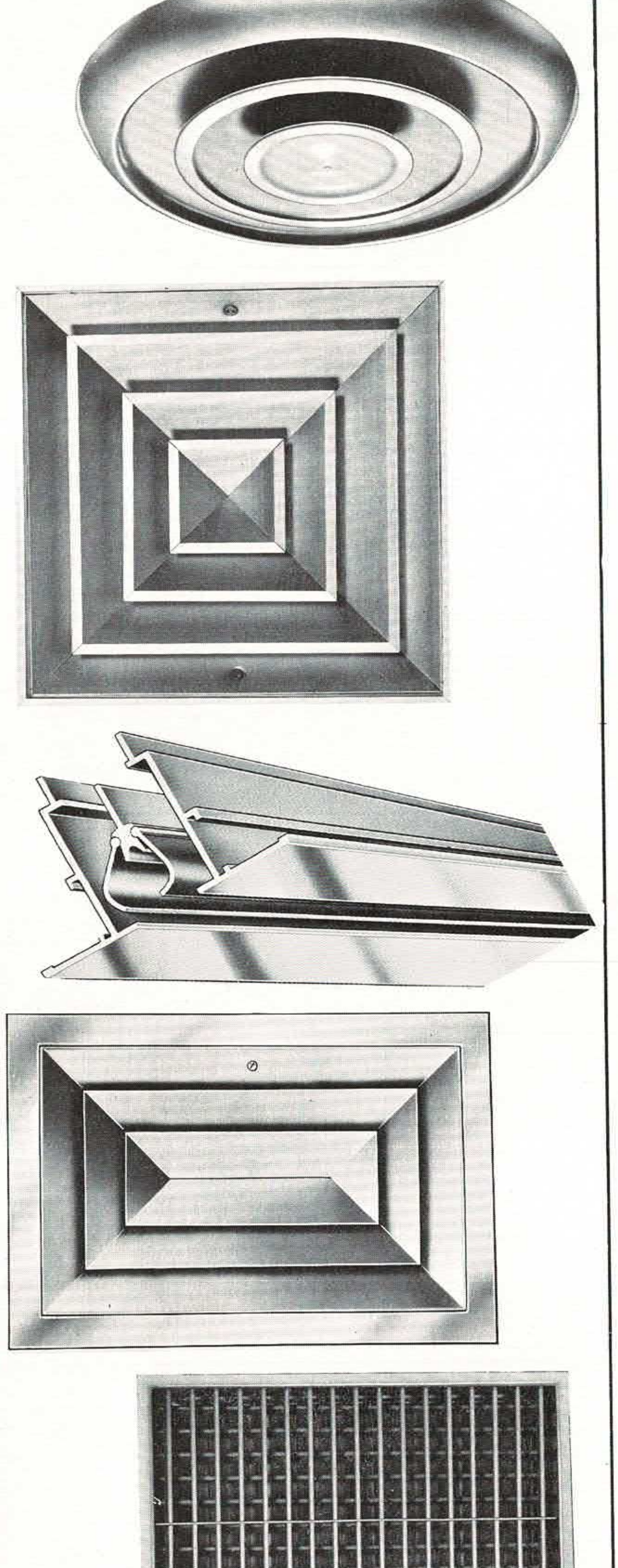


FIG. 38 Group A Outlets
Sidewall, Ceiling Circular, Linear

GROUP A – Outlets Mounted in or Near the Ceiling With a Horizontal Discharge

Group A includes high sidewall type outlets and ceiling diffusers (See Fig. 38). The characteristics from this type of outlet are shown from a sidewall outlet with a 20° right and left deflection of the primary air. In this case, the total air during cooling is shown to have considerable overblow.

The region in which the total air enters the occupied zone is of the greatest concern and may be caused either by overblow, as shown in the diagram, or simply the drop region of a relatively free space condition.

The drop and amount of overblow could be controlled by installing a greater number of outlets, or by spreading the air pattern in a greater arc over the ceiling, such as a full 360° pattern of discharge from a ceiling diffuser.

Since the cool air is discharged at the ceiling level, cooling will be effected from the ceiling down and no stagnant zone will be present.

During heating, it is shown that the total air remains near the ceiling forming a large stagnant zone across the whole room at the level just below the terminal point of the total air. It can be visualized that if the total air covers the ceiling and down the far wall to the floor, the stagnant zone would be broken up. In other words, ideally, it would be desired to cover the whole room by an envelope of total air. It is possible to decrease the stagnant zone and obtain more air in the occupied zone by reducing the spread from the outlet or by reducing the number of outlets for the same quantity of air being handled.

From these characteristics it can be seen that the primary application of the group A outlets is for ventilation and cooling. In this application, it is desirable to use a relatively large number of outlets with considerable spread to obtain the optimum performance. For heating, it would be desirable to use fewer outlets and use a discharge pattern with as little spread as possible. The long throw from this type unit would be utilized to hit the exposed wall and bring the warm total air down to the floor level along the exposed surface. For a year-around system these units may be selected to give a long throw with a non-spreading jet which would hit the wall with some velocity. Then, during the cooling season, these units could be deflected to give a spreading pattern with a shorter throw, which possibly would not reach the exposed wall.

Due to the characteristics shown for these units, the cooling temperature differential would not be critical. However, during heating, with high side-wall units, the temperature differential should be limited to approximately 25°. With ceiling diffusers, with a full circular pattern, the temperature differential should be limited to approximately 20°.

Return intakes generally should be located in the stagnant zones shown in Fig. 31. Therefore, the preferred location for return air intakes used with Group A outlets is near the floor for heating and year around applications. During cooling the location of return air intakes is not critical.

GROUP B – Outlets Mounted in or Near the Floor with Non-Spreading Vertical Jets

These outlets include floor registers, short baseboards, low sidewall outlets -- any type of outlet which has a vertical discharge with a non-spreading air stream. (See Fig. 39)

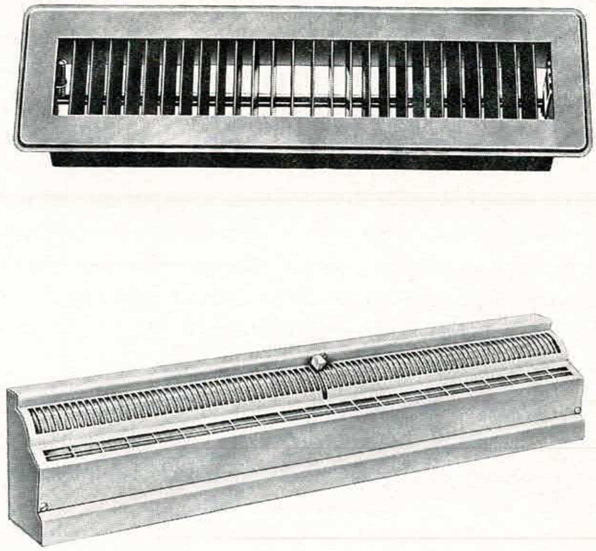


FIG. 39

Group B Outlets
Floor Registers, Short Baseboard,
Non Spreading

During cooling, the total air will reach the ceiling and spread out over the ceiling with the total air dropping at some point into the occupied space forming a stagnant zone above the terminal point of the total air and located on the opposite portion of the room from the outlet. The occupied zone below the total air will be cooled properly and the air motion will be relatively uniform.

During heating, the total air will follow the ceiling across and down the opposite wall and form a stagnant zone between the floor and the terminal point of the total air. It will be noted that this stagnant zone will usually be somewhat smaller in magnitude than that found with the Group A outlets. Consequently, the temperature gradients in the occupied zone with these outlets will also be less than those experienced with the Group A outlets.

In the selection of these units, the location would not be too critical. They may be located adjacent or on an inside wall or an outside wall. The exception to this, of course, would be a case in which the air was introduced directly under a window which provides a large source of heat gain or heat loss. However, since the air stream is not spread, the amount of natural convection currents counteracted by this stream is generally minor.

The general selection of this type unit is dependent more on the cooling characteristics than the heating characteristics. This requires that the supply velocity be high enough to project the cool air up to the ceiling level. For these conditions, and with a floor register, a minimum velocity of 500 fpm with a 20° temperature differential is required, and that only 250 fpm with a 15° temperature differential is required. During heating, temperature differentials between 25° and 35° could be handled satisfactorily. These characteristics imply that this type outlet would have good application with heating, cooling and ventilation, utilizing the same relatively high flow rate during all seasons of operation. The preferred location of the return intakes with these outlets is the same as with the Group A outlets.

GROUP C – Outlets In or Near the Floor with Vertical Spreading Jets

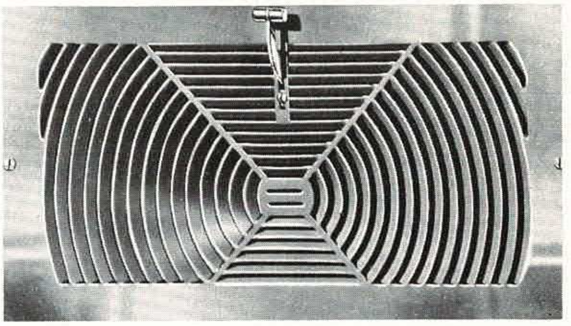
The outlets in this group may be floor diffusers, baseboards, and low sidewall units, which have wide spreading vertical jets in contrast to the non-spreading jets of the Group B outlets. (See Fig. 40)

With outlets of this type, during cooling, the total air may or may not reach the ceiling level because of the strong tendency for the total air to fold back on the side spreading primary air streams as shown in the diagram for the Group C outlets. This results in a relatively large stagnant zone near the ceiling and consequently, the only portion of the room which would be cooled would be the portion below the point of maximum projection of the total air. The region of maximum velocity within the space would be in and near the total air.

During heating, the total air from this unit spreads out across the wall on which the outlet is placed, up to and across the ceiling and down the opposite wall. It will be noted that in this case, the stagnant zone during heating is less than with the other two groups. This is true even though the throw of the primary air may be considerably less than that experienced with the other groups. The very satisfactory room air and temperature conditions with this type of outlet can be attributed partially to the fact that the primary air results in reduced buoyancy effects in the total air at the ceiling level.

It is shown by these diagrams that this type of unit should primarily be applied to the heating cycle in which nearly optimum conditions can be obtained. Also, because of the wide spreading jets, this type outlet is better placed on the perimeter of a building because these jets will overcome nearly all of the natural convection currents which are formed at this exposure.

Although these units provide the optimum distribution during heating, the selection may still be based on the cooling characteristics. With this type of unit, the minimum supply velocity required to have the total air reach the ceiling level depends, to a great extent, upon the amount of spread. With a floor diffuser, a supply velocity of 750 fpm with a 20° temperature differential is required, whereas with a 15° temperature differential, a supply velocity during heating is not of critical importance; however, higher velocities will produce better results than low velocities. The temperature differential which may be handled with this type unit during heating may be in the order of 65° to 75°.



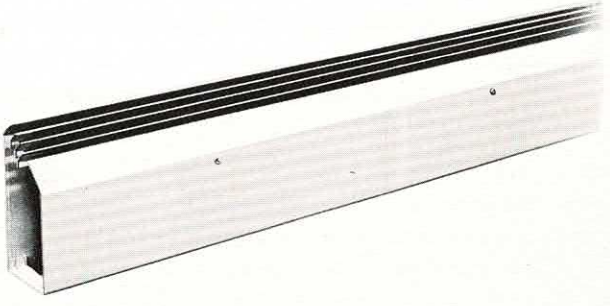


FIG. 40
Group C Outlets
Floor Diffusers, Long Baseboards

GROUP D – Outlets Mounted in or Near the Floor with a Horizontal Discharge. (See Fig. 41)

Outlets in this group include baseboard units, low sidewall units which have a horizontal discharge, either with a spreading jet or a single straight out jet. As shown by the characteristic diagrams, the cool air "puddles" on the floor resulting in a large stagnant zone across the full room. Since the only cooling which would be effected would be below the projection of the total air, this type of distribution is considered unsatisfactory for cooling.

During heating, the warm total air rises at some point in the room, depending on the velocity and temperature of the primary air as it is discharged horizontally across the floor. Since the warm air is introduced at a point to counteract the cool air which tends to accumulate on the floor, the temperature variations within the space would be very uniform. However, from the diagram, it will be noted that the high velocity total air is within the occupied space. For a continuous blower operation with constant supply temperature, this condition would not be too objectionable. Normally, a system has a cyclic operation and

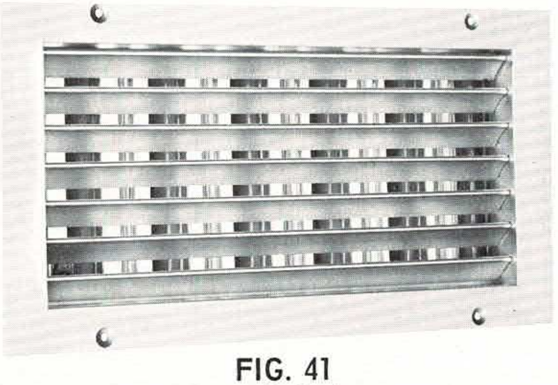


FIG. 41
Group D Outlets
Low Sidewall, Baseboards, Horizontal Discharge

at the point near the blower cut-off relatively high velocity and low temperature air is discharged within the occupied space. To minimize the effects of cyclic operation, the supply velocity should be limited to 300 fpm.

The application of these units is obviously preferred for heating only, and thus the return intake location is not critical.

The Return Intake

From the previous consideration of the stagnant zone, the return generally should be located inside the stagnant zone. In this location, the warmest air is returned during cooling and the coolest air during heating. Rather than have two locations for a year around application, the return should be located according to the relative size of the heating and cooling stagnant zones for the type of outlet selected. For example, with horizontal projection at high levels, no appreciable stagnant zone exists during cooling, but a large stagnant zone might exist during heating. Thus, the return should be located near the floor to favor the heating performance.

In the discussion of air distribution, the temperature differential considered is from supply outlet to the room. The air conditioner handles the load based on the supply and intake return temperatures, according to the formula:

q_T = $1.08\ Q\Delta_t$, where:

q_T = BTU/hr. = $q_R + q_E$

Q = CFM

Δ_t = $T_I - T_S$ (Intake temp. - Supply Temp.)

q_T = Total load, BTU/hr.; q_R = Load handled in Room;

q_E = Load handled Externally)

If the intake temperature and the room temperature are the same, then the room load is satisfied by the formula:

q_R = $1.08\ Q\ \Delta_t$, where:

q_R = BTU/hr. (Room load) $= q_T$; $q_E = 0$

Q = CFM

Δ_t = $T_R - T_S$ (Room Temp. - Supply Temp.)

In this case, either the room temperature or the intake temperature can be used in the calculation.

However, by utilizing the stratification, it is possible that the intake temperature be 5 or 10 degrees warmer than the room temperature even though the room temperature is maintained at the desired level.

For example, if 400 CFM at 20° Δ_t satisfies the room load (q_R); and the return air temperature (T_I) is increased 5°F. with the same CFM (Q), the supply air temperature (T_S) must increase 5°F. as well.

If Δ_t for T_I and T_S stays the same, then a part of the total load is handled externally to the room as shown by the formula:

q_E = $1.08\ Q\Delta_t$, where:

q_E = External load ($q_T - q_R$)

Q = CFM

Δ_t = $T_I - T_R$ (Intake Temp. - Room Temp.)

Then: $(1.08)(400)(20) = (1.08)(Q)(25)$

$$Q = \frac{(400)(20)}{25} = 320 \text{ CFM}$$

$$\%\ \text{decrease} = \frac{(400 - 320)}{400} \times 100 = 20\%$$

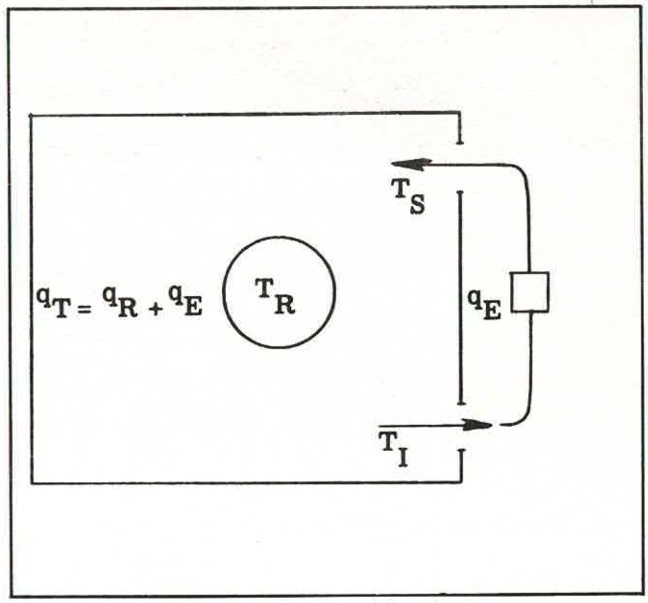
It should be noted that even though there is a 20% decrease in the required CFM, there is no reduction in the refrigeration load. The room load is still based on the difference between supply and room temperature and the refrigeration load is still based on the difference between the supply and inlet temperatures.

Based on sensible heat alone, a 5°F rise in return air temperature requires an equal rise in the supply air temperature with a constant air flow rate. During heating, this increased supply air temperature is undesirable, as it increases the buoyancy effects and the size of the stagnant zone.

An example would be one in which a high sidewall outlet and a high sidewall return are being used for heating, with for example, a slab construction. Since the warm air is introduced at the ceiling level and returned at this point, as the supply air temperature gets warmer and warmer so also does the return air temperature. Therefore, in order to satisfy the room load it is necessary for the supply air temperature to rise in accordance with the rise in the return temperature. This ever cycling spiral can result in a situation in which the burner is operating continuously and the room is never satisfied. During cooling, the increased supply air temperature is beneficial and reduces the buoyancy effects.

Remember, the supply air maintains the conditions within a space by mixing and dilution. Any removal of excess warm or cold air which is allowed to stratify before mixing within the space will permit lower temperature differentials or lower air flow rates. Removal of excess warm or cold air is accomplished with hoods in certain industrial processes, but it can also be done by selecting the supply outlet performance to promote the formation of the stagnant zone directly from the local heat gain or loss. The return intake or exhaust would then be located in the stagnant zone.

In addition to the location of the return intake as discussed above, the intake should be sized to return the proper amount of air to the air conditioner with a minimum static pressure requirement and noise level. In general, most commercial return grilles have a free area of between 45 and 55 percent, because they are designed for the so-called "no see through." With this type of grille the face velocity should not exceed approximately 500 fpm to have a reasonable pressure requirement and a reasonable sound level. In general, these units should be sized on the available pressure requirements and the sound data, rather than relying on indicated free area values.



Temperature relationships in a room.

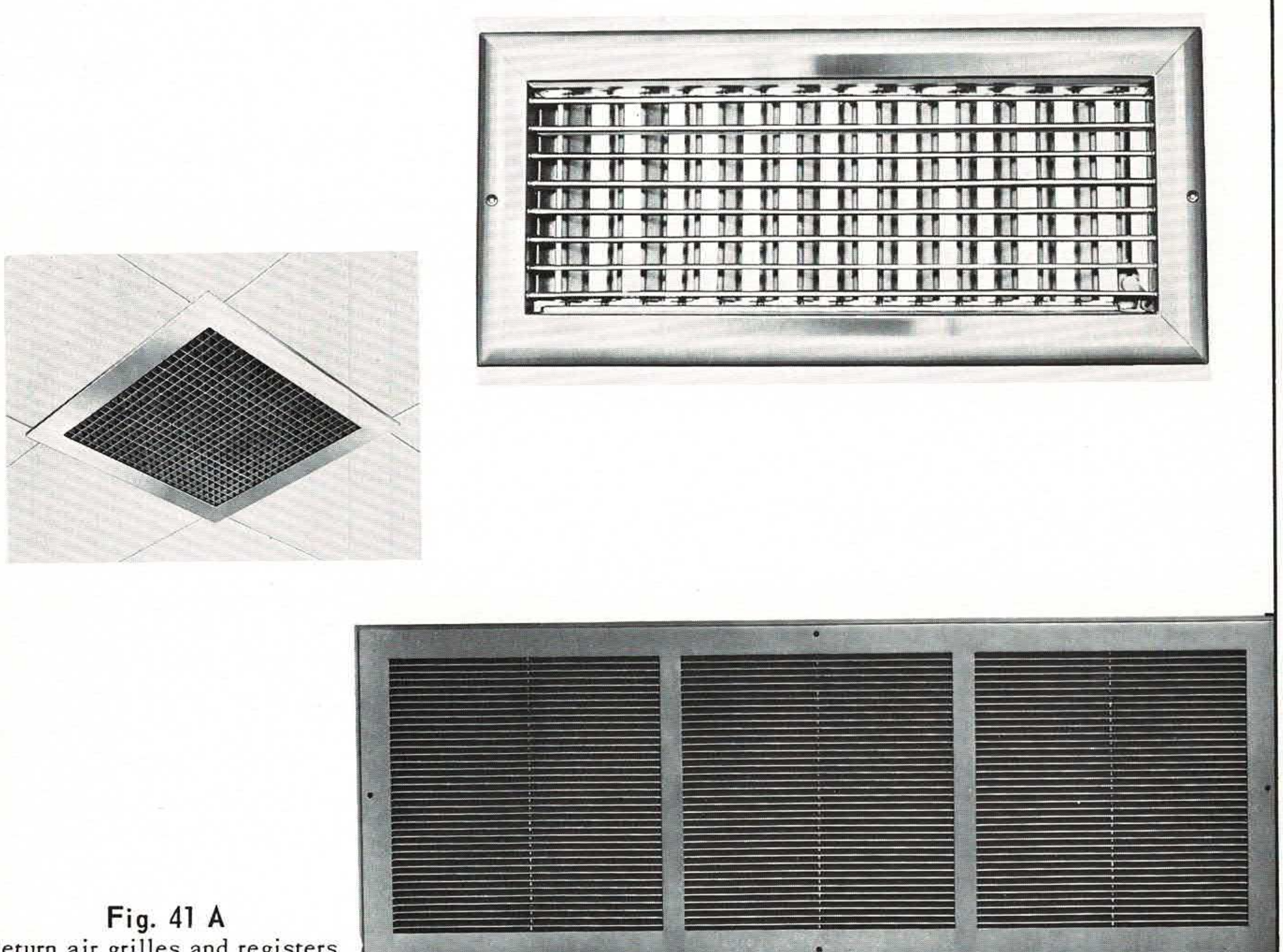


Fig. 41 A
Return air grilles and registers

CHAPTER 17
DAMPERS AND TERMINAL UNITS

Dampers and Pressure Reducing Valves

A damper is a primary element in the duct system and is used for controlling air flow rates by introducing a resistance to air flow in the system. In high pressure systems the damper is referred to as a pressure reducing valve.

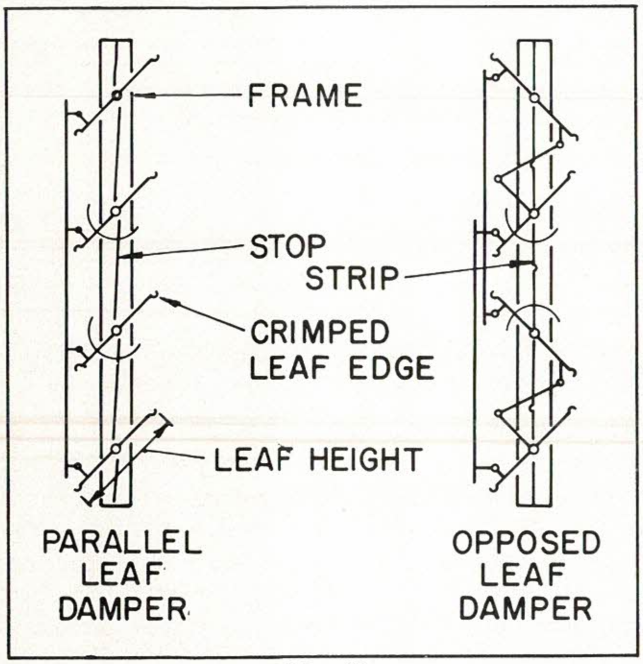


FIG. 42

Parallel and opposed damper operating linkage.

Fig. 42 shows two types of multiple blade dampers which are the parallel blade and opposed blade dampers. The terms parallel and opposed refer to the movement of the adjacent blades. In the parallel blade damper all of the blades move in parallel. The opposed blade damper has a linkage which causes the adjacent blades to move in opposite directions.

Partial closing of the damper or pressure reducing valve increases the resistance to air flow of the duct system. The reduction in air flow with closure of the damper may or may not be proportional to the amount of adjustment of the damper. That is, closing the damper half-way does not necessarily mean that the air volume will be reduced to fifty percent of that volume which flows through the damper when it is wide

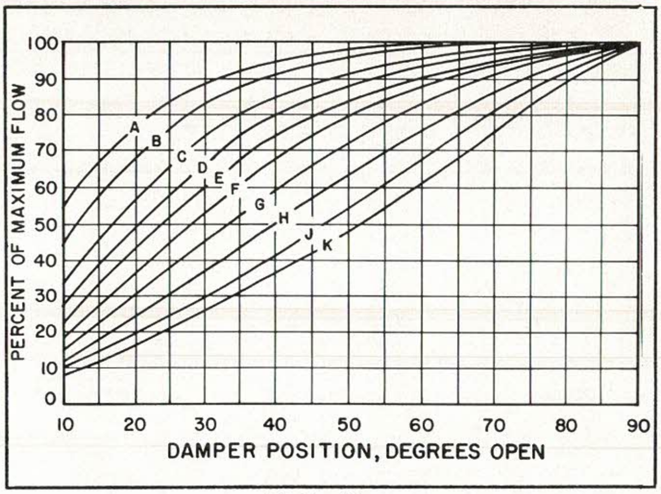


FIG. 43

Flow characteristics for a parallel operating damper.

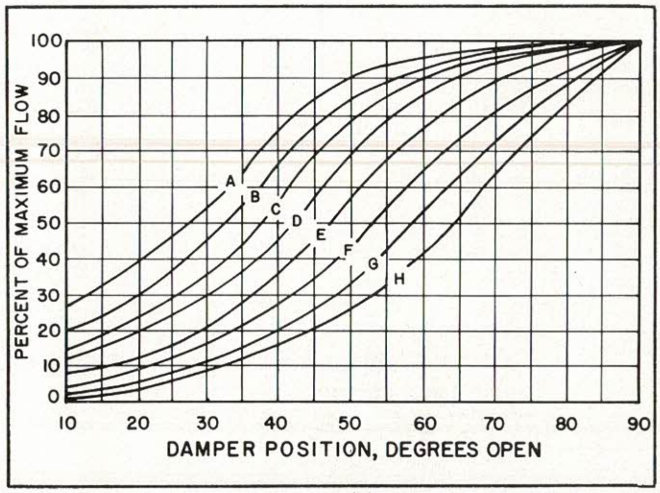


FIG. 44

Flow characteristics for an opposed operating damper.

open. The relation between the position of the damper and the percent of air that flows through the damper with respect to the air flow through the wide open damper is termed the "flow characteristic."

Typical flow characteristic curves for the parallel blade and opposed blade dampers are shown in Fig. 43 and Fig. 44. In Fig. 43 the flow characteristic curves for the parallel blade damper show that as the damper is closed the flow reduction may be proportional to the closure of the damper as is shown by curve J, or partial closure of the damper may have little effect on the flow as is shown by curve A.

The manner in which the damper reacts in the duct system is determined by how complicated the system is. If the system is very simple and the damper makes up a major part of the resistance in the system then any movement of the damper will change the resistance of the entire system and good control of the air flow will result. If the damper resistance is very small in relation to that of the entire system a poor flow characteristic such as curve A in Fig. 43 and Fig. 44 will result. In Table 1 typical ratios of damper to system resistance are shown for each flow characteristic curve.

TABLE 1

Parallel-leaf dampers		Opposed-leaf dampers	
Open damper resistance, percent of system resistance	Flow characteristic curve	Open damper resistance, percent of system resistance	Flow characteristic curve
0.5 – 1.0	A	0.3 – 0.5	A
1.0 – 1.5	B	0.5 – 0.8	B
1.5 – 2.5	C	0.8 – 1.5	C
2.5 – 3.5	D	1.5 – 2.5	D
3.5 – 5.5	E	2.5 – 5.5	E
5.5 – 9.0	F	5.5 – 13.5	F
9.0 – 15.0	G	13.5 – 25.5	G
15.0 – 20.0	H	25.5 – 37.5	H
20.0 – 30.0	J		
30.0 – 50.0	K		

Typical ratios of damper to system resistance for flow characteristic curve.

The set of curves for the opposed blade damper, Fig. 44, show that for a given ratio of damper to system resistance a better flow characteristic usually results than with the parallel blade damper, Fig. 43. The improved characteristic results from the fact that as the opposed blade damper is closed it introduces more resistance to air flow for a given position than does the parallel blade damper.

In balancing systems it should be realized that the flow characteristics of a damper are not constant and will vary from one system to another. The actual effect of closing the damper can only be determined by measurements in the particular system unless the system designer has taken into account the damper flow characteristics in his system design.

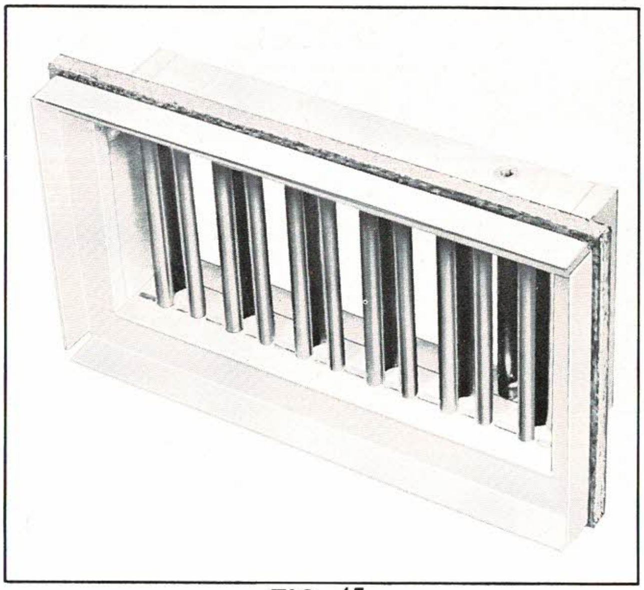


FIG. 45

Typical pressure reducing valve.

Terminal Units

Terminal units are single duct, dual duct or induction units which are located either in or above the conditioned space. As the name implies, the units are located where the duct system terminates and the air is introduced into the space to be conditioned.

The terminal unit has several functions. First, it must supply air at a proper temperature to take care of the load in the conditioned space and this is done in response to a room thermostat located in the space. The unit also contains some type of device to regulate the air flow to the space. Pressure is reduced in the terminal unit to a level where the air can be introduced into the space. Any noise that is generated within the unit in the reduction of the pressure must be attenuated within the terminal unit.

Dual Duct Units

A typical dual duct unit is shown in Fig. 46. The unit is supplied with both hot and cold air. The inlet valve is positioned by a pneumatic motor in response to a room thermostat to supply air at the proper temperature to satisfy the load within the space. The dual duct unit may supply warm air for heating or cool air for cooling, or mix the two air streams to satisfy any condition in the space.

It is important to have an automatic air flow control device within the dual duct unit since the pressure in the hot and cold ducts may vary over a wide range due to the fact that the demand for the hot or cold air will vary throughout the building. The air flow may be regulated with a mechanical volume regulator as shown in

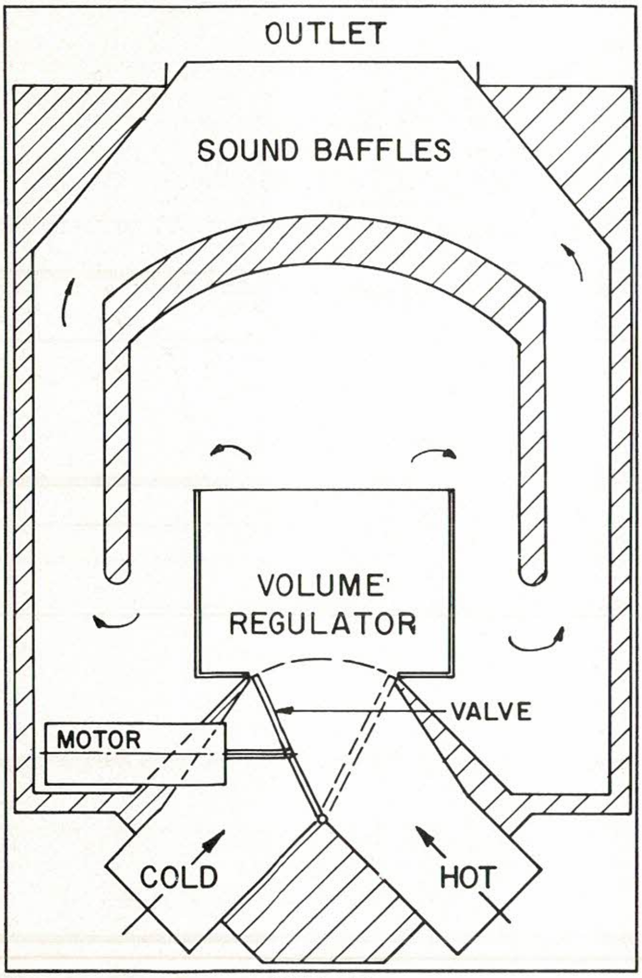


FIG. 46
Dual duct unit with single motor actuated by the room thermostat to supply warm or cool air.

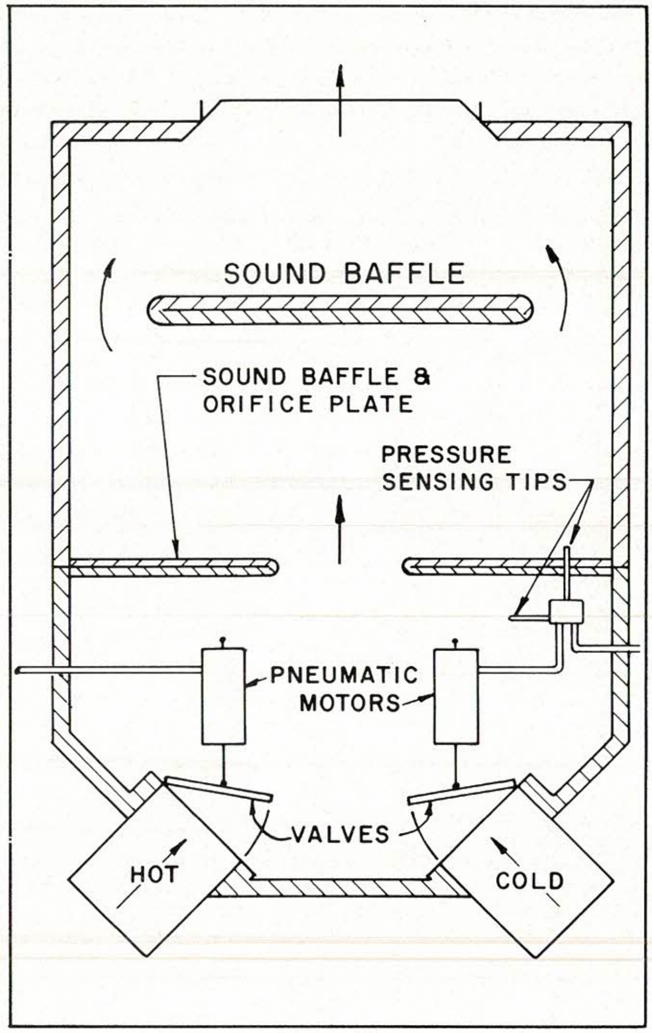


FIG. 47
Dual duct unit with two motors positioned to maintain the space temperature by a pneumatic relay.

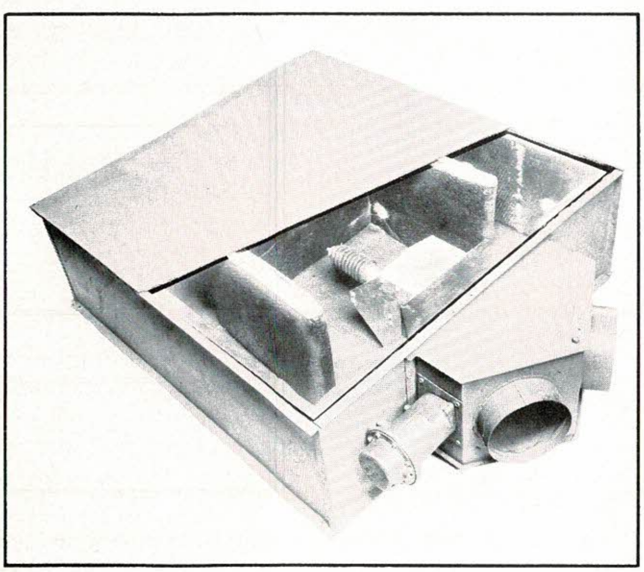


FIG. 48 Dual duct mixing box unit with single motor.

Fig. 48. Mechanical volume regulators utilize curtains and perforated plates or damper blades which decrease the available flow area as the pressure at the inlet to the box increases. The inlet pressure is balanced against springs on the curtains or damper blades in such a way that the flow area varies to maintain a constant air flow rate through the box. This type of flow regulating device depends on the energy in the flowing air stream to actuate it.

A second type of automatic flow regulation is provided by the system shown in Fig. 47. With this arrangement, two pneumatic motors are used. The motor connected to the hot valve responds to the room thermostat and opens or closes the hot valve to maintain proper temperature in the space. The motor on the cold valve is connected to the room thermostat but also has a pneumatic relay connected to it which senses the pressure difference across an orifice in the box. Therefore, the motors are positioned to maintain the space temperature but the pneumatic relay will adjust the motor on the cold valve to maintain a constant pressure difference across the orifice. Both types of dual duct units with mechanical or pneumatic constant volume control are adjusted by the manufacturer and do not usually require adjustment by the installer.

The volume regulating features in the dual duct units also take care of reducing the pressure in the duct to a level which is consistent with that which exists in the low velocity duct that is connected to the discharge of the box. In performing the function of volume regulation and pressure reduction, considerable noise is generated and this must be attenuated within the terminal unit.

Sound attenuation within the terminal unit is provided with sound baffles as shown in Figs. 46 and 47. The sound baffles are solid metal plates installed to place a barrier in the direct line of the sound that is generated. The baffle reflects the sound back into the box where it can be absorbed by the box lining. Common materials for the box lining are glass fiber blankets. The glass fiber blanket also provides thermal insulation so that the conditioned air within the box will not be heated or cooled by the air in spaces surrounding the box.

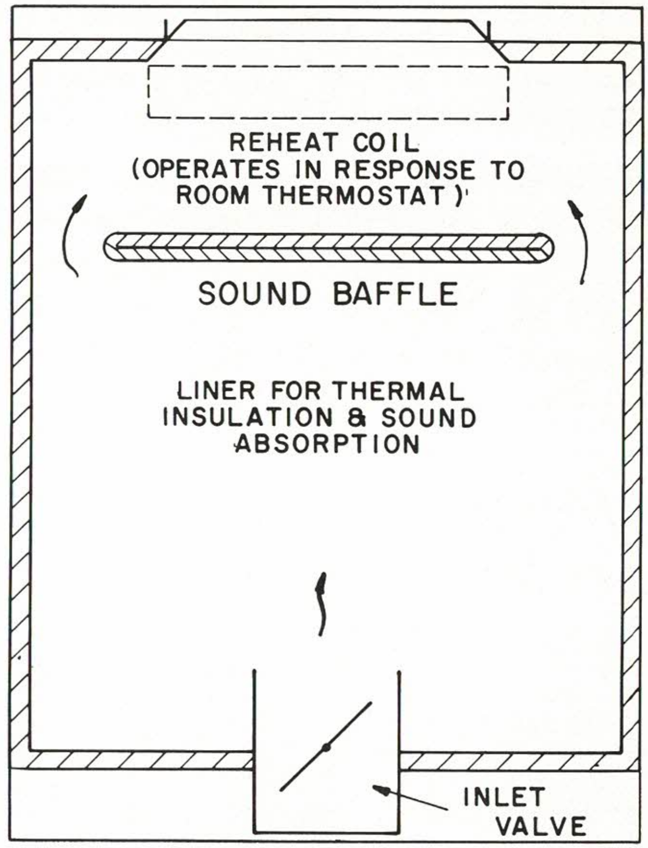


FIG. 49

Single duct unit with reheat coil supplied with air at temperature to handle the cooling load.

Single Duct Units

Single duct units, Fig. 49, as the name implies, have only one duct connection and are supplied with air at a temperature that will take care of the cooling load. In a very simple single duct system it is possible to adjust the temperature of the air to take care of the varying space load. However, in larger systems it is necessary to install a reheat coil in the box or immediately downstream from it which is regulated by a room thermostat. In this case, constant temperature air is supplied to the box and the reheat coil (which may be hot water, steam or electric) is operated by the room thermostat to add heat to the air stream as required to maintain a constant temperature within the conditioned space.

The single duct unit usually has a manually operated inlet valve which is adjusted when the system is initially balanced and then left in that position to maintain a constant air flow through the box. To balance a system with the manually operated valve it is necessary to determine the air flow either by measuring the air flow at the box outlet or in some cases by measuring the pressure difference across an orifice within the box. Some of the more recent single duct units include a mechanical constant volume regulator such as the type described for the dual duct units. In this case the mechanical volume regulator is pre-set at the factory and no balancing is required in the field.

Noise is also generated in these units due to pressure induction and it is necessary to have a sound baffle within the unit and also a liner to absorb the noise that is generated.

Induction Units

A typical induction unit is shown in Fig. 50. The induction unit is usually a single duct unit which is supplied with primary air at a constant temperature. The primary air, at high pressure, is supplied to a nozzle within the unit. The air discharges from the nozzle behind a coil and creates a low pressure region. The low pressure region causes air to be induced from the room and this is pulled through a water coil mounted on the unit. The water coil, in response to a room thermostat, is supplied with heated or chilled water to maintain a mixture of primary and induced air that will satisfy the load within the space and maintain a constant air temperature in the conditioned space.

A damper or pressure reducing valve in the inlet of the unit is adjusted at the time the system is balanced to maintain the proper quantity of primary air. A pressure tap is provided on the unit so that the quantity of primary air can be determined from a pressure measurement which is referred to a calibration curve supplied by the manufacturer.

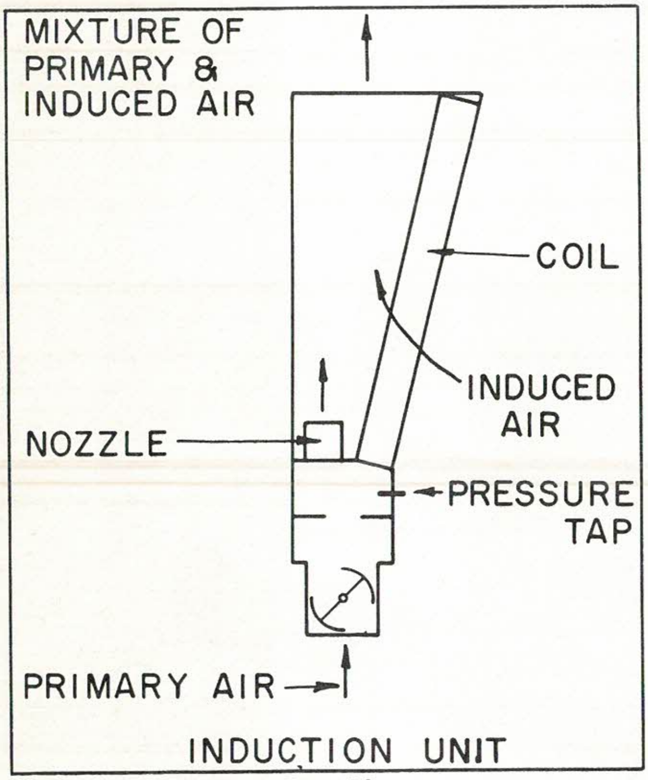


FIG. 50

Single duct induction unit supplied with primary air at constant temperature.

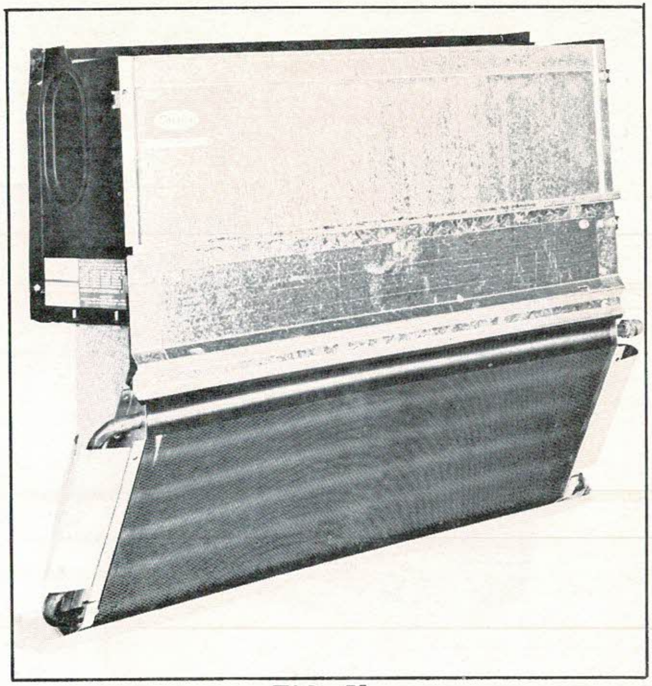


FIG. 51

Typical manufactured induction unit.

This unit is also provided with a means for sound attenuation.

Another type of induction unit which is called the dual-induction unit incorporates the features of both the dual duct unit and the induction unit. This type of terminal unit has a dual duct inlet valve which is supplied with both hot and cold air and includes a mechanical constant volume regulator. The volume regulator is pre-set at the factory and no further adjustment is required for system balancing. The induction coil is included to provide additional cooling capacity. The mixture of hot and cold air is supplied to the nozzles and room air is induced through the coil. The coil is used only at times of peak load and reduces the size of the cold air duct system required in the building. However, it is also necessary to have chilled water piping to take care of the coils.

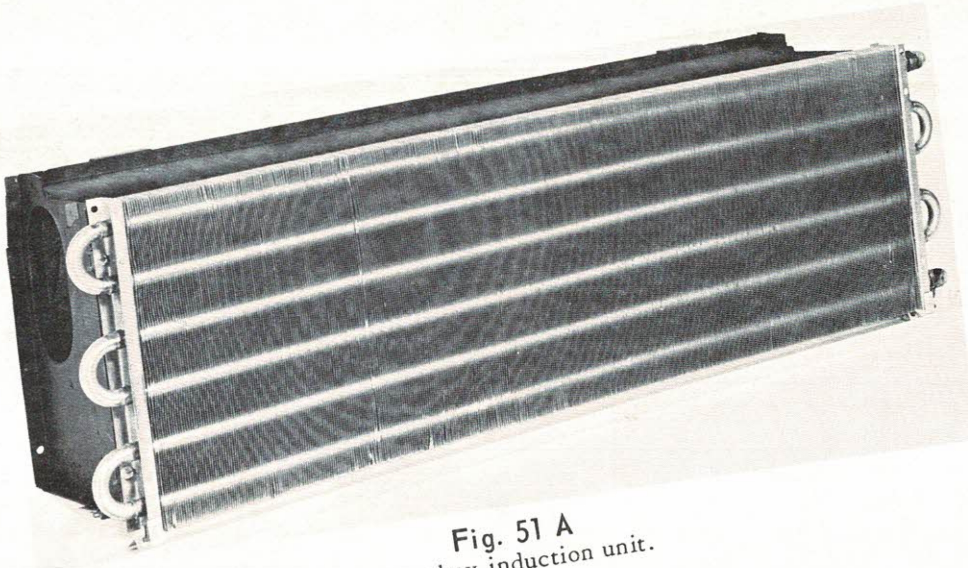


Fig. 51 A
Low-boy induction unit.

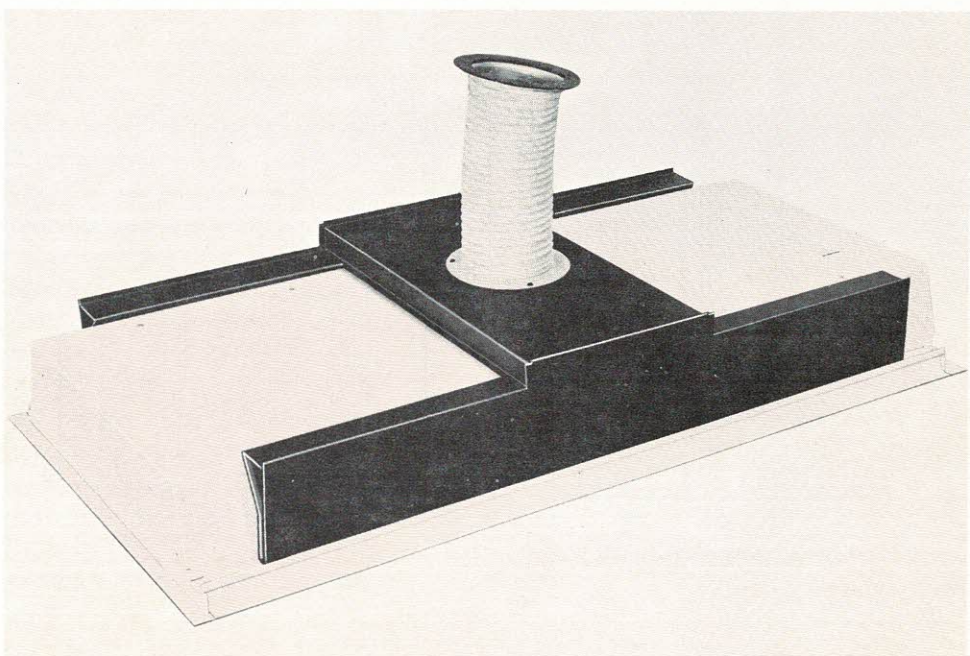


Fig. 51 B
Light fixture with Troffer diffuser

SECTION IV

INSTRUMENTS

CHAPTER 18

DRAFT GAGE

One of the most effective and yet elementary tools for determining what is going on inside a duct is the draft gage, which also may be known as a manometer. The draft gage may be a simple U-shaped, hollow glass or plastic tube partially filled with water with a scale, calibrated in inches between the legs of the U. Or it can be the inclined manometer, using a well, an inclined calibrated runway with a partial filling of colored oil. Or, it can be one of a variety of dry types usually working on the principle of unequal pressures exerted on bellows.

The draft gage measures the pressure differential resulting from the pressures on each leg. Fig. 52 illustrates three situations useful to air balancing:

The static pressure (SP) is that which is required to overcome the frictional resistance of the air against the surface of the duct. The velocity pressure (VP) is that which is required to produce the velocity flow. The total pressure (TP) is the sum of these two components.

As the velocity of the air in the duct varies as the square root of the velocity pressure ($V = 4005 \sqrt{VP}$), then by knowing the size of duct and the velocity pressure it is possible to calculate the flow of air in CFM within the duct. System statics, which is the "pumping" pressure a fan must develop, can be read by using method "A" in Fig. 52 to read the discharge static and suction static of a fan and combining the results. See Table 1 of Chapter 19.

There are dry type draft gages. One such operates on a bellows that actuates a helix which in turn moves a pointer across a printed scale. Others operate by weight or blade displacement. Generally speaking, dry gages are not as accurate as a liquid filled U tube and frequently do not stand up to rough handling.

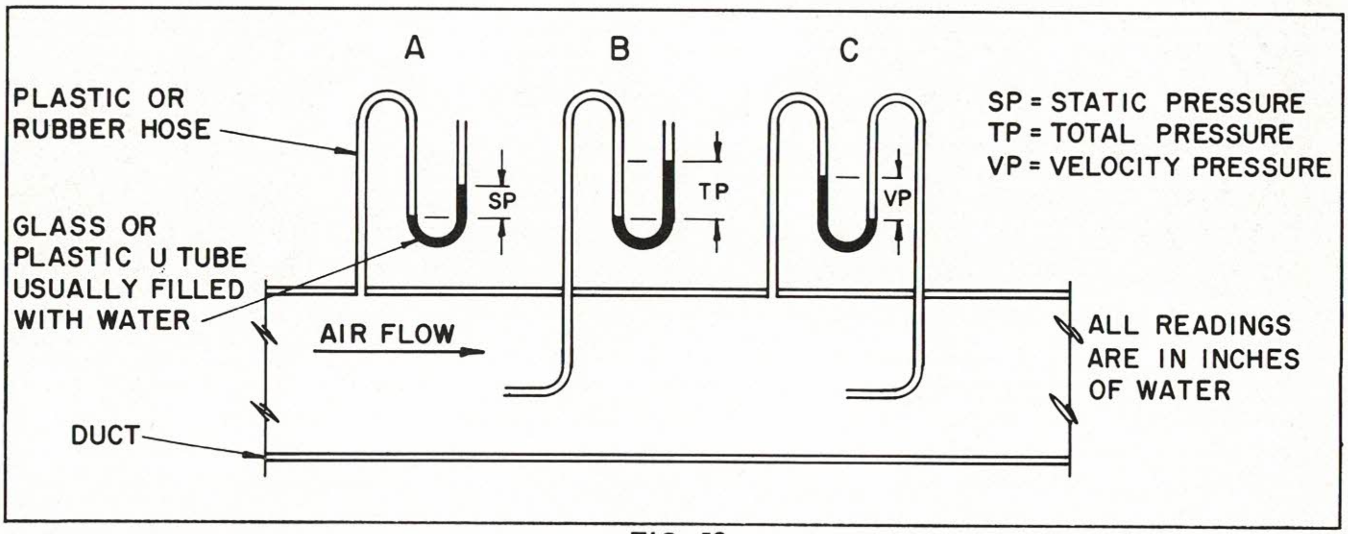
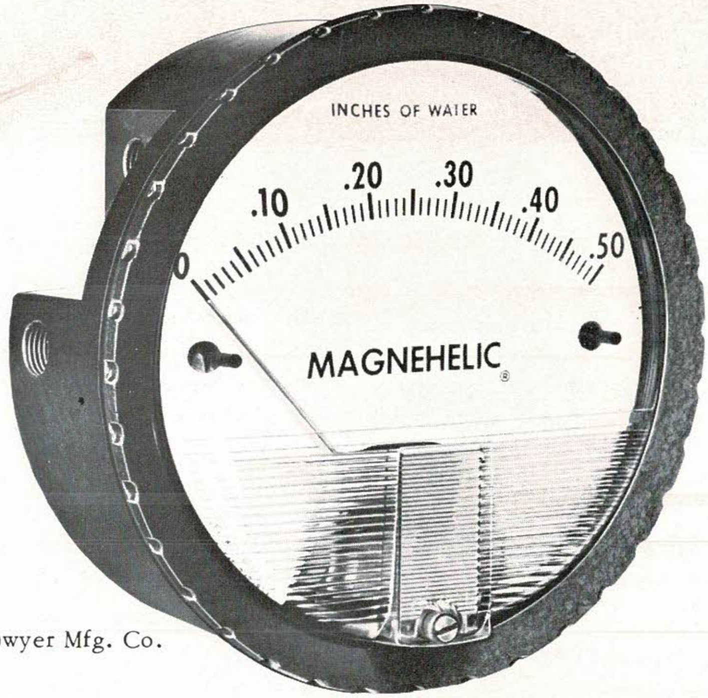


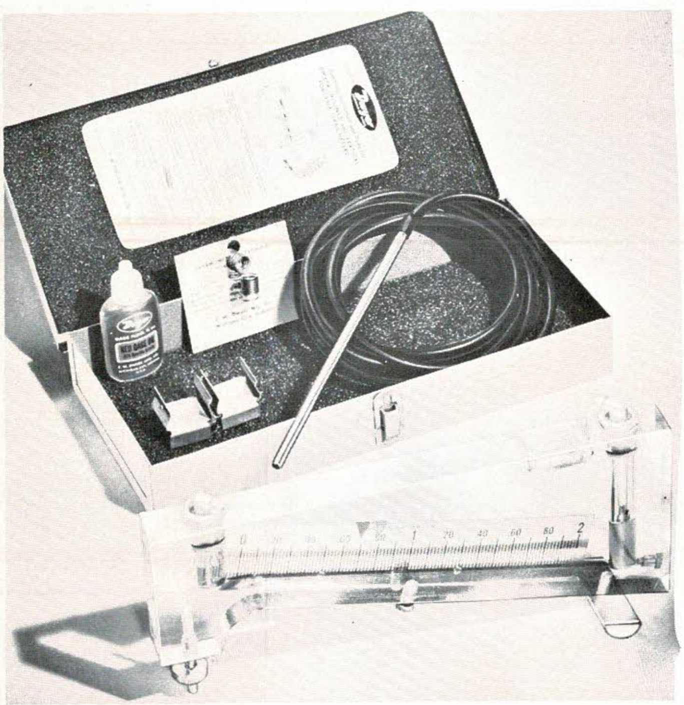
FIG. 52

Diagram showing draft gage hookups to read static pressure, velocity pressure, total pressure.

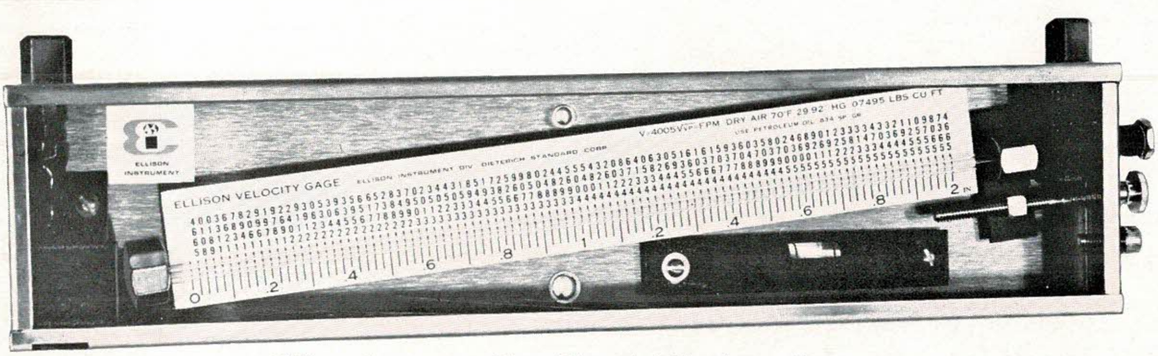


F.W. Dwyer Mfg. Co.

Left – Magnahelic draft gage. Left, center – Inclined portable manometer. Right, center – Slack tube manometer, Bottom – Portable inclined draft gage.



F.W. Dwyer Mfg. Co.



Ellison Instrument Div., Dieterich Products Corp.

CHAPTER 19

THE PITOT TUBE*

Construction

The Pitot tube (named after the Frenchman Pitot who first designed a device of this type) is an instrument for measuring static pressure, velocity pressure and total pressure in a stream of air or gas. As shown in Fig. 54, it consists of two concentric tubes which usually are L-shaped for convenient handling and which end in two separate outlets for connection to draft gages. The essential part of the instrument is the short straight portion of the "L" which is inserted into the air stream, parallel to the direction of flow and with the opening always pointed upstream. The inner tube has an open mount, facing the air stream, and conveys to a draft gage the so-called impact pressure which is the total pressure at this point of the air stream. The outer tube has a number of small radial holes in its cylindrical wall and the annular space between the two tubes conveys to another draft gage the static pressure. Since velocity pressure is the difference between total pressure and static pressure (VP = TP−SP), it can be measured on a third draft gage by simply connecting the total pressure outlet of the Pitot tube to one side of the draft gage and the static pressure outlet to the other side.

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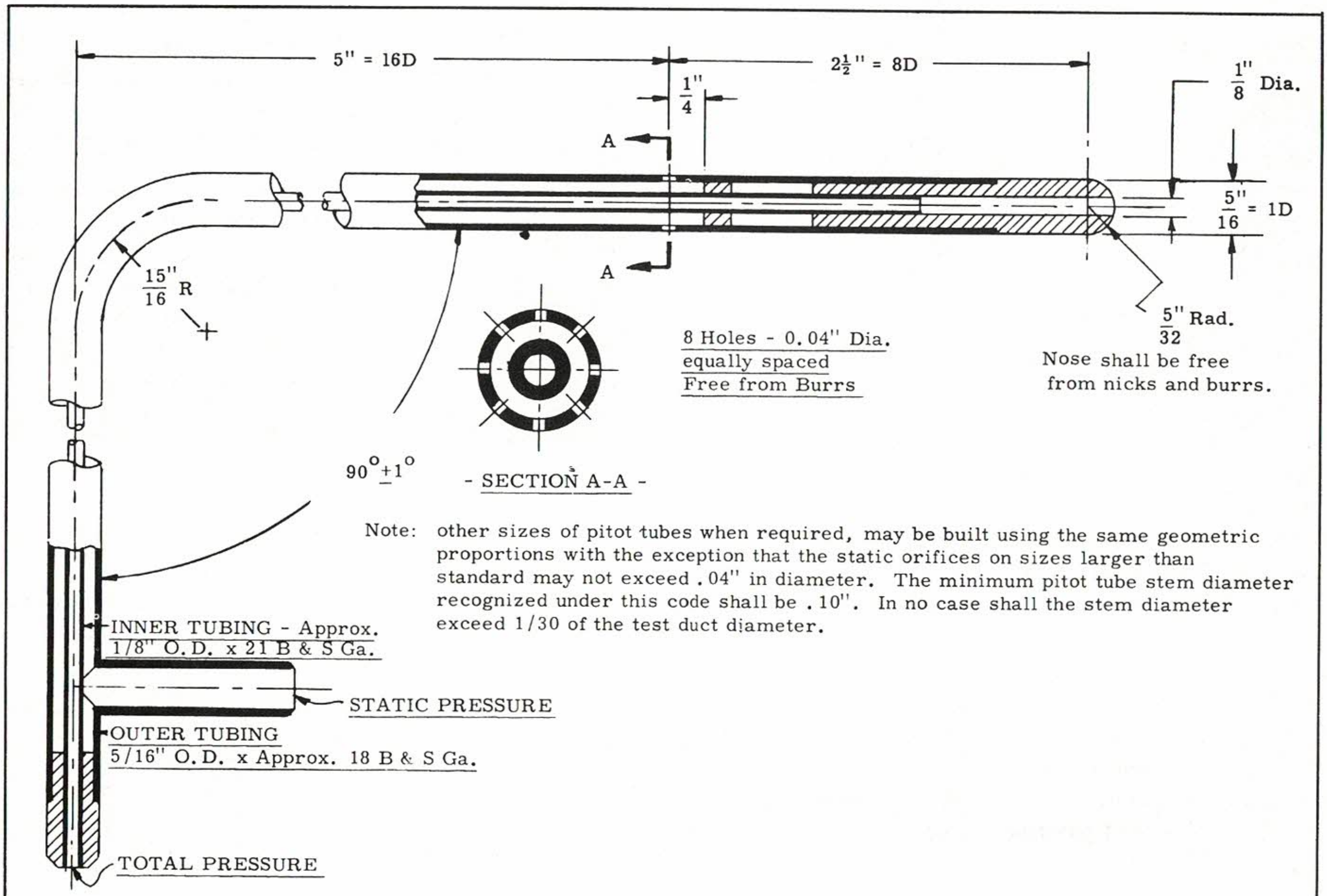


FIG. 54

Diagram showing the two concentric tubes which comprise the Pitot tube. The Pitot tube must be connected to a draft gage or gages. Note construction of that part of the tube which is inserted in the air stream.

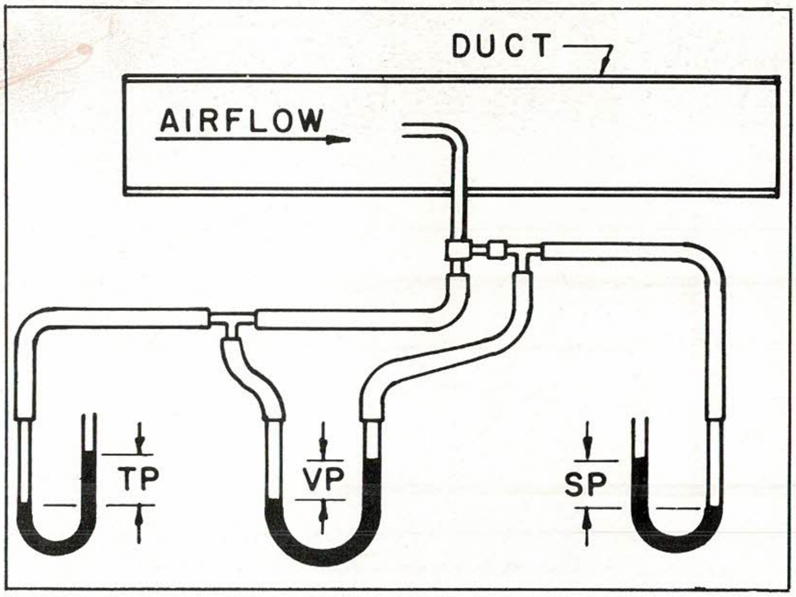


FIG. 55

Diagram showing connections between Pitot tube and the U-tubes to read the TP, VP, SP.

Connections

If static, velocity and total pressure are to be measured simultaneously, three draft gages are connected, as shown in Figs. 56A, 56B, 56C, depending on the specific application. In any case, however, the three values measured will then fulfill the equation

$$VP = TP - SP$$

or

$$TP = SP + VP$$

In conducting tests, it is frequently sufficient to measure only two of these three pressures, since the third one can be obtained by simple addition or subtraction. Care must be taken, however, so that the signs of the various pressures are correct: compressive pressures are positive, suction pressures are negative.

The simplest case is that of an air stream which is blown through a duct. The hose connections for this case are as shown in Fig. 56A. The static pressure is positive, the total pressure is positive and larger than the static pressure, and the velocity pressure, which is always counted positive, is equal to the difference TP – SP.

If the air stream is exhausted from the duct, the static pressure is negative and the hose connections will depend on whether the velocity pressure is larger or smaller than the numerical value of the static pressure: if it is larger, the total pressure will be positive and the hose connections will be as shown in Fig. 56B; if it is smaller, the total pressure will be negative and the hose connections will be as shown in Fig. 56C.

The various connections between the Pitot tube and the draft gages are frequently made with rubber hose. Precaution must be taken so that all passages and connections are dry, clean and free of leaks and of bends and other obstructions. The branching out of the rubber hose can be accomplished by the use of a T-tube or by the use of a 2-stem nipple adapter which can be purchased as an accessory to the draft gage.

Calculation of the Flow Velocity

After the velocity pressure (VP) at a certain point of an air or gas stream has been measured, the corresponding velocity is calculated as

$$V = 1096.5 \sqrt{\frac{VP}{d}} \quad \ldots\ldots\ldots\ldots (A)$$

where V = feet per minute

VP = inches of water

d = the density of the test air (or gas) in lbs. per cubic foot

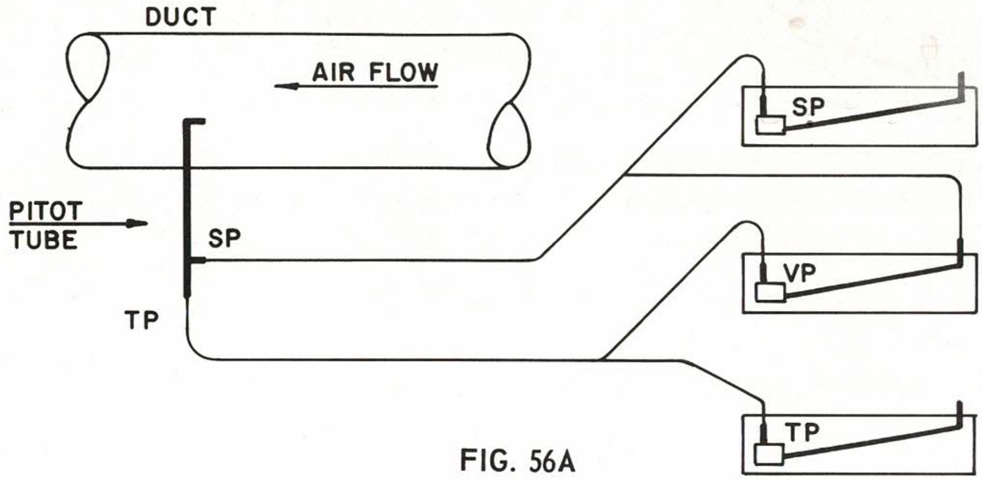


FIG. 56A

PITOT TUBE CONNECTIONS IF AIR STREAM IS BLOWN INTO DUCT

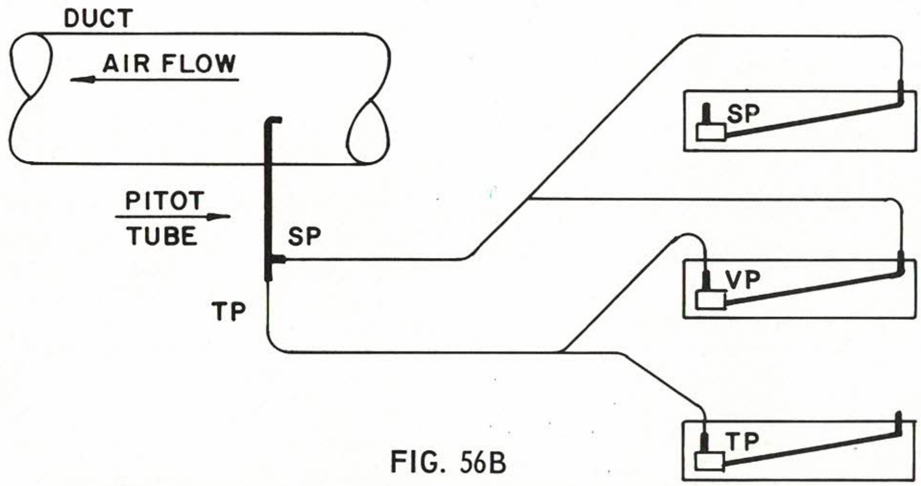


FIG. 56B

PITOT TUBE CONNECTIONS IF AIR STREAM IS EXHAUSTED FROM DUCT & TP IS POSITIVE

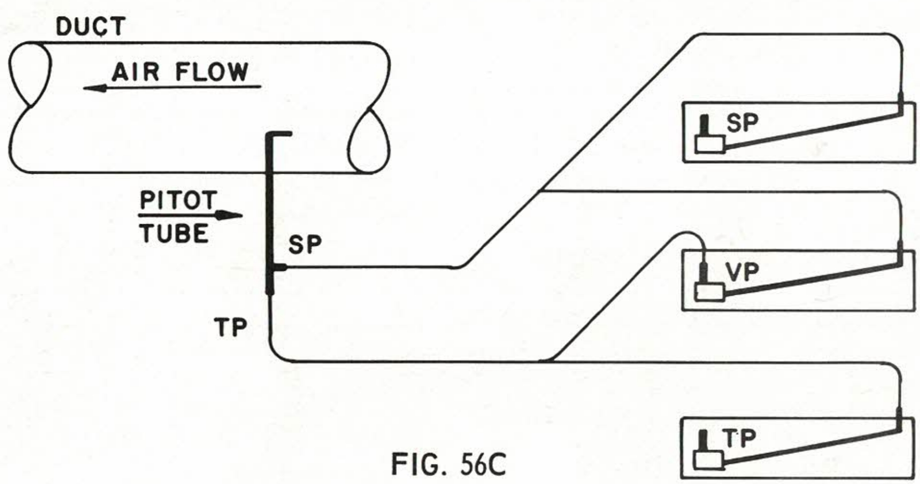


FIG. 56C

PITOT TUBE CONNECTIONS IF AIR STREAM IS EXHAUSTED FROM DUCT & TP IS NEGATIVE

Top – FIG. 56A; Center - FIG. 56B – Bottom - FIG. 56C

If the density d = 0.07495 lbs. per cubic foot (density of standard air), the formula (A) simplifies itself to

$$V - 4005 \sqrt{VP} \quad \ldots\ldots\ldots\ldots\ldots \text{(B)}$$

Examples for using formula (B)

For VP = in. W.G.,

we get $V = 4005 \times \sqrt{1} = 4005$ FPM

For VP = 1 in. W.G.,

we get $V = 4005 \times \sqrt{2} = 5664$ FPM

Instead of using formula (B), one may also use (Fig. 57) Table 1 which gives the velocities, calculated from formula (B), for a wide range of velocity pressures.

If the density of the test air is different from that of standard air, formula (A) has to be used. If for instance the density was found to be d = 0.070 lbs. per cubic foot and the velocity pressure was measured as 0.500 inches of water, the velocity will be

$$V = 1096.5 \times \sqrt{\frac{0.500}{0.070}} = 1096.5 \times \sqrt{7.143} =$$

$$= 1096.5 \times 2.673 = 2930 \text{ FPM}$$

The same result can also be obtained by the use of the charts shown on pages 73 and 74. This method is quicker, but less accurate and is, therefore, recommended particularly for fast estimates.

A third method to determine the velocity for non-standard air densities is as follows: First look up the velocity corresponding to the velocity pressure in Table 1 (Fig. 57) as if the density were that of standard air; then multiple that velocity by a correction factor, obtained from Table 2 (Fig. 57)

In the above example we get for the velocity pressure VP = 0.500 in. W.G. from Table 1, a corresponding velocity V = 2832 FPM. For the density d = 0.070 lbs. per cubic foot, we find in Table 2 a correction factor

R = 1.0348 and by simple multiplication we calculate the actual velocity as V = 2832 x 1.0348 = 2930 FPM.

Here is another example: Suppose the velocity pressure was VP = 0.105 in. W.G. and the density d = 0.568 lbs. per cubic foot. From Table 1 (Fig. 57) we get a velocity V = 1298 FPM, from Table 2 (Fig. 57) - by interpolation between 0.056 and 0.057 - a correction factor R = 1.1487; the actual velocity, therefore, if V = 1298 x 1.1487 = 1491 FPM. Chart on page 74 gives the same result.

Determination of the Flow Volume

The object of an actual test usually is not only to determine the flow velocity at a certain point, but to determine the volume flowing through a duct. This vol-

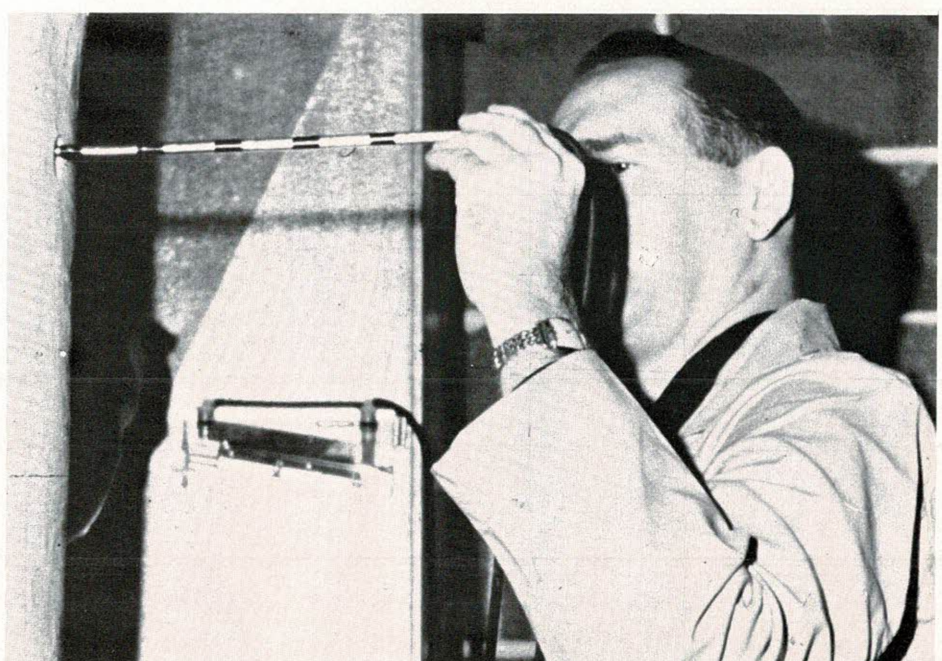


FIG. 55 A
To use the pitot tube the hoses are connected to an inclined draft gage and the pressures are read and noted.

Table No. 1. $V=4005\sqrt{vp}$ = feet per minute. vp = velocity pressure in inches of water, standard air density.

VP	V	VP	V	VP	V	VP	V	VP	V
.001"	127	.041	811	.081	1140	.121	1393	.161	1607
.002	179	.042	821	.082	1147	.122	1399	.162	1612
.003	219	.043	831	.083	1154	.123	1404	.163	1617
.004	253	.044	840	.084	1161	.124	1410	.164	1622
.005	283	.045	849	.085	1167	.125	1416	.165	1627
.006	310	.046	859	.086	1175	.126	1422	.166	1632
.007	335	.047	868	.087	1181	.127	1427	.167	1637
.008	358	.048	877	.088	1188	.128	1433	.168	1642
.009	380	.049	887	.089	1193	.129	1439	.169	1646
.010	400	.050	896	.090	1201	.130	1444	.170	1651
.011	420	.051	904	.091	1208	.131	1449	.171	1656
.012	439	.052	913	.092	1215	.132	1455	.172	1661
.013	457	.053	922	.093	1222	.133	1461	.173	1666
.014	474	.054	931	.094	1228	.134	1466	.174	1670
.015	491	.055	939	.095	1234	.135	1471	.175	1675
.016	507	.056	948	.096	1241	.136	1477	.176	1680
.017	522	.057	956	.097	1247	.137	1482	.177	1685
.018	537	.058	964	.098	1254	.138	1488	.178	1690
.019	552	.059	973	.099	1260	.139	1493	.179	1695
.020	566	.060	981	.100	1266	.140	1498	.180	1699
.021	580	.051	989	.101	1273	.141	1504	.181	1704
.022	594	.062	996	.102	1279	.142	1509	.182	1709
.023	607	.063	1004	.103	1285	.143	1515	.183	1713
.024	620	.064	1012	.104	1292	.144	1520	.184	1718
.025	633	065	1020	.105	1298	.145	1525	.185	1723
.026	645	.066	1029	.106	1304	.146	1530	.186	1727
.027	658	.067	1037	.107	1310	.147	1536	.187	1732
.028	670	.068	1045	.108	1316	.148	1541	.188	1737
.029	682	.069	1052	.109	1322	.149	1546	.189	1741
.030	694	.070	1060	110	1328	.150	1551	.190	1746
.031	705	.071	1067	.111	1334	.151	1556	.191	1750
.032	716	.072	1075	.112	1340	.152	1561	.192	1755
.033	727	.073	1082	.113	1346	.153	1567	.193	1759
.034	738	.074	1089	.114	1352	.154	1572	.194	1764
.035	749	.075	1097	.115	1358	.155	1577	.195	1768
.036	759	.076	1104	.116	1364	.156	1582	.196	1773
.037	770	.077	1111	.117	1370	.157	1587	.197	1777
.038	780	.078	1119	.118	1376	.158	1592	.198	1782
.039	791	.079	1125	.119	1382	.159	1597	.199	1787
.040	801	.080	1133	.120	1387	.160	1602	.200	1791

Continued in Next Column

TABLE No. 2. Correction Factors for Air Density to be Applied to Velocities from Table 1.

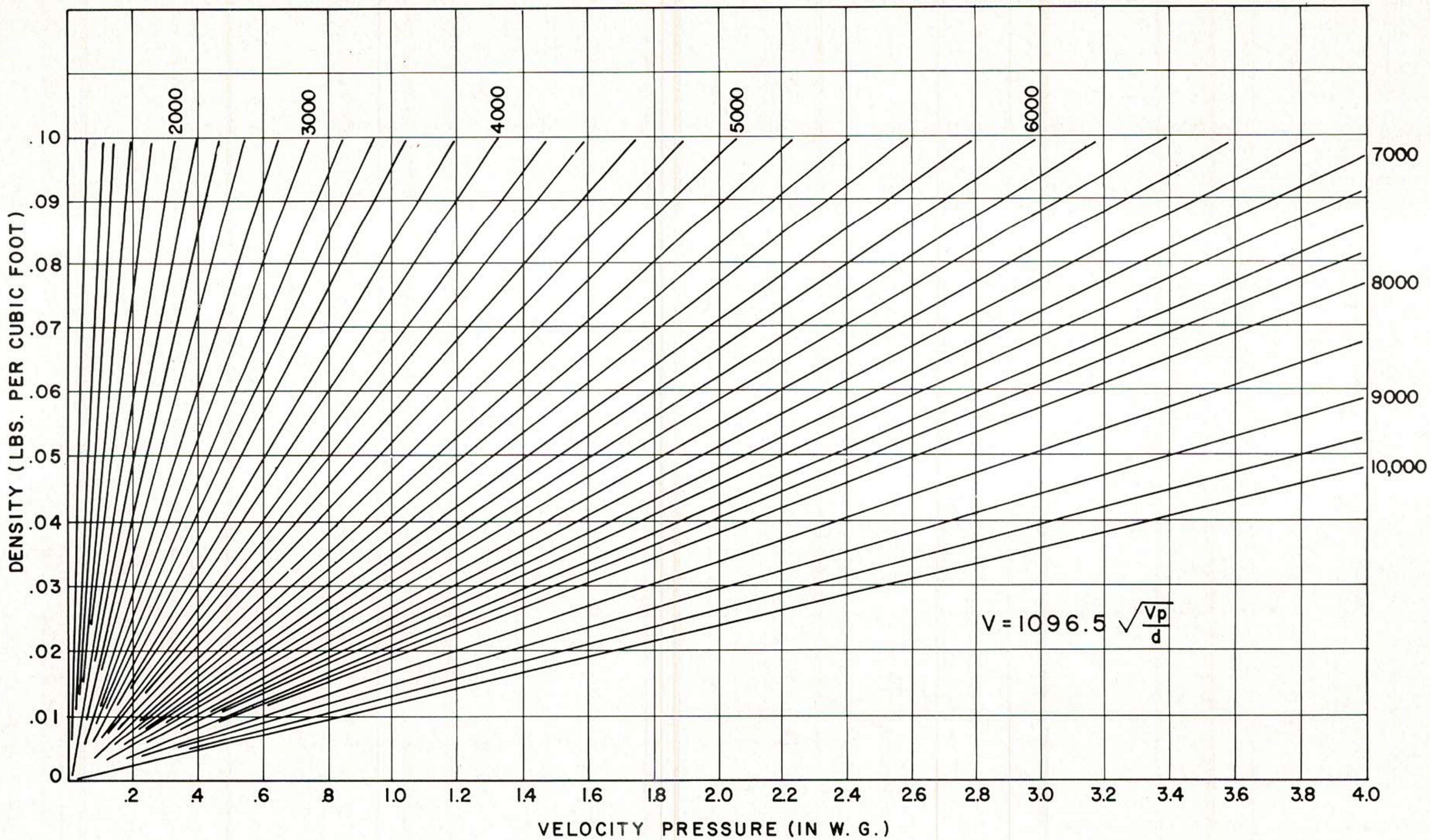
Density	Factor	Density	Factor	Density	Factor
0.010	2.7378	0.040	1.3689	0.070	1.0348
0.011	2.6104	0.041	1.3521	0.071	1.0275
0.012	2.4993	0.042	1.3359	0.072	1.0203
0.013	2.4012	0.043	1.3203	0.073	1.0133
0.014	2.3139	0.044	1.3052	0.074	1.0064
0.015	2.2354	0.045	1.2906	0.075	0.9997
0.016	2.1644	0.046	1.2765	0.076	0.9931
0.017	2.0998	0.047	1.2629	0.077	0.9866
0.018	2.0407	0.048	1.2496	0.078	0.9803
0.019	1.9862	0.049	1.2368	0.079	0.9741
0.020	1.9359	0.050	1.2244	0.080	0.9680
0.021	1.8893	0.051	1.2123	0.081	0.9620
0.022	1.8458	0.052	1.2006	0.082	0.9561
0.023	1.8053	0.053	1.1892	0.083	0.9503
0.024	1.7673	0.054	1.1782	0.084	0.9446
0.025	1.7316	0.055	1.1674	0.085	0.9391
0.026	1.6979	0.056	1.1569	0.086	0.9336
0.027	1.6662	0.057	1.1467	0.087	0.9282
0.028	1.6362	0.058	1.1368	0.088	0.9229
0.029	1.6077	0.059	1.1271	0.089	0.9177
0.030	1.5807	0.060	1.1177	0.090	0.9126
0.031	1.5550	0.061	1.1085	0.091	0.9076
0.032	1.5305	0.062	1.0995	0.092	0.9026
0.033	1.5071	0.063	1.0908	0.093	0.8978
0.034	1.4848	0.064	1.0822	0.094	0.8930
0.035	1.4634	0.065	1.0739	0.095	0.8883
0.036	1.4430	0.066	1.0657	0.096	0.8836
0.037	1.4233	0.067	1.0577	0.097	0.8791
0.038	1.4045	0.068	1.0499	0.098	0.8746
0.039	1.3864	0.069	1.0423	0.099	0.8701
0.040	1.3689	0.070	1.0348	0.100	0.8658

Table No. 1. $V=4005\sqrt{vp}$ = feet per minute. vp = velocity pressure in inches of water, standard air density.

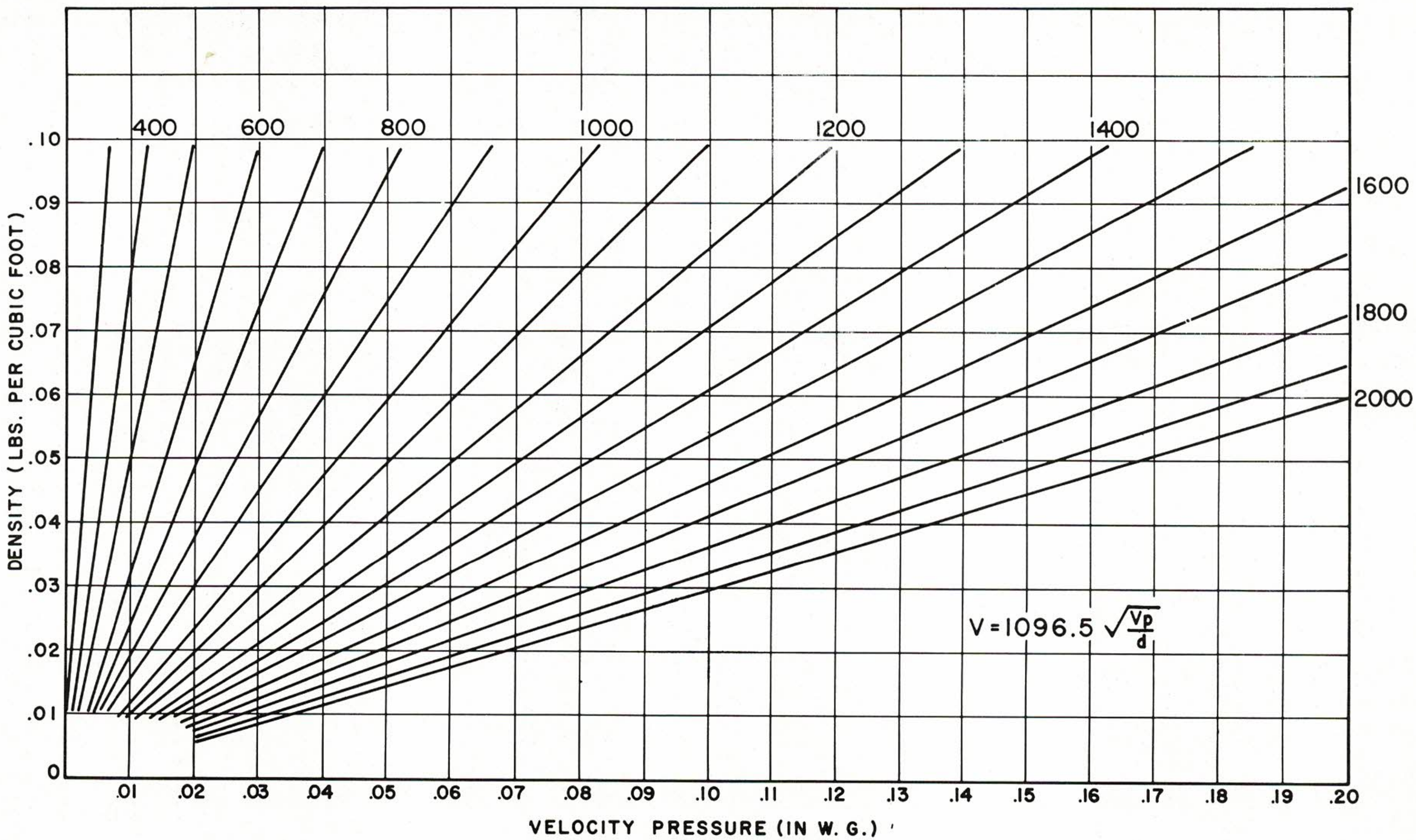
VP	V	VP	V	VP	V	VP	V	VP	V
.01	400.5	.81	3604	1.61	5082	2.41	6217	3.21	717
.02	566.4	.82	3625	1.62	5098	2.42	6230	3.22	718
.03	693.7	.83	3657	1.63	5114	2.43	6243	3.23	7198
.04	801.0	.84	3669	1.64	5129	2.44	6256	3.24	7209
.05	895.5	.85	3690	1.65	5144	2.45	6269	3.25	7220
.06	981	.86	3709	1.66	5160	2.46	6281	3.26	7231
.07	1060	.87	3729	1.67	5175	2.47	6294	3.27	7242
.08	1133	.88	3758	1.68	5191	2.48	6307	3.28	7253
.09	1201	.89	3779	1.69	5206	2.49	6319	3.29	7264
.10	1266	.90	3800	1.70	5222	2.50	6332	3.30	7275
.11	1328	.91	3821	1.71	5237	2.51	6345	3.31	7286
.12	1387	.92	3842	1.72	5253	2.52	6358	3.32	7297
.13	1444	.93	3863	1.73	5268	2.53	6370	3.33	7308
.14	1498	.94	3884	1.74	5283	2.54	6383	3.34	7319
.15	1551	.95	3904	1.75	5298	2.55	6395	3.35	7330
.16	1602	.96	3924	1.76	5313	2.56	6408	3.36	7341
.17	1651	.97	3945	1.77	5328	2.57	6420	3.37	7352
.18	1699	.98	3965	1.78	5343	2.58	6433	3.38	7363
.19	1746	.99	3985	1.79	5359	2.59	6445	3.39	7374
.20	1791	1.00	4005	1.80	5374	2.60	6458	3.40	7385
.21	1835	1.01	4025	1.81	5388	2.61	6470	3.41	7396
.22	1879	1.02	4045	1.82	5403	2.62	6482	3.42	7406
.23	1921	1.03	4064	1.83	5418	2.63	6495	3.43	7417
.24	1962	1.04	4084	1.84	5433	2.64	6507	3.44	7428
.25	2003	1.05	4103	1.85	5447	2.65	6519	3.45	7439
.26	2042	1.06	4123	1.86	5462	2.66	6532	3.46	7450
.27	2081	1.07	4142	1.87	5477	2.67	6544	3.47	7460
.28	2119	1.08	4162	1.88	5491	2.68	6556	3.48	7471
.29	2157	1.09	4181	1.89	5506	2.69	6569	3.49	7482
.30	2193	1.10	4200	1.90	5521	2.70	6581	3.50	7493
.31	2230	1.11	4219	1.91	5535	2.71	6593	3.51	7503
.32	2260	1.12	4238	1.92	5550	2.72	6605	3.52	7514
.33	2301	1.13	4257	1.93	5564	2.73	6617	3.53	7525
.34	2335	1.14	4276	1.94	5579	2.74	6629	3.54	7535
.35	2369	1.15	4295	1.95	5593	2.75	6641	3.55	7546
.36	2403	1.16	4314	1.96	5608	2.76	6654	3.56	7556
.37	2436	1.17	4332	1.97	5623	2.77	6666	3.57	7567
.38	2469	1.18	4350	1.98	5637	2.78	6678	3.58	757
.39	2501	1.19	4368	1.99	5651	2.79	6690	3.59	758
.40	2533	1.20	4386	2.00	5664	2.80	6702	3.60	7599
.41	2563	1.21	4405	2.01	5678	2.81	6714	3.61	7610
.42	2595	1.22	4423	2.02	5692	2.82	6725	3.62	7620
.43	2626	1.23	4442	2.03	5706	2.83	6737	3.63	7630
.44	2656	1.24	4460	2.04	5720	2.84	6749	3.64	7641
.45	2687	1.25	4478	2.05	5734	2.85	6761	3.65	7652
.46	2716	1.26	4495	2.06	5748	2.86	6773	3.66	7662
.47	2746	1.27	4513	2.07	5762	2.87	6785	3.67	7672
.48	2775	1.28	4531	2.08	5776	2.88	6797	3.68	7683
.49	2804	1.29	4549	2.09	5790	2.89	6809	3.69	7693
.50	2832	1.30	4566	2.10	5804	2.90	6820	3.70	7704
.51	2860	1.31	4583	2.11	5817	2.91	6832	3.71	7714
.52	2888	1.32	4601	2.12	5831	2.92	6844	3.72	7724
.53	2916	1.33	4619	2.13	5845	2.93	6855	3.73	7735
.54	2943	1.34	4636	2.14	5859	2.94	6867	3.74	7745
.55	2970	1.35	4653	2.15	5872	2.95	6879	3.75	7755
.56	2997	1.36	4671	2.16	5886	2.96	6890	3.76	7766
.57	3024	1.37	4688	2.17	5899	2.97	6902	3.77	7776
.58	3050	1.38	4705	2.18	5913	2.98	6913	3.78	7787
.59	3076	1.39	4722	2.19	5927	2.99	6925	3.79	7797
.60	3102	1.40	4739	2.20	5940	3.00	6937	3.80	7807
.61	3127	1.41	4756	2.21	5954	3.01	6948	3.81	7817
.62	3153	1.42	4773	2.22	5967	3.02	6960	3.82	7827
.63	3179	1.43	4790	2.23	5981	3.03	6971	3.83	7838
.64	3204	1.44	4806	2.24	5994	3.04	6983	3.84	7848
.65	3229	1.45	4823	2.25	6008	3.05	6994	3.85	7858
.66	3254	1.46	4840	2.26	6021	3.06	7006	3.86	7868
.67	3279	1.47	4856	2.27	6034	3.07	7017	3.87	7879
.68	3303	1.48	4873	2.28	6047	3.08	7028	3.88	7889
.69	3327	1.49	4889	2.29	6060	3.09	7040	3.89	7899
.70	3351	1.50	4905	2.30	6074	3.10	7051	3.90	7909
.71	3375	1.51	4921	2.31	6087	3.11	7063	3.91	7919
.72	3398	1.52	4938	2.32	6100	3.12	7074	3.92	7929
.73	3422	1.53	4954	2.33	6113	3.13	7085	3.93	7940
.74	3445	1.54	4970	2.34	6126	3.14	7097	3.94	7950
.75	3468	1.55	4986	2.35	6139	3.15	7108	3.95	796
.76	3491	1.56	5002	2.36	6152	3.16	7119	3.96	797
.77	3514	1.57	5018	2.37	6165	3.17	7131	3.97	7980
.78	3537	1.58	5034	2.38	6179	3.18	7142	3.98	7990
.79	3560	1.59	5050	2.39	6191	3.19	7153	3.99	8000
.80	3582	1.60	5066	2.40	6204	3.20	7164	4.00	8010

FIG. 57

Tables for the determination of air velocity when the air handled has a density different from "standard" air. See text for problem solutions.



Velocity as function of velocity pressure and air or gas density for velocity range from 1,000 to 10,000 fpm.



Velocity as function of velocity pressure and air or gas density for velocity range from 300 to 2,000 fpm.

TABLE NO. 3

Pipe diam.	Readings in one diam.	Distances of Pitot Tube Tip From Pipe Center Point 1	Point 2	Point 3	Point 4	Point 5
3 in.	6	.612"	1.061"	1.369"		
4 in.	6	.812"	1.414"	1.826"		
5 in.	6	1.021"	1.768"	2.282"		
6 in.	6	1.225"	2.121"	2.738"		
7 in.	6	1.429"	2.475"	3.195"		
8 in.	6	1.633"	2.828"	3.651"		
9 in.	6	1.837"	3.182"	4.108"		
10 in.	8	1.768"	3.062"	3.950"	4.677"	
12 in.	8	2.122"	3.674"	4.740"	5.612"	
14 in.	10	2.214"	3.834"	4.950"	5.857"	6.641"
16 in.	10	2.530"	4.382"	5.657"	6.693"	7.589"
18 in.	10	2.846"	4.929"	6.364"	7.530"	8.538"
20 in.	10	3.162"	5.477"	7.071"	8.367"	9.487"
22 in.	10	3.479"	6.025"	7.778"	9.203"	10.435"
24 in.	10	3.795"	6.573"	8.485"	10.040"	11.384"
26 in.	10	4.111"	7.120"	9.192"	10.877"	12.333"
28 in.	10	4.427"	7.668"	9.900"	11.713"	13.282"
30 in.	10	4.743"	8.216"	10.607"	12.550"	14.230"
32 in.	10	5.060"	8.764"	11.314"	13.387"	15.179"
34 in.	10	5.376"	9.311"	12.021"	14.223"	16.128"
36 in.	10	5.692"	9.859"	12.728"	15.060"	17.176"

ume in cubic feet per minute is calculated as the product of the cross sectional area of the duct in square feet, at the point of measurement, multiplied by the average flow velocity in FPM through that cross section.

In the case of straight, smooth air flow through a small, round duct (up to about 1 foot in diameter), the average velocity is approximately equal to 91% of the center velocity.

To find the average velocity in larger ducts, however, it is necessary to take observations at several points in the duct, so located that each represents an equal area across the stream. This is known as making a traverse. In this case it is necessary for accuracy to find by means of Table No. 1 (Fig. 57) the velocity corresponding to each individual reading. These should then be averaged to give the mean velocity in the duct.

Where accuracy is not essential it is permissable to average the observed velocity heads and then by a single reference to table No. 1 obtain the corresponding mean velocity head. This latter method will invariably give results that are slightly too high, the error amounting to from .1% to 1.0%. The greatest error will occur when there is the greatest variation in the values of the maximum and minimum observed readings. When the various observed values are very nearly alike it is always permissable to use the second method.

Traverse

The Pitot tube should be parallel to the duct and the measurements should be taken where the air flow is straight, relatively free from turbulence, and where the area of duct is accurately known. The most satisfac-factory readings are in long, straight circular ducts. The point of readings are usually taken at not less than 7-½ diameters from fan connection outlet or pipe bend fittings. When using a straightener, codes recommend 6 duct diameters from fan outlet connection or pipe bend fittings to outlet side of straightener, and 1-½ diameters between straightener and tip of Pitot tube facing the flow.

In open end inlet ducts, straight flow, the point of readings are usually taken at 7-½ diameters from the end of the duct.

TABLE A

Station	V.P.	fpm	Station	V.P.	fpm
5	.43''	2626	1	.64''	3204
4	.51''	2860	2	.62''	3153
3	.56''	2997	3	.57''	3024
2	.60''	3102	4	.53''	2916
1	.62''	3153	5	.45''	2687

TABLE NO. 4

Readings in one Diameter	Constants To Be Multiplied By Pipe Diameter For Distances of Pitot Tube Tip From Pipe Center				
	P. 1	P. 2	P. 3	P. 4	P. 5
6	.2041	.3535	.4564		
8	.1768	.3062	.3953	.4677	
10	.1581	.2738	.3535	.4183	.4743

Fig. 58 shows the locations for Pitot tube tip making a 10-point traverse across one circular pipe diameter. In making traverse across two pipe diameters, readings are taken at right angles to each other. The traverse points shown represent 5 annular zones of equal area.

To measure the volume of air flowing for instance in a 20'' pipe, Table No. 3 shows that 10 readings across the diameter are recommended. Distances of point one from both sides of the center is 3.162''; point two 5.477''; point three 7.071''; point four 8.367''; point five 9.487''. Fig. 58 shows how the readings are to be taken across the pipe, one reading on each side of center for each of the distances indicated. Starting the traverse on one side, velocity readings are taken across the pipe at points 5, 4, 3, 2, 1; 1, 2, 3, 4, 5 in the order shown. Assume readings to be as shown in Table A above.

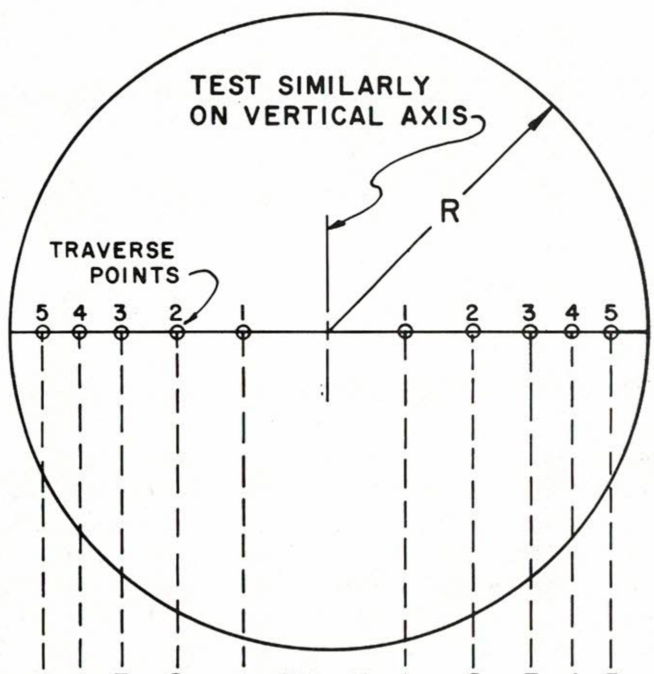


FIG. 58

Locations of the Pitot tube in making a ten point traverse in a round duct. The points indicate five annular zones of equal area.

The velocity for each of the 10 VP readings are obtained from Table No. 1 (Fig. 57) and the average velocity is 2972 fpm.

For distances of traverse points from pipe center for pipe diameters other than those given in Table No. 3, use constants in Table No. 4.

Example: Assume pipe diameter 20.5 in., number of readings 10. Point 1 = 20.5 x .1581 = 3.241 in.; point 2= 20.5 x .2738 = 5.612 in., etc.

The average velocity in square or rectangular ducts can be determined by making a traverse of the center points of not less than 16 equal areas. See Fig. 59 Larger ducts will require more than 16 equal areas.

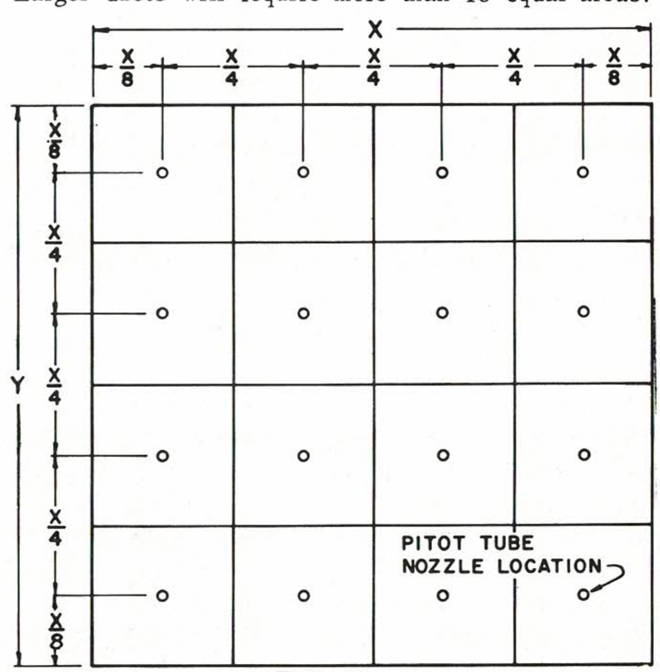


FIG. 59

For rectangular duct, 16 points of reading are recommended as shown. The number of reading points required will depend on the size and shape of the duct – the several velocities should be accounted for.

CHAPTER 20

MICROMANOMETERS
(Hook Gages)

These instruments are designed to read small differences in air pressure accurately and they usually have a wide scale range. Most scales read 0 in. to 4.0 in. H_2O in hundredths of an inch on the vertical scale and in thousandths of an inch on a vernier scale.

There is more than one variation of this instrument. The most common type contains two glass vials about 2 or 3 inches in diameter. A pointed needle or hook is positioned by a micrometer until the point dimples the water surface but does not break the surface tension. The difference in level is determined by the difference in micrometer readings.

Another variation of this instrument has a single vial or well and an inclined scale. The well is positioned by a micrometer or vernier adjustment. It is very important that all micromanometers be accurately levelled before and after each reading.

These instruments are particularly well suited for readings at hoods, perforated ceilings, etc. They may also be used to calibrate other instruments.

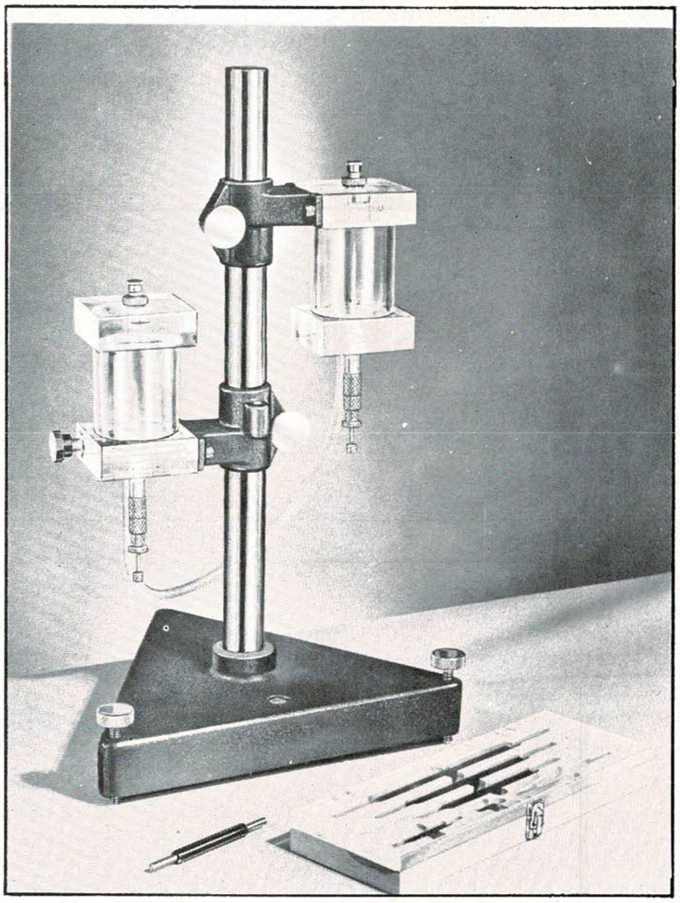


FIG. 60

For very precise differences in air pressure the micromanometer will read in hundredths and thousandths of an inch.

CHAPTER 21
ANEMOMETERS

Method For Determining The Outlet Flow Factor

The manufacturers of air diffusers have equipment which provides a means for measuring the total air flow to an outlet and means for determining the factor which must be applied to velocity measurements made at the outlet face with several types of instruments.

Fig. 61 shows the arrangement of equipment necessary to determine the flow factors for an individual outlet. The basic equipment includes a fan, flow straighteners, the flow measuring station, such as a nozzle, the outlet itself and the velocity measuring instrument.

Air flow through any system is defined by the continuity equation which is:

$$Q = AV$$

where Q = the air flow in CFM
A = the flow area, sq. ft.
V = the flow area velocity, fpm

In the system shown in Fig. 61 the total air delivered by the fan is measured with the nozzle and this total quantity of air also flows through the grille or diffuser mounted on the end of the duct. Therefore, since the air flow is measured with the orifice and the velocity at the grille face is measured with the Anemometer, an equivalent area for the grille or diffuser may be determined by substitution in the continuity equation. This area is known as the K-factor area and is determined from the equation:

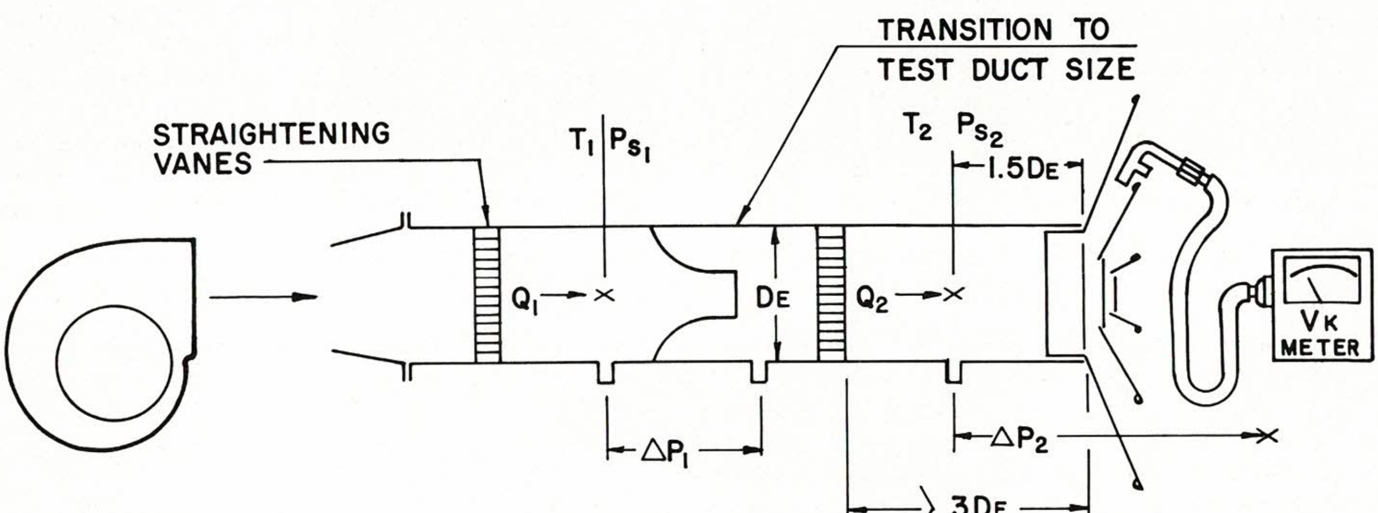
$$A_K = \frac{Q}{V}$$

where Q = the airflow in CFM
V = the flow area velocity in fpm
A_K = the flow factor area in sq. ft.

For simplicity the factor K is substituted for the area A_K. Thus, the K-factor used to measure air flow at the grille face is in reality a flow area. This area does not have any physical measurements but must be related to the type of instrument that is used. *As will be shown later the same grille will have different factors depending on the type of instrument that is used. However, the diffuser manufacturer can determine the proper factor for each grille and each type of instrument.*

To properly determine the K-factor, the manufacturer will measure the velocity at the grille face at several different air flow rates. The results of these tests are then plotted on a chart as shown in Fig. 62. The horizontal axis is the measured air flow rate in CFM and the vertical axis is the velocity in fpm.

The curve shown in Fig. 62 was drawn through the points which are indicated by the triangular symbols



TYPICAL AIR OUTLET TEST SETUP
FOR DETERMINATION OF A_K, V_K

FIG. 61

Diagrammatic layout of typical test equipment used to determine the flow factors for an individual outlet.

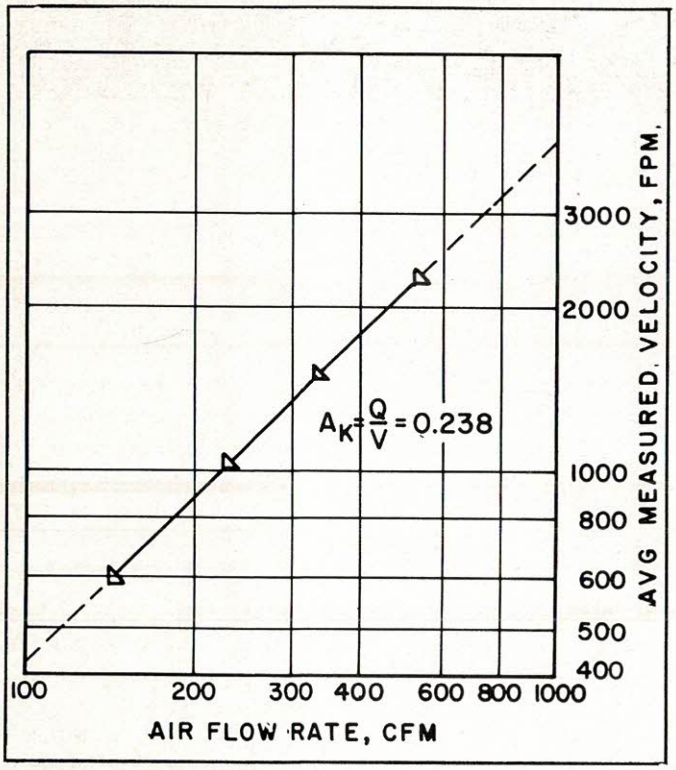


FIG. 62

Typical K factor test curve

that lie on the curve. The K-factor can be determined from the intersection of the curve with the 1000 fpm line by substituting in the K-factor equation the values for the CFM and the 1000 fpm velocity. For this particular case the K-factor is 0.238.

The manufacturer must test each type of supply outlet and return inlet to determine the K-factors for all of the different types of instruments. The manufacturer must also determine the location on the outlet face for making the measurements.

In the case of high sidewall outlets or linear grilles it is usually necessary to average several readings over the face of the grille. With ceiling diffusers the manufacturer will determine the location on the diffuser that gives the correct reading for a given air flow rate. It is important that the man in the field follow the manufacturer's instruction as to *where* the probe should be positioned on the particular supply outlet and the number of readings that are required for that particular outlet. *Care should also be taken to apply the proper factor to the type of instrument that is used.*

TYPES OF ANEMOMETERS

An anemometer is a velocity measuring device. This discussion will be limited to those which are commonly used in measuring the air flow at supply outlets and return inlets.

One of the most common velocity measuring instruments is the Alnor velometer which is a swinging vane anemometer. Inside the air tight case is an aluminum vane which deflects the pointer on the scale in proportion to the air velocity at the probe. Air flows through the probe and the connecting tube into the case and then through the channel which contains the aluminum vane. Fig. 63

There are several probes that may be used with this instrument and there is a corresponding scale for each probe. The probe, the connection tube and the flow passage through the instrument make up an air flow system. The instrument has been calibrated for this particular system and therefore it is important that the length of the connecting tube not be changed. Also, another tube should not be substituted for the tube supplied with the instrument.

This instrument may be used for measurements of air flow through supply outlets and return inlets provided that the proper jet is used and the proper flow factor applied. The instrument may also be used without a probe for measuring some of the lower velocities where the instrument case itself is placed in the air stream and air flows through the screened inlet.

The Bacharach instruments shown in Fig. 64 have scales which read directly in fpm. With earlier types of instruments it was necessary to use a stop watch to time the speed of rotation of the propeller which was indicated on the scale. The instruments shown can be used to measure the air flow from grilles by determining the average velocity over the face or may be used with other types of diffusers with a special attachment that directs the jet of air through the instrument. The instrument may be used with an extension handle to reach high sidewall outlets and is provided with a scale lock which locks the indicator in place once the reading is made.

A second type of anemometer (Fig. 65) is the rotating vane instrument. This anemometer has a small lightweight propeller which rotates with increasing speed as the air flows through the instrument. The instrument is calibrated in feet and has to be used with a timing instrument to determine velocity.

It is important to remember that the instrument comprises an air flow system and that it is important to position the instrument correctly and to apply a proper K-factor to determine the actual quantity of air flowing through the outlet.

There is considerable controversy on the use of this instrument where air is supplied or exhausted from a grille; the position of the rotating vane anemometer to

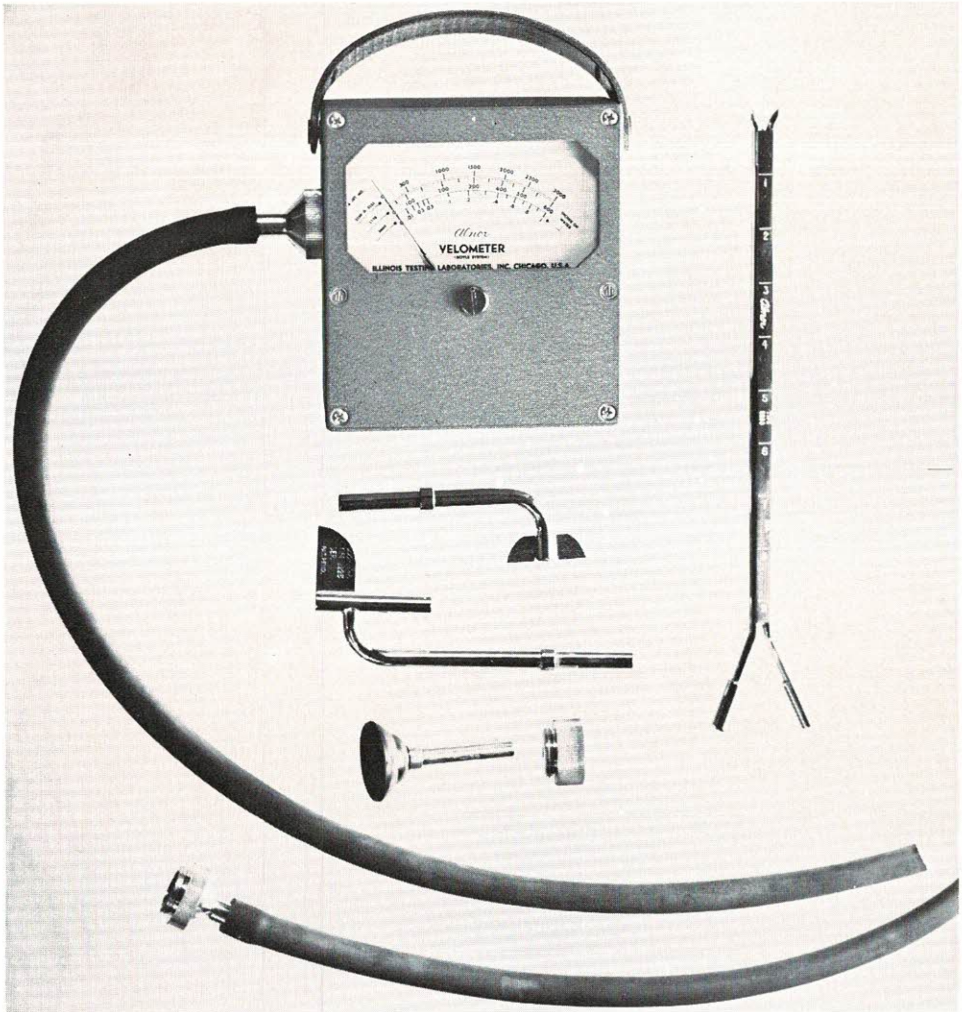
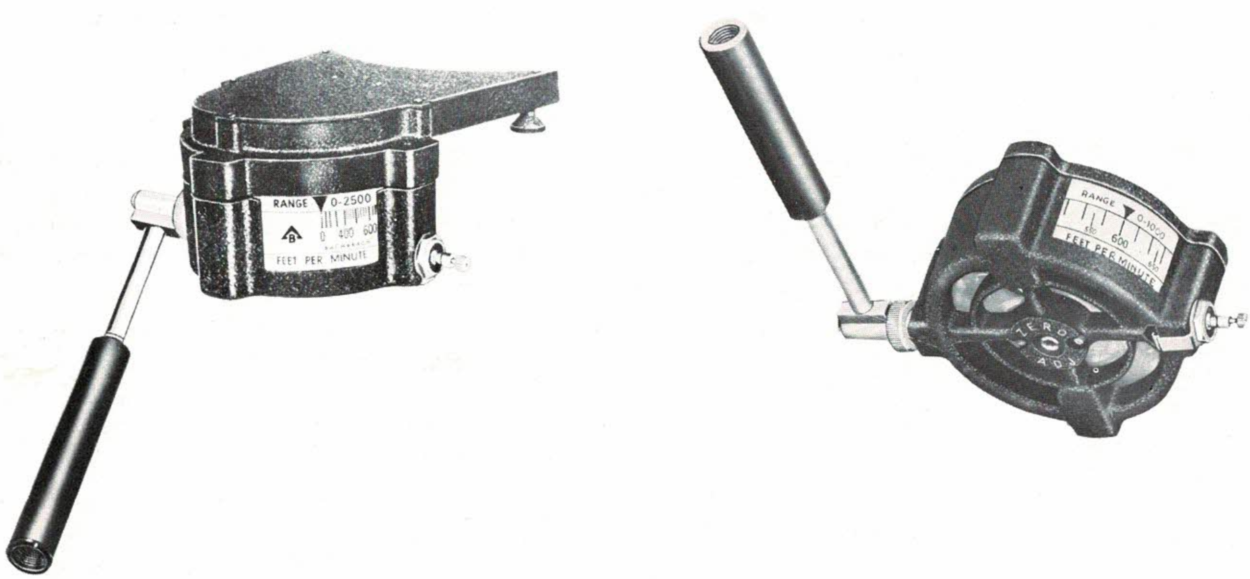


FIG. 63
Swinging vane anemometer. Air flow through probe and tube and the velocity is registered on the dial.

FIG. 64
Two types of direct reading fpm meters. The instrument can be used to read directly on the face of an outlet or used with a special attachment to read fpm for a diffuser.



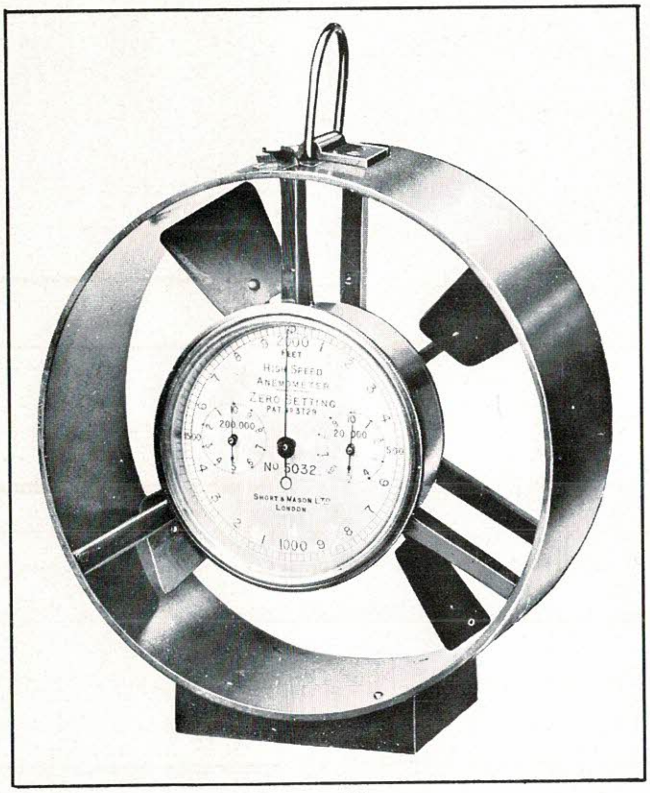


FIG. 65
Rotating vane anemometer calibrated in feet and must be used with a timing instrument to determine velocity.

the grille; and the resulting effect on the K-factor. The American Conference of Governmental Industrial Hygienists' - "Industrial Ventilation Manual" states that there is a difference as shown in the TABLE B, below.

If the opening is covered with a grille, the instrument should touch the grille face but should not be pushed in between the bars. For a free opening without a grille, the anemometer should be held in the plane of the entrance edges of the opening. The anemometer must always be held in such a manner that the air flow through the instrument is the same direction as was used for calibration (usually from the back toward the dial face).

This points up the importance of following the outlet manufacturer's recommendations very carefully when using this instrument.

The rotating vane anemometer has caused controversy as to accuracy between fixed stationary readings versus a traveling averaging time reading. This does not appear to be a large error.

A third type of anemometer is the "hot wire" anemometer. The operation of this instrument depends on the fact that the resistance of a heated wire will change with its temperature. Typical of these instruments is the Anemotherm Air Meter shown in Fig. 66 and Fig. 67.

The probe of this instrument (Fig. 67) is provided with a special type of wire element which is supplied with current from batteries contained in the instrument case. As air flows over the element in the probe the temperature of the element is changed from that which exists in still air and the resistance change is indicated as a velocity on the indicating scale of the instrument.

There are several scales on the instrument and each of these has a red line marked on it to which the pointer must be adjusted when the check button is pushed. Once the instrument is adjusted for a particular scale it can only be used on that scale range. If the velocity to be measured is greater than the scale range the next higher scale range must be selected and the instrument adjusted for that range.

The probe that is used with this instrument (Fig. 67) is quite directional and must be located at the proper point on the diffuser or grille as indicated by the manufacturer.

The instrument is also provided with temperature scales that can be utilized simply by setting the proper selector button. Static pressures can be measured if the proper cap is placed over the probe.

Another type of anemometer is shown in Fig. 68. This is also a flow-through type instrument and is actually

TABLE B

Opening	Correction Factor*
Pressure openings, more than 4 in. wide, up to 600 sq. in. area, with free opening 70% or more of gross area, no directional vanes	1.03
Suction opening, more than 4 in. wide, up to 600 sq. in. area, with flange 2 in. wide, free-open area 60% or more of gross area	0.85

Volume: For suction openings, cfm = (factor) (velocity) gross area)

For pressure openings, cfm = (factor) (velocity) $\left(\frac{\text{gross area} + \text{net area}}{2}\right)$

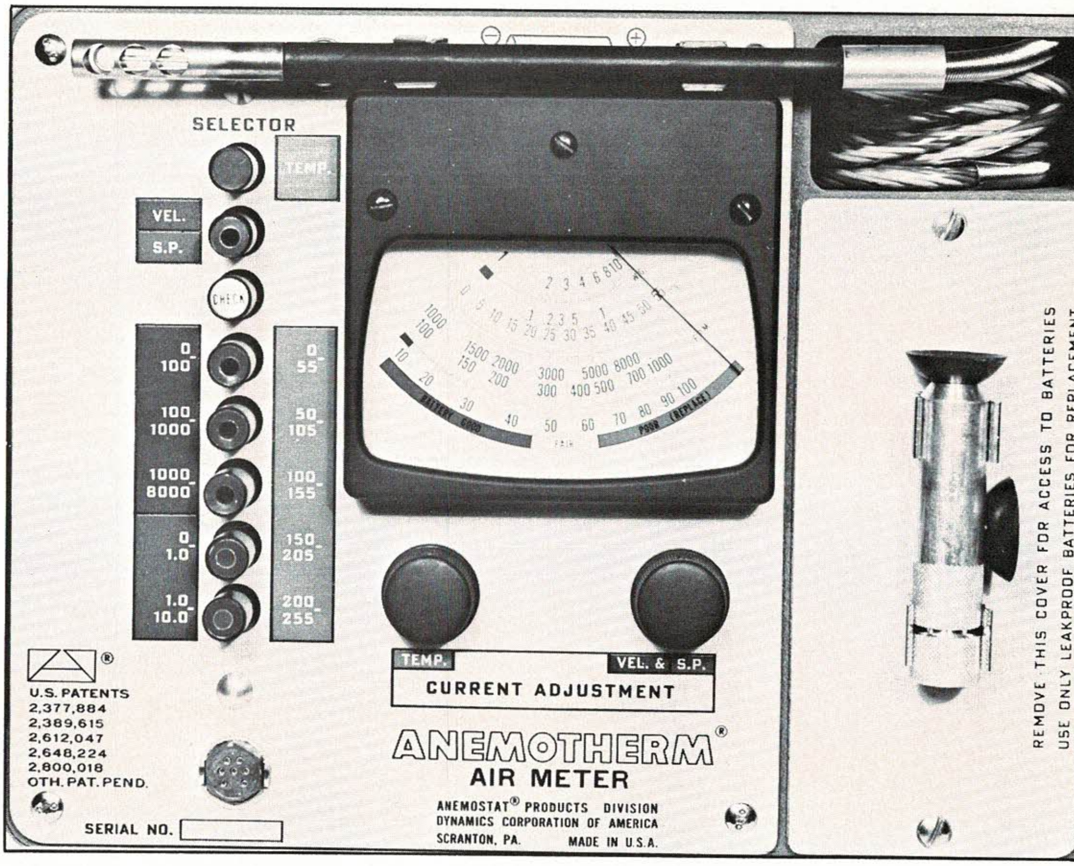
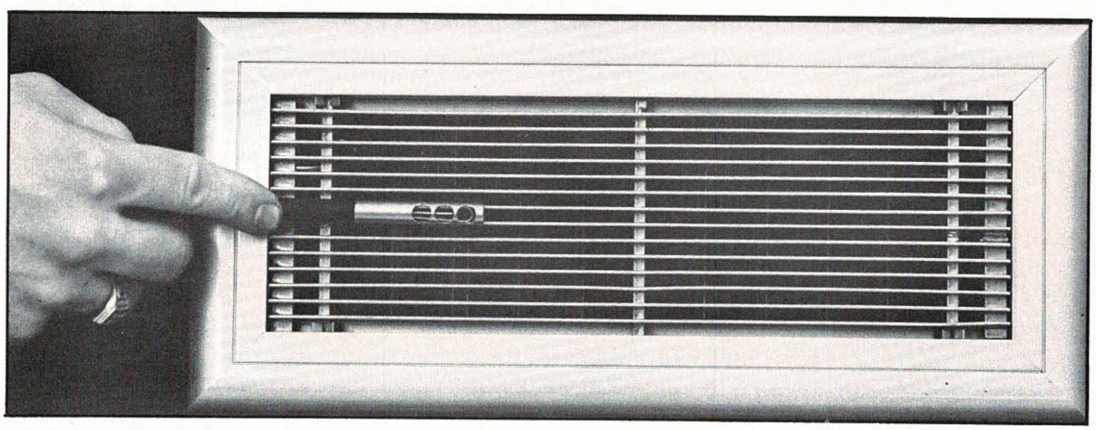


FIG. 66
"Hot wire" anemometer. The several scales read in velocities.

FIG. 67
The probe of the "hot wire" anemometer is quite directional and must be used in the precise location recommended by the outlet manufacturer.



a variable area flow meter. Air flows through the probe, the connecting tubing and then through the instrument. A plastic ball enclosed within the instrument tube is supported by the air stream and rises up in the tube as the air velocity increases. The instrument tube itself is tapered and the flow area varies as the plastic ball rises in the tube. The probe, the tube and the instrument case comprise a complete air flow system and the instrument has been calibrated as a system. Therefore, it is important that the length of the connecting tube not be changed and that the proper probe be used with the instrument.

This instrument is low in initial cost and also has the advantage that it is light weight and quite portable. Some of the air diffuser manufacturers have provided flow factors for use with it.

Calibrating Instruments

All anemometers are shipped from the manufacturer as calibrated units, but they will not remain in calibration indefinitely. Therefore, it is recommended that the contractor have available a means for checking the accuracy of the instruments either periodically or before each balancing job is undertaken.

The Air Diffusion Council members have developed a velocity meter test stand which is shown in Fig. 69. This is a small self-contained unit which has its own blower and only requires a simple draft gage which is connected to the air flow measuring station

FIG. 68

This anemometer "system" requires a matched probe, tube and instrument. Outlet manufacturers must provide the K-factors for this instrument.

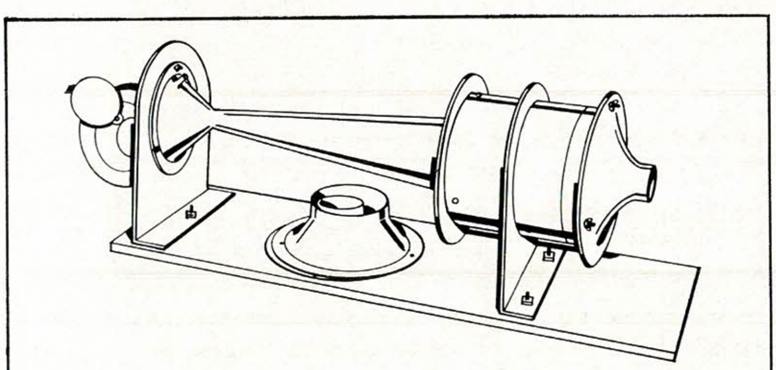
contained in the test stand. Several nozzles may be attached to the meter depending on the velocity that is to be maintained at the nozzle face.

Application of the Anemometer to Air Flow Measurements

Correct air flow measurements are dependent on the proper positioning of the probe of the anemometer as indicated by the air diffuser manufacturer's literature.

Fig. 70 shows the use of an Alnor Velometer to determine the air flow from a linear grille. For this application the manufacturer has shown the proper positioning of the probe to obtain an average velocity. The velocity multiplied by the flow factor and the length of the section then indicates the total CFM for that section of the grille. Notice that the chart included in Fig. 70 lists factors for both supply and return for several different nominal widths of the grille. The factor varies not only with the width but with the angle of the vanes in the grille. The factors apply only to the particular instrument that is mentioned.

FIG. 69
Velocity meter test stand developed by Air Diffusion Council with its own blower and nozzles for connection to a draft gage.



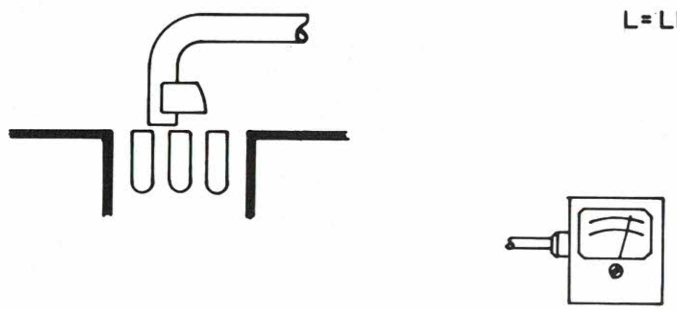
AIR FLOW MEASUREMENTS:

1. DETERMINE THE AVERAGE VELOCITY AT THE FACE OF THE GRILLE WITH THE ALNOR 2220-A OR 2220 PROBE POSITIONED AS SHOWN.

2. FOR LISTED SIZES: CFM = (FLOW FACTOR) VL

V = AVERAGE MEASURED VELOCITY, fpm

L = LENGTH OF SECTION, FT.



FLOW FACTOR, FT.^{2}/FT. OF LENGTH

MODEL	LISTED SIZE	$1\frac{1}{2}$"	2"	$2\frac{1}{2}$"	3"	$3\frac{1}{2}$"	4"	5"	6"	FLOW FACTOR COEF. *
C-1500	SUPPLY	.027	.043	.059	.078	.098	.116	.152	.190	.036
$\frac{1}{4}$" O.C. – 0°	RETURN	.023	.036	.056	.067	.082	.100	.128	.160	.030
C-1600	SUPPLY	.031	.044	.059	.072	.087	.105	.135	.165	.032
$\frac{1}{4}$" O.C. –15°	RETURN	.028	.039	.051	.062	.073	.089	.115	.140	.028
CM-1500	SUPPLY	.030	.049	.067	.089	.111	.132	.173	.216	.041
$\frac{1}{2}$" O.C. – 0°	RETURN	.026	.041	.064	.077	.094	.114	.146	.182	.035
CM-1600	SUPPLY	.035	.050	.067	.082	.099	.120	.154	.188	.037
$\frac{1}{2}$" O.C. –15°	RETURN	.031	.045	.058	.071	.083	.102	.131	.160	.031
C-2500	SUPPLY	.030	.049	.071	.095	.114	.135	.180	.225	.043
$\frac{1}{2}$" O.C. – 0°	RETURN	.025	.042	.060	.080	.098	.115	.152	.193	.037
C-2600	SUPPLY	.032	.048	.066	.085	.106	.125	.165	.210	.040
$\frac{1}{2}$" O.C. –15°	RETURN	.028	.042	.058	.073	.090	.105	.140	.180	.034

* FOR LISTED SIZES GREATER THAN 6", THE FLOW FACTOR IN FT. /FT. OF LENGTH MAY BE CALCULATED BY THE FOLLOWING EQUATION:

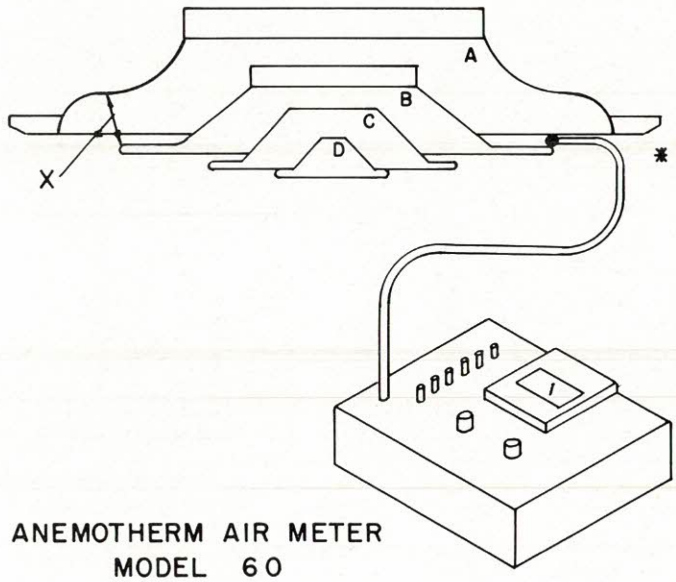
$$\frac{\text{FLOW FACTOR}}{\text{FT. OF LENGTH}} = (\text{LISTED WIDTH, IN.} - 0.75) \times \text{FLOW FACTOR COEF.}$$

THESE FACTORS APPLY TO ALL ABOVE DIFFUSERS AS MANUFACTURED BY TITUS MFG. CORP. WITH THE FOLLOWING SUFFIXES; .06, .07, 30, 35, & 40

FIG. 70

Diagramatic procedure for determining the air flow from a specific linear grille. Location of probe is specified by manufacturer. The flow factors apply only to the specified velometer.

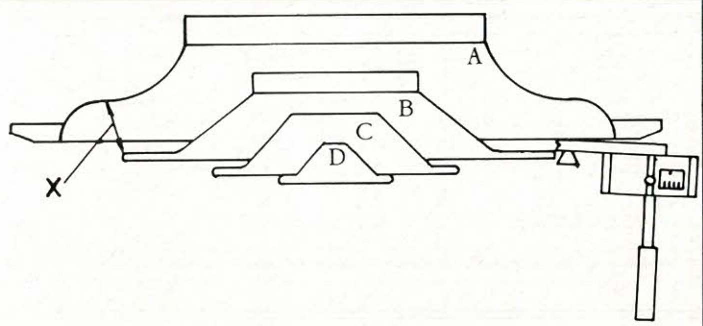
NECK DIAM. IN	CONES X	UP FACTOR	CONES X	UP FACTOR
6	$\frac{5}{8}$	0.18	$1\frac{5}{16}$	0.22
8	$\frac{13}{16}$	0.32	$1\frac{3}{4}$	0.38
10	1	0.50	$2\frac{3}{16}$	0.60
12	$1\frac{3}{16}$	0.72	$2\frac{5}{8}$	0.86
14	$1\frac{3}{8}$	0.98	$3\frac{1}{16}$	1.15
16	$1\frac{5}{8}$	1.25	$3\frac{1}{2}$	1.50
18	$1\frac{13}{16}$	1.60	$3\frac{15}{16}$	1.95
20	2	2.00	$4\frac{3}{8}$	2.40
24	$2\frac{3}{8}$	2.90	$5\frac{1}{4}$	3.40
30	3	4.50	$6\frac{5}{8}$	5.40
36	3	5.15	$6\frac{5}{8}$	6.30



ANEMOTHERM AIR METER
MODEL 60

* ROTATE PROBE FOR MAXIMUM VELOCITY

NECK DIAM. IN	CONES X	UP FACTOR	CONES X	UP FACTOR
6	$\frac{5}{8}$	0.12	$1\frac{5}{16}$	0.15
8	$\frac{13}{16}$	0.20	$1\frac{3}{4}$	0.27
10	1	0.32	$2\frac{3}{16}$	0.42
12	$1\frac{3}{16}$	0.45	$2\frac{5}{8}$	0.59
14	$1\frac{3}{8}$	0.61	$3\frac{1}{16}$	0.80
16	$1\frac{5}{8}$	0.79	$3\frac{1}{2}$	1.05
18	$1\frac{13}{16}$	1.00	$3\frac{15}{16}$	1.30
20	2	1.25	$4\frac{3}{8}$	1.63
24	$2\frac{3}{8}$	1.80	$5\frac{1}{4}$	2.35
30	3	2.80	$6\frac{5}{8}$	3.60
36	3	3.18	$6\frac{5}{8}$	4.15

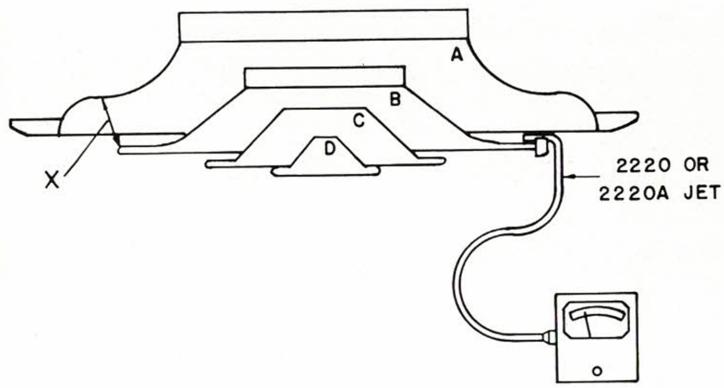


FLORITE AIR VELOCITY METER
MODEL MLD

FIG. 71

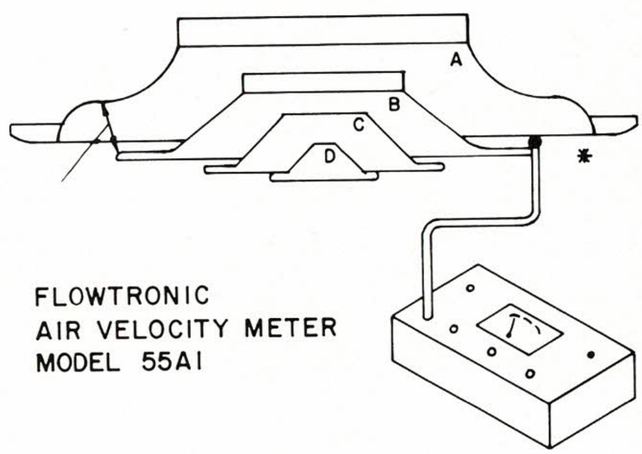
This chapter has pointed out that the air flow measuring instrument – the probe – the location of the reading – and the outlet must be "matched" if the proper factor is to be selected. Fig. 71 "matches" two velocity meters with the same diffusers to illustrate this requirement.

NECK DIAM. IN.	CONES X	UP FACTOR	CONES X	UP FACTOR
6	5/8	0.12	1 5/16	0.15
8	13/16	0.20	1 3/4	0.27
10	1	0.32	2 3/16	0.42
12	1 3/16	0.45	2 5/8	0.59
14	1 3/8	0.61	3 1/16	0.80
16	1 5/8	0.79	3 1/2	1.05
18	1 13/16	1.00	3 15/16	1.30
20	2	1.25	4 3/8	1,63
24	2 3/8	1.80	5 1/4	2.35
30	3	2.80	6 5/8	3.60
36	3	3.18	6 5/8	4.15



ALNOR VELOMETER

NECK DIAM. IN	CONES X	UP FACTOR	CONES X	UP FACTOR
6	5/8	0.14	1 5/16	0.18
8	13/16	0.25	1 3/4	0.32
10	1	0.39	2 3/16	0.50
12	1 3/16	0.56	2 5/8	0.71
14	1 3/8	0.76	3 1/16	0.98
16	1 5/8	1.00	3 1/2	1.30
18	1 13/16	1.25	3 15/16	1.60
20	2	1.55	4 3/8	2.00
24	2 3/8	2.25	5 1/4	2.90
30	3	3.55	6 5/8	4.50
36	3	4.10	6 5/8	5.10

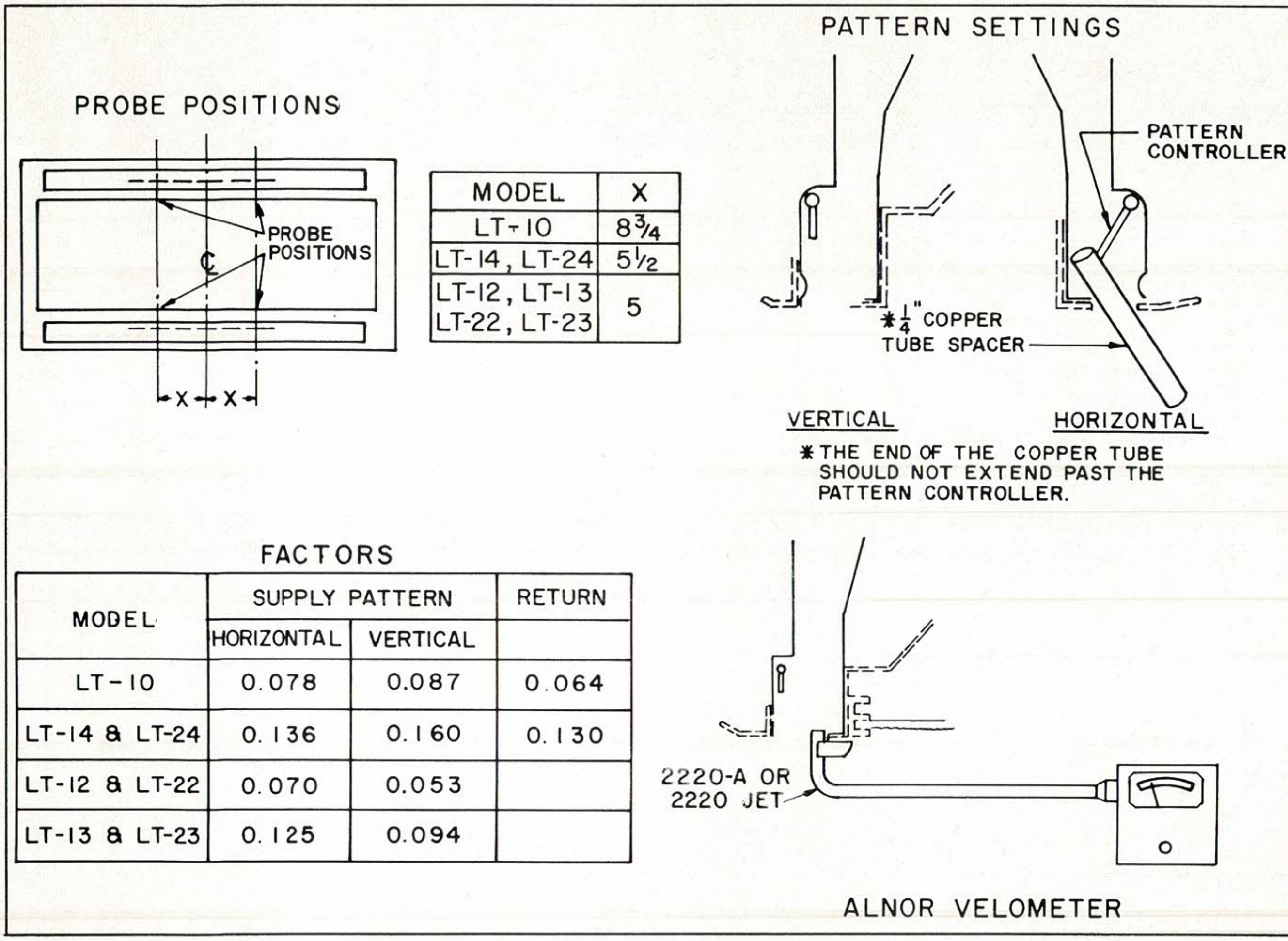


FLOWTRONIC
AIR VELOCITY METER
MODEL 55A1

* ROTATE PROBE FOR MAXIMUM VELOCITY

FIG. 72

Two more examples of the need for "matching" the testing instruments with the diffuser – all this in accordance with the recommendations of the diffuser manufacturer who, presumably, has laboratory tested his outlets with specified meters.



MODEL	X
LT-10	8¾
LT-14, LT-24	5½
LT-12, LT-13 LT-22, LT-23	5

FACTORS

MODEL	SUPPLY PATTERN		RETURN
	HORIZONTAL	VERTICAL	
LT-10	0.078	0.087	0.064
LT-14 & LT-24	0.136	0.160	0.130
LT-12 & LT-22	0.070	0.053	
LT-13 & LT-23	0.125	0.094	

FIG. 73

"Troffers" require specific probe positions as shown in the upper left detail and explained in the text. Since troffers may be used for supply or return the manufacturer will supply factors for both applications.

Typical factors for an adjustable round ceiling diffuser are shown in Fig. 71. For this type of diffuser the factors apply to the diffuser with the cones set in a particular position. The manufacturer has indicated four locations for the probe on a particular cone of the diffuser. These locations were selected to give a maximum reading for a particular air flow rate. Factors have been listed for five types of instruments and apply only to that instrument with the probe located as shown in the figures.

Fig. 73 shows the locations for measurement of air flow from the light troffer diffuser. For this diffuser the factors apply with the probe located at specific distances from the centerline of the diffuser. The average velocity measured at the locations, multiplied by the factor, indicates the total air flow from the diffuser. Since the diffuser may be adjusted for two types of discharge it is necessary that the proper factor be applied. Notice that a factor is also given for measurement of air flow with the diffuser used as a return inlet.

CHAPTER 22

FLOW MEASURING HOOD

A conical or pyramid shaped hood to collect discharge air and guide it over a flow instrument has frequent application as follows:

1. When balancing a large number of ceiling diffusers of common size a hood may permit reading from the floor and eliminate the need for a ladder.

2. When balancing troffer diffusers.

3. When the flow factors for a given outlet are not known.

4. When the discharge pattern does not contain a location to obtain an average velocity reading. A typical cone is shown in Fig. 74.

The balancing cone should be tailored for the particular job. To keep weight to a minimum, aluminum is normally used. Cardboard cones have been used. The large end of the cone should be sized to fit over the complete diffuser and should have a sponge rubber seal to eliminate leakage and to avoid ceiling marks. Care should be taken by the balancing technician that the sponge rubber is clean in order not to soil the ceiling while taking readings.

The cone should terminate in a straight section to permit good readings to be obtained. A flow reading instrument such as a velometer can be used to take the readings. The proper factors to apply to these readings should be calculated and should be verified by testing at a point where flow can be checked by a Pitot tube and draft gage.

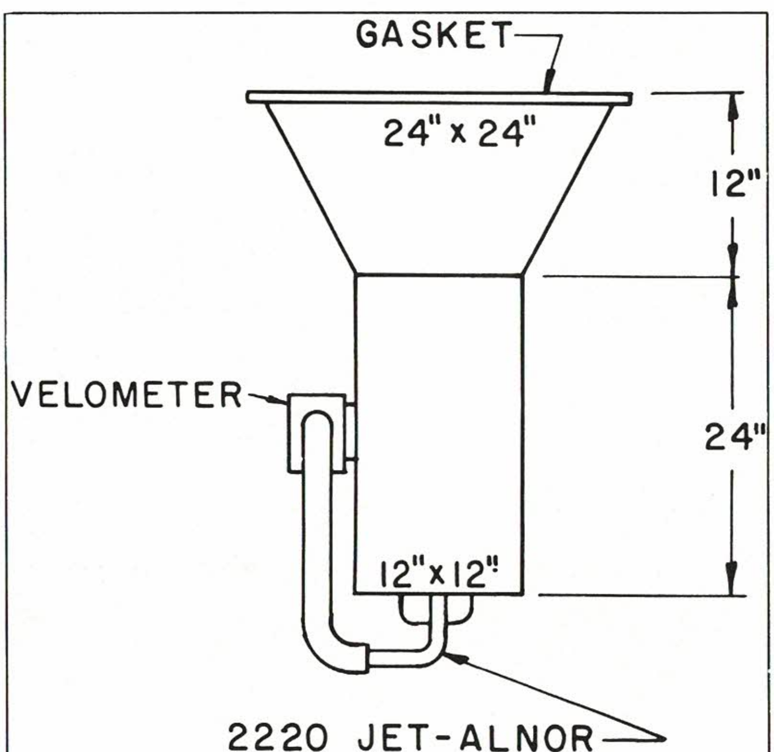


FIG. 74
Construction and dimensions for a typical flow measuring hood.

CHAPTER 23
SPEED MEASURING INSTRUMENTS

Revolution Counter

A revolution counter is a small, inexpensive, hand held counting device that is pressed to the center of a rotating shaft for a timed interval. It functions similarly to the total mileage indicator of an automobile odometer. It is used in conjunction with a stop watch and is normally engaged for a period of 30 or 60 seconds.

Since this type of instrument cannot normally be reset to zero the readings at the beginning and end of the test are noted and the number of revolutions determined by subtracting to get the difference between the two readings. This instrument is normally satisfactory for all balancing work.

Tachometer (Centrifugal Type)

This is a small, hand held instrument that directly indicates the instantaneous speed on the face of a dial through a mechanism that is similar to the fly wheel governor on a stationary steam engine or the speedometer of an ordinary passenger car. It is used in a manner similar to a revolution counter but no stop watch is required. Accuracy is normally good. The instrument should be periodically calibrated.

Tachometer (Chronometric Type)

This is an instrument (Fig. 75) that combines a precision stop watch and a revolution counter in one integrated unit. After the instrument is placed to the center of the rotating shaft, pressing a button will energize, simultaneously, the counter and the stop watch. After running for an accurately timed interval that is normally six seconds, the instrument stops accumulating revolutions even though it is still in contact with the shaft. The scale is calibrated so that if a six-second stop watch mechanism is used each actual revolution of the shaft indicates ten on the face of the dial so that readings are directly in rpm.

Electric Tachometer

This normally consists of a small, hand-held, rotating generator connected by a short lead to an instrument which reads the output of the generator. The instrument is calibrated directly in rpm. It is used in a manner similar to a centrifugal type tachometer. Accuracy is about 1% of full scale range.

Vibrating Reed Tachometers

This instrument has a series of tuned metal reeds. When this instrument is placed in contact with a vibrating machine, any reed that has a natural frequency in resonance with the impressed vibration will vibrate at maximum amplitude. This instrument is very easy to use and is especially suitable to very high rotating speeds. Its accuracy is quite good, but in order to be suitable, a large number of instruments covering various ranges or instruments with large numbers of reeds are required if accuracy is desired.

Strobescope

This instrument has a blinking light. The frequency of the blinking light is electronically controlled and adjustable. When the frequency of the blinking light is adjusted to equal the frequency of the rotating machine, the machine will appear to stand still. This unit is moderately expensive but is extremely accurate and further need not be in contact with the machine when it is being used.

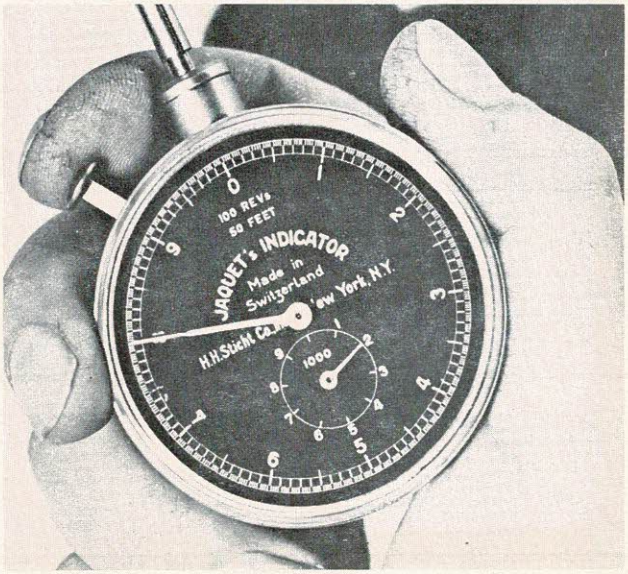


FIG. 75

Tachometer combining a revolution counter with a stop watch.

CHAPTER 24

THERMOMETERS

Dry bulb thermometers are any thermometers that read the dry bulb temperature of the air. The most common types used are the familiar glass stem, liquid-filled thermometers. The liquid generally is mercury or colored alcohol. For maximum accuracy the scale range should be as short as is practical and graduations should be marked in no greater than one degree steps. The accuracy of a good quality, liquid-filled thermometer should be within one per cent of scale range.

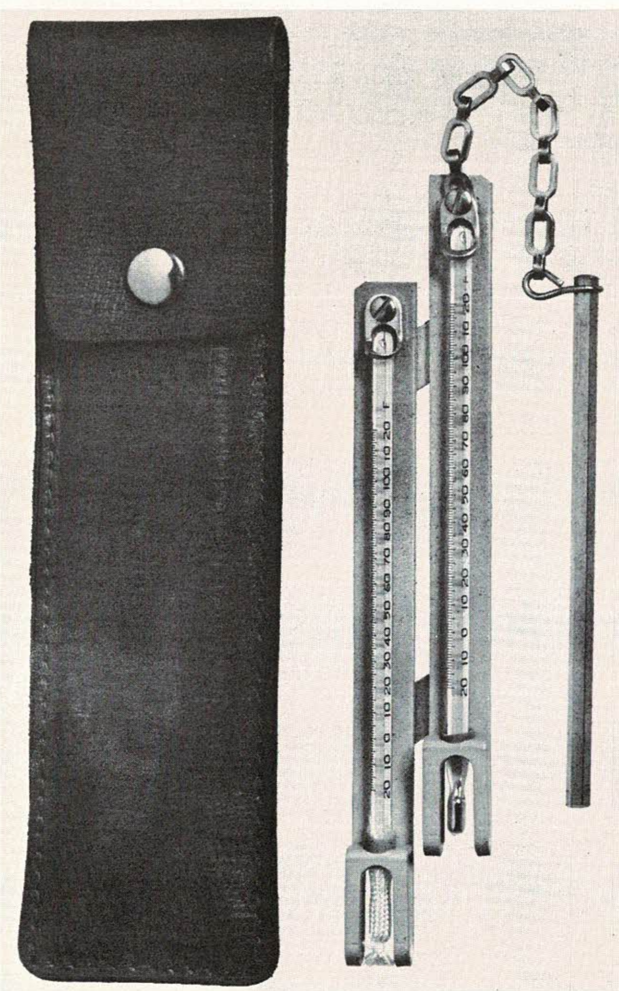


FIG. 76

Sling psychrometer with wet bulb and dry bulb stems and handle to whirl the instrument until the wet bulb settles.

Another type of dry bulb thermometer is the bimetal type in which a flat bimetal strip is spiral wound around a stem which carries an indicator hand which is located over a circular dial. A change in temperature will cause the bimetal to change shape and this, in turn, will turn the stem with the hand. This type of thermometer will have an accuracy of about two per cent of scale range.

Electric thermometers are usually one of two types. One is the thermocouple type, in which the thermocouple generates an electric current when temperature varies between the hot and cold junction of dissimilar metals. By reading the amount of current generated, an indication of temperature is given.

The other type of electric thermometer is the resistance thermometer, the best known example of which is the Anemotherm, covered in another chapter of this manual, since it is also used as an air velocity measuring instrument.

All the above instruments will read the dry bulb temperature of air, which indicates as well the sensible heat content of the air. In order to get an indication of the total heat content of any mixture of air and water vapor, a wet bulb temperature is required. The wet bulb temperature is that measured by a thermometer, usually a glass, liquid-filled type, which has a cloth wick over the bulb. The wick is saturated with water and the thermometer moved rapidly through the air. As water is evaporated from the wick, the temperature of the wick and the bulb is reduced below the dry bulb temperature. This wet bulb temperature, along with the simultaneous dry bulb temperature, establishes a point on a psychrometric chart from which humidity and heat content of the air can be determined.

To get a simultaneous reading of wet and dry bulb temperatures, a wet and a dry bulb thermometer are mounted side by side on a frame fitted with a handle by which the two thermometers can be whirled through the air. The whirling is stopped from time to time and readings taken until the temperature reading of the wet bulb thermometer starts to rise, the lowest temperature being taken as the wet bulb reading. This device is called a Sling Psychrometer. See Fig. 76

CHAPTER 25
AMMETER, VOLTMETER AND OHMETER

As explained in the electrical portion of the Technical Section III, electrical measurements and the subsequent proper calculations can determine the work being done by a motor. In addition, a check of amperage and voltage is always valuable to be sure that the motor is operating under safe and proper conditions.

The balancer should never assume that voltages are correct. Before a motor is permitted to operate a voltage reading should be taken. This is done with a voltmeter. The reading should be taken as near to the motor as possible. However, to avoid stripping the insulation from the motor lead connections it is the usual practice to take these readings on the load terminals of the starter if the starter is not too remote from the motor.

The voltmeter has either a straight or dial type scale that reads from zero to its maximum capacity. Some meters have adjustable scales. Such a meter should be adjusted to a scale in which you anticipate your reading will fall. There are two leads from the voltmeter which either clamp on or contact the terminals. When reading single phase voltage the leads should be applied to the two load terminals. The resulting single reading is the voltage of the current being applied to the motor.

When reading three phase current it is necessary to apply the voltmeter terminals to Pole No. 1 and Pole No. 2; then to Pole No. 2 and Pole No. 3; and finally to Pole No. 3 and Pole No. 1. This will result in three readings each of which will likely be a little different but which should be close to each other. For practical purposes they may be averaged. If there is a wide diversity in the readings, shut the motor down and notify the proper authorities through the channels established for the project.

If the average voltage delivered to the motor varies by more than a few volts from the nameplate rating of the motor, several things can occur. A rise in voltage may damage the motor and will cause a drop in the amperage reading. A drop in the voltage will cause a rise in the amperage and can cause the overload protectors on the starter to "kick out." In either case, it is advisable to promptly report high or low voltage situations.

The most popular way of reading amperage is with the "clamp on" type of ammeter. This device has a scale of a straight or dial type and either an attached or remote set of "jaws" designed to clamp about a wire. The scale on this instrument may also be divided in ranges and be adjustable. Fig. 77

Care should be exercised to pick the proper range. Readings may be taken at the motor leads or from the load terminals of the starter. To determine the amperages of single phase motors, place the clamp about one wire. When involved with three phase current, take readings on each of three wires and average the results.

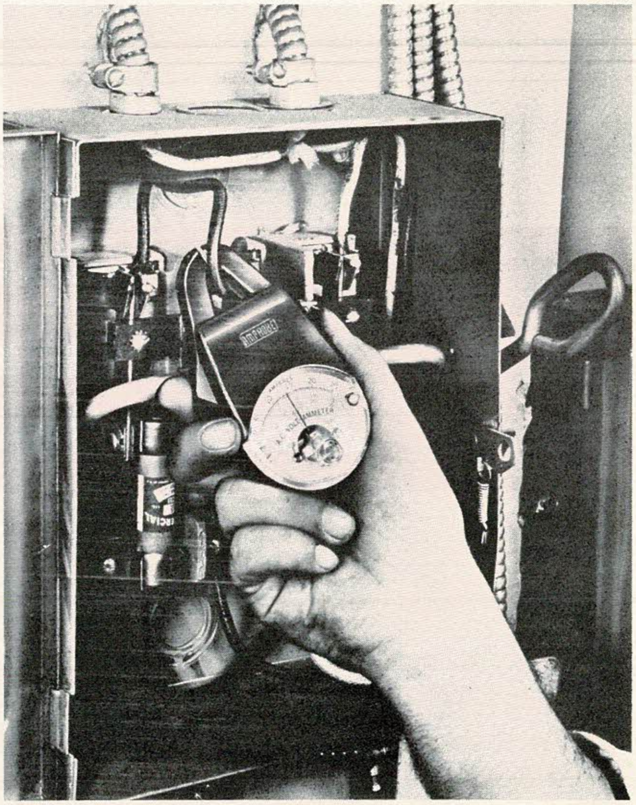


FIG. 77

Taking an amperage reading with a clamp-on volt-ammeter.

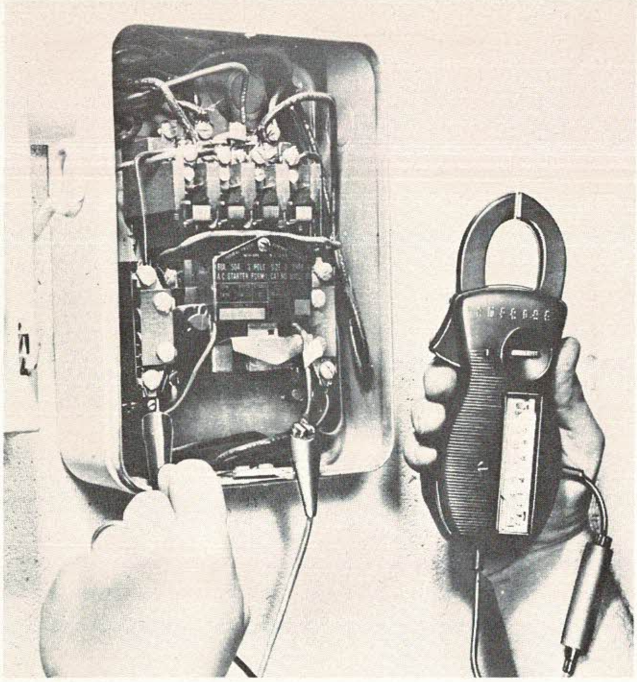


FIG. 78
Checking resistance with a clamp-on volt-amp-ohmmeter

With this information the balancer should be ready to calculate brake horsepower as outlined in Section III, Chapter 14, of this manual.

Some ammeters and voltmeters are combined in a single instrument. This combination unit is handy since it reduces the number of individual instruments the balancer must carry with him.

The ohmeter is a device used to measure resistances. In the air conditioning field its principal use would be to test the continuity of circuits. This would be a problem more likely to occur with installers of electric or electronic temperature controls, burner or refrigeration controls than with the air balancer.

Again, this instrument can be found in combination with a voltmeter-ammeter. Fig. 78

CHAPTER 26

SMOKE BOMBS AND GENERATORS

These are devices generally used for the study of air flows and for the detection of leaks.

Smoke bombs come in various sizes giving different lengths of burning time during which highly visible, non-toxic smoke readily mixes with the air simplifying the observation of flow patterns.

When testing for leaks sufficient smoke should be used to fill a volume 15 to 20 times larger than the duct or enclosure volume to be tested.

Smoke sticks are convenient in that they provide a small indicating stream of smoke, like the puff from a cigarette, but continuous for about ten minutes.

Smoke guns are valuable in tracing air currents, determining the direction and velocity of air flow and the general behavior of either warm or cold air in conditioned rooms.

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