DDS 512-2

DESIGN DATA SHEET

A METHOD FOR DETERMINING THE SIZE OF VENTILATION DUCTS

DEPARTMENT OF THE NAVY
NAVAL SEA SYSTEMS COMMAND
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DEPARTMENT OF THE NAVY, NAVAL SEA SYSTEMS COMMAND

SECTION DDS512-2

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DDS512-2-a. **References**

(a) General Specifications for Ships of the United States Navy, Section 512, "Heating, Ventilation and Air Conditioning," Naval Sea Systems Command, Department of the Navy, current edition.

(b) Duct Size Calculator, Navy #0938-036-0010, Bureau of Ships, Department of the Navy, October 1, 1968.

(c) ASHRAE Handbook 1985 Fundamentals, American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), Atlanta, GA, 1985.

(d) "HVAC Duct System Design Manual," 2nd Edition, Sheet Metal and Air Conditioning Contractors' National Association Inc. (SMACNA), Vienna, VA, April 1981.

(e) Design Data Sheet DDS512-1. "Pressure Losses of Ventilation Fittings," dated 30 September 1987.

(f) "Heating, Ventilating and Air Conditioning Equipment Manual," S9512-BS-MMA-010, Naval Sea Systems Command, Department of the Navy, September 30, 1985.

DDS512-2-b. Abbreviations, Symbols and Subscripts

1. Abbreviations

cubic feet per minute	cfm
feet	ft
feet per minute	fpm
inch, inches	in
inches of water, gauge	in wg
square feet	ft^2
square inches	in^2

2. Symbol

Symbol	Parameter	Unit
A	area or sum of C_q factors	in^2, ft^2; --
c	sum of total pressure loss coefficients for each duct section of common velocity pressure	--
C_q	a parameter used to simplify calculations	--
D	diameter	
d	diameter	in, ft
f	friction factor per unit length of duct	in, ft in wg/ft
H	duct height (i.e., the duct dimension parallel to the axis of a bend) or available total pressure after deducting losses for equipment	in; in wg
K	pressure loss coefficient	
L	length of straight duct	--
L_e	equivalent length of duct	ft
N	number of terminals to be installed in duct section	ft --
P, p	pressure	
p_v	velocity pressure	in wg
Q	air flow rate	in wg
Qm	mean air flow rate	cfm
V	velocity	cfm
W	duct width (i.e., the duct dimension in the plane of a bend)	fpm in
π	Pi	--

3. Subscripts

vd	velocity pressure in duct four feet downstream of terminal
vm	maximum velocity pressure in branch preceding last terminal
vu	velocity pressure in duct two feet upstream of terminal
1,2,..n	successive sections of duct

DDS512-2-c. General

1. This document presents the NAVSEA approved method for sizing ducts. In a satisfactory design, ducts are proportioned so that the required air flow rates are delivered at the terminals. Since ventilation systems are generally tested late in the construction schedule, corrective measures are expensive regarding both time and material. Careful initial design will minimize these corrective measures.

2. The term "pressure loss" used herein (except where noted in reference (e)) is always total pressure loss. The pressure loss coefficients referred to in the text take into account direction of flow and static pressure regain. Units of pressure used throughout this document are: inches of water, gauge (in wg).

3. The following definitions are used in the duct size calculation method presented in this document.

longest run - that portion of the duct system in which the largest total pressure loss is developed.

main - that duct which contains the fan and delivers air to/from that fan to any number of branch ducts or branch main ducts.

branch main - that duct which connects the main duct to two or more individual branch ducts.

branch - that duct which carries air to/from the main duct or a branch main duct to/from a single terminal.

4. The longest run usually extends from the system intake or discharge to the last (most distant) terminal. The longest run, however, cannot be determined simply by looking at a system schematic. It must be determined by performing a specific series of pressure loss calculations. These calculations are based upon the following two assumptions:

- An assumed fan - This assumption is based on the known system air flow rate; and an estimated system pressure.

- An assumed set of duct sizes - This assumption is based on the known system air flow rate and; either a maximum pressure loss/ft of duct or a maximum flow velocity.

Once the longest run has been determined, a safety factor of 10 percent should be applied to the calculated pressure loss. This value (including the safety factor) is the system pressure loss.

5. The sum of the system pressure loss (developed in 4, above) and the relative pressure of the terminal space (i.e. the space at the end of the longest run), if any, is equal to the pressure which must be developed by the system fan when it is operating at the required system air flow rate (cfm). For a terminal space which must be maintained at a positive pressure, the pressure (in wg) within the space is treated as a loss in the supply calculation and as a gain in the exhaust calculation. For a space which is kept under a negative pressure, the reverse is true; e.g., the pressure of the terminal space is treated as a gain in supply calculations, and a loss in exhaust calculations. Examples of spaces in which a positive pressure must be maintained are operating rooms and spaces that have natural exhaust. Examples of spaces in which a negative pressure must be maintained are toilet rooms, shower rooms, or spaces that have a natural supply.

DDS512-2-d. Derivation of Formulas

1. The duct sizing method presented in this Design Data Sheet is based on the following three principles:

(a) The ductwork shall be sized so that the pressure loss per foot of equivalent length of duct is, essentially, constant throughout the longest run.

(b) Each branch shall be sized so that the pressure loss per foot of equivalent length of duct is, essentially, constant throughout that branch.

(c) The total pressure loss due to friction in a length of straight duct equal to 39 times the diameter (or equivalent diameter) of the duct shall be equal to the velocity pressure, p_v, in that duct section. This principal is based on tests performed on typical, fairly clean, naval duct systems.

2. When restated mathematically, principle (c), above, can be utilized to calculate the diameter, D, of the duct main.

$$p_v = 39Df \qquad (1)$$

Since, by definition:

$$p_v = \left(\frac{V}{4006}\right)^2 \qquad (2)$$

And, since:

$$V = \frac{Q}{A} \qquad (3)$$

When A is in units of square feet; or

$$V = \frac{144Q}{A} \qquad (3A)$$

when A is in units of square inches. For round duct:

$$V = \frac{Q}{A} = \frac{Q}{\pi\left(\frac{D}{2}\right)^2} = \frac{Q}{(0.785)(D)^2} \qquad (4)$$

Solving for D in terms of Q and f yields:

$$p_v = 39Df = \left(\frac{V}{4006}\right)^2$$

Therefore,

$$V^2 = 39Df(4006)^2$$

And, by substituting equation (4) for V:

$$\left(\frac{Q}{.785(D)^2}\right)^2 = 39Df(4006)^2$$

And, rearranging:

$$(D)^5 = \frac{Q^2}{(.78\ \ ^2 \times 39f(4006)^2}$$

Simplifying,

$$D = \frac{(0.0192)(Q)^{0.4}}{(f)^{0.2}} \qquad (5)$$

The total pressure available for ductwork and fittings after deducting losses due to equipment is designated: H. Therefore, H can be defined as the sum of all duct losses (Lf) and all duct fitting losses (cp_v).

$$H = Lf + c_1p_{v1} + c_2p_{v2} + \ldots.c_np_{vn} \qquad (6)$$

Substituting 39Df for p_v:

$$H = Lf + c_1 39D_1f + c_2 39D_2f + \ldots.c_n 39D_nf$$

Substituting equation (5) for D:

$$H = Lf + c_1 39f\left(\frac{.0192(Q_1)^{0.4}}{(f)^{0.2}}\right) +$$

$$c_2 39f\left(\frac{.0192(Q_2)^{0.4}}{(f)^{0.2}}\right) + \ldots.c_n 39f\left(\frac{.0192(Q_n)^{0.4}}{(f)^{0.2}}\right)$$

Clearing all fractions and factoring yields:

$$H = Lf + (f)^{0.8}[0.749c_1(Q_1)^{0.4} + 0.749c_2(Q_2)^{0.4} + \ldots.0.749c_n(Q_n)^{0.4}]$$

Then, by letting $0.749c_1(Q_1)^{0.4} = C_{q1}$, etc., the equation above can be simplified to:

$$H = Lf + (f)^{0.8}(C_{q1} + C_{q2} + \ldots.C_{qn})$$

Finally, by letting $(C_{q1} + C_{q2} + \ldots.C_{qn})$ equal the dimensionless factor, A, the equation above can be reduced to:

$$H = Lf + A(f)^{0.8}$$

Dividing both sides of this equation by H yields:

$$1 = \frac{L}{H}(f) + \frac{A}{H}(f)^{0.8} \qquad (7)$$

By assigning successive values to the friction factor, f, the straight line relationships between (L/H) and (A/H) shown in figure 1 were plotted. This information has been incorporated into reference (b). Figure 1 and reference (b) can be used, interchangeably, to determine the duct friction factor, f.

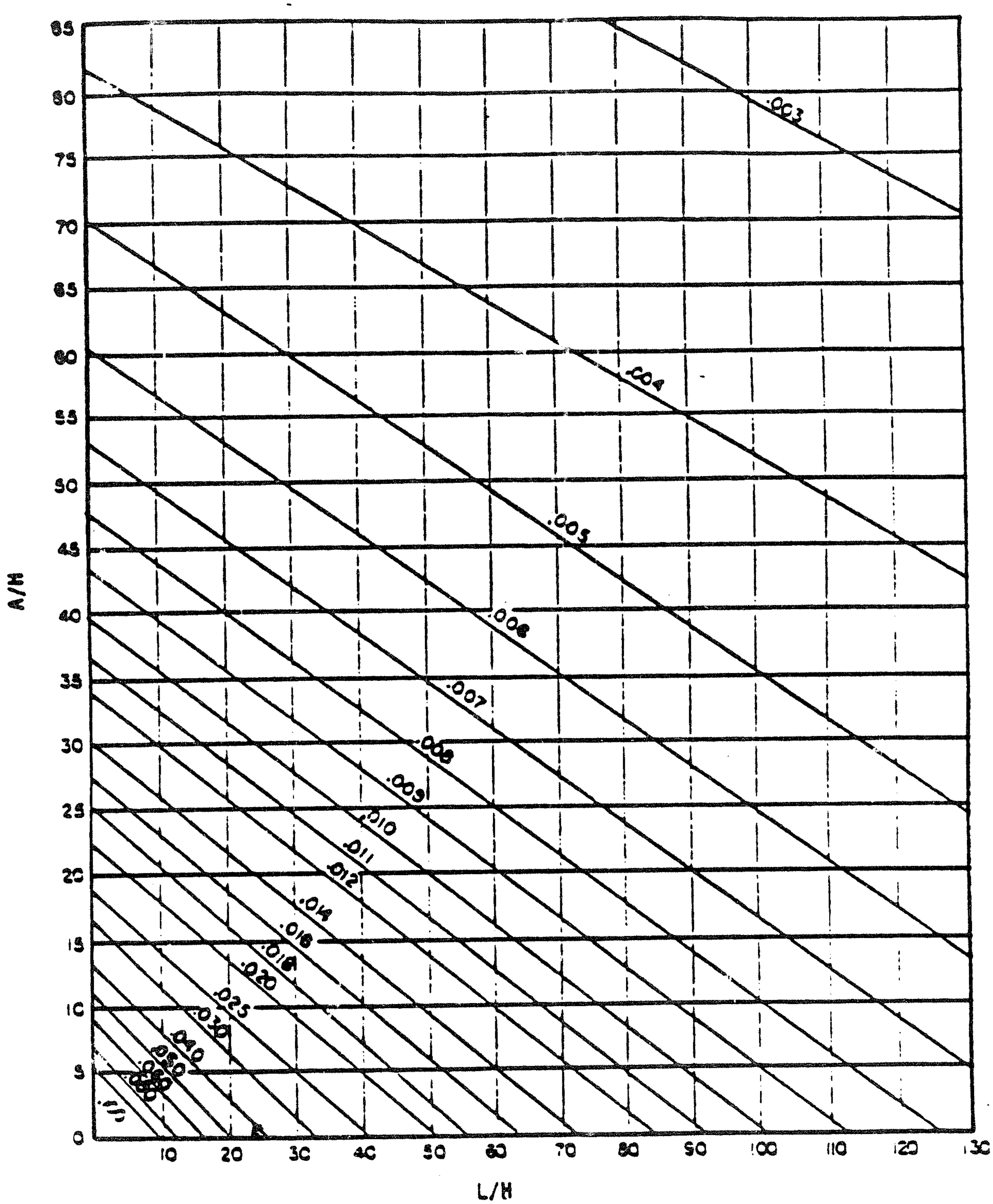

Figure 1. Friction Loss per foot of Duct in Terms of L/H versus A/H

DDS512-2-e. Design Procedure for the Main, Branch Main, and Branches

1. Designing the main

(a) On a schematic drawing of the duct system, indicate all component parts of the system (fan, heaters, cooling coils, and other fittings) in their respective locations and determine the air flow rates through each device, duct section, and terminal.

(b) Estimate the system pressure requirements. As a trial, select (from standard fan characteristic curves) a fan that will deliver the required air flow rate and develop the estimated pressure. If there is a choice of fans, further development will indicate the most suitable.

(c) Determine the total pressure losses developed by each of the various devices located in the main. Subtract the sum of these known losses from the available fan pressure. In addition to the pressure losses developed by the various equipment devices, a safety factor (usually 10 percent of the fan's maximum total pressure) should be subtracted from the available fan pressure. The remaining pressure, H, is to be distributed throughout the duct main. The basis for distributing the remaining pressure will be an equal loss of pressure for each foot of equivalent length of duct.

(d) Determine the pressure loss coefficients (K) of each of the duct fittings in the system using reference (e). Some pressure loss coefficients will have to be calculated based on assumed duct dimensions, since the actual duct dimensions have not yet been calculated.

(e) Determine the c-value for each section of duct. The c-value is the sum of the pressure loss coefficients in each section.

(f) Determine the length of straight duct, L, in each duct section.

(g) Determine the C_q factor for each section. The value of C_q is a function of the c-value (Step e, above) and the air flow rate (cfm) in each duct section. C_q factors can be found by using reference (b).

(h) Determine A, the sum of the C_q factors.

(i) Determine the system friction factor, f (in wg/ft). the friction factor, f, is a function of A/H and L/H and can be found using reference (b) or figure 1.

(j) Determine the required minimum diameter of each duct section using reference (b) or the diameter formula developed previously:

$$D = \frac{(0.0192)(Q)^{0.4}}{(f)^{0.2}} \qquad (5)$$

Based on the calculated diameter, determine the air velocity in the first section of main using reference (b). If there is a choice of fans, the one resulting in a velocity nearest the maximum given in reference (a) is usually the most desirable. If the fan selected has an inlet (and outlet) duct connection different from the duct size selected above, transitions shall be utilized to mate the duct to the fan. Note that transitions shall not be located within close proximity (see reference (a)) of the fan inlet or outlet. Also, the air velocity within the duct sections adjacent to the fan shall not exceed the maximum velocity permitted in reference (a). The actual pressure drop per linear foot, f, must then be determined for this section.

(k) Using the original system schematic, complete the sizing of the duct main, basing sizes on the calculated f-value. Rectangular or flat oval duct may be substituted for round duct so long as the equivalent duct diameter is, essentially, equal to the calculated diameter (based on the friction factor, f). Equivalent duct diameters can be taken from tables in reference (d) or from manufacturer's catalogs. They may, also, be calculated from formulas given in reference (c) chapter 33, equations 31 and 32. For example, a duct carrying 2,500 cfm with a friction loss of f = 0.017 in wg/ft requires a round duct of 12 inches diameter. The equivalent sizes for differently shaped ducts are:

Shape	Size (in)	Area (ft^2)
Round	12	0.785
Flat oval	7 x 18	.802
Rectangular	6 x 21.5	.896

Note that the friction loss per foot remains constant, but the cross-sectional areas have changed significantly. If shop practices of individual yards require that standard duct dimensions be utilized, it will be necessary to correct the friction factor, f, and the velocity pressure, p_v, to reflect the actual duct dimensions.

(l) Determine velocity pressure for each section of the main using actual area selected for each section. Areas of rectangular or flat oval duct can be calculated or taken from tables 6-2 and 6-3 in reference (d). The velocity, V, and velocity pressure, p_v, should be calculated using equations (3) and (2), as follows:

$$V = \frac{Q}{A} \qquad (3)$$

And,

$$p_v = \left(\frac{V}{4006}\right)^2 \qquad (2)$$

The velocity pressure (velocity head) can also be determined using reference (b).

The velocity pressure of ducts with a given air flow rate and a constant equivalent diameter, D_e, will vary with the actual cross-sectional area. In the previous example, velocities and velocity pressures vary as follows:

Shape	Size (in)	Velocity (fpm)	f (in wg/ft)	Velocity pressure (in wg)
Round	12	3,200	0.017	0.63
Flat oval	7 x 18	3,120	0.017	0.60
Rectangular	6 x 21.5	2,790	0.017	0.48

(m) Correct all errors made in selecting pressure loss coefficients by using actual duct dimensions instead of estimated duct sizes.

(n) Determine the pressure loss (in wg) developed in each duct section (Lf) and each fitting (Kp_v) in the duct main, using the corrected pressure loss coefficients and friction factors.

(o) Calculate the available pressure at the first branch takeoff by deducting the pressure loss developed in the first duct section from the rated fan pressure. Find the pressure available at each consecutive branch takeoff by deducting the loss in the section from the pressure available at the preceding branch take-off.

(p) Ensure that the total pressure available at the beginning of the last duct section is greater than the pressure losses developed in that section. Otherwise, there will not be sufficient velocity pressure to move the air into the space. As described previously, the total pressure available at the beginning of the last duct section must be sufficient to maintain the relative pressure requirements of the terminal space.

Discrepancies in the actual and expected available pressures are introduced by:

(1) Changes from nominal to standard duct dimensions.

(2) Changes from round to equivalent rectangular or flat oval ducts.

(3) Changes from assumed pressure loss coefficients to actual pressure loss coefficients.

If the pressure available at the beginning of the last duct section does not exceed the pressure loss developed in that section, the size of the last section, (and, possibly, some or all of the preceding sections) must be increased. If changing the duct sizes does not correct the pressure imbalance, it may be necessary to pick a different fan.

Special Note: In designing a duct system, it may be necessary or convenient to make several branch takeoffs in a duct with constant cross-sectional area (i.e., the area of the duct is held constant, rather than reduced after every branch takeoff, to facilitate fabrication). In this case, a mean air flow rate should be determined and used to calculate the size of the (constant cross-section) duct. The mean air flow rate is determined by the following formula:

$$Q_m = \left(\frac{Q_1^{\,2} L_{e1} + Q_2^{\,2} L_{e2} + \ldots . Q_n^{\,2} L_{en}}{L_{e1} + L_{e2} + \ldots . L_{en}} \right)^{0.5} \qquad (8)$$

2. Designing the branch main and branches

The pressure used as a basis for sizing branch mains and branches is the pressure available at each takeoff as determined in step 1(o) of the design of the duct main. The procedure used for designing branch mains and branches is similar to that used for designing the main. The methodology is simplified for branches since there is no air flow rate reduction. For this reason, the calculated value of C_q becomes the A value.

DDS512-2-f. Example Problem

The ventilation supply system shown schematically in figure 2 is the basis for the calculations in this example problem. The total air flow rate of the system, determined by a summation of the air flow rates of the connected terminals is 4,100 cfm. The spaces supplied by this system are assumed to be mechanically exhausted and the pressure within spaces is assumed to be atmospheric (i.e. neither positively or negatively pressurized). Based on an estimated system pressure loss of 4.5 in wg (See Appendix A), an inspection of the characteristic fan curves for Navy standard fans indicates that no fan directly matches the system air flow rate and pressure (reference (f), figure 1.4). However, a Navy standard A4-1/2 fan can be used to deliver 4,100 cfm at up to 5.2 in wg.

Once the preliminary fan selection has been made, each flow path within the system must be evaluated to determine which path has the greatest total pressure loss. This flow path is sometimes referred to as the "longest leg" in the sample problem. Flow paths A-G, A-H, A-J and A-K must be investigated. Flow paths A-L and A-O will not be investigated since they are located within close proximity to the system fan, and since, by inspection, the pressure losses in these paths appear to be less than the expected losses in the four other flow paths. Where flow paths are not readily eliminated by inspection, the designer must evaluate all of the flow paths in the system to determine the longest leg.

Designing the Main

The evaluation of flow path A-G is presented below. The lower case letters in parentheses at the beginning of each step of the evaluation refer back to the procedure given in paragraph DDS512-2-e.1.

Step (c) Record in column 11 of table 1 total pressure loss for various devices located in flow path A-G (in inches of water, gauge). Equipment sizes in the sample system were determined using the velocity criteria in reference (a). Total pressure losses for equipment items are:

Section A-B:

(1) Steam Preheater. Reference (f) indicates that the total pressure loss of a size 29M heater at 4,100 cfm is equal to 0.38 in wg.

(2) 15AF Navy Standard Filters. The sample problem has four filters arranged in a bank. Airflow through each filter is one fourth of the total. The face velocity for each filter can be calculated by substituting into equation (3A), as follows:

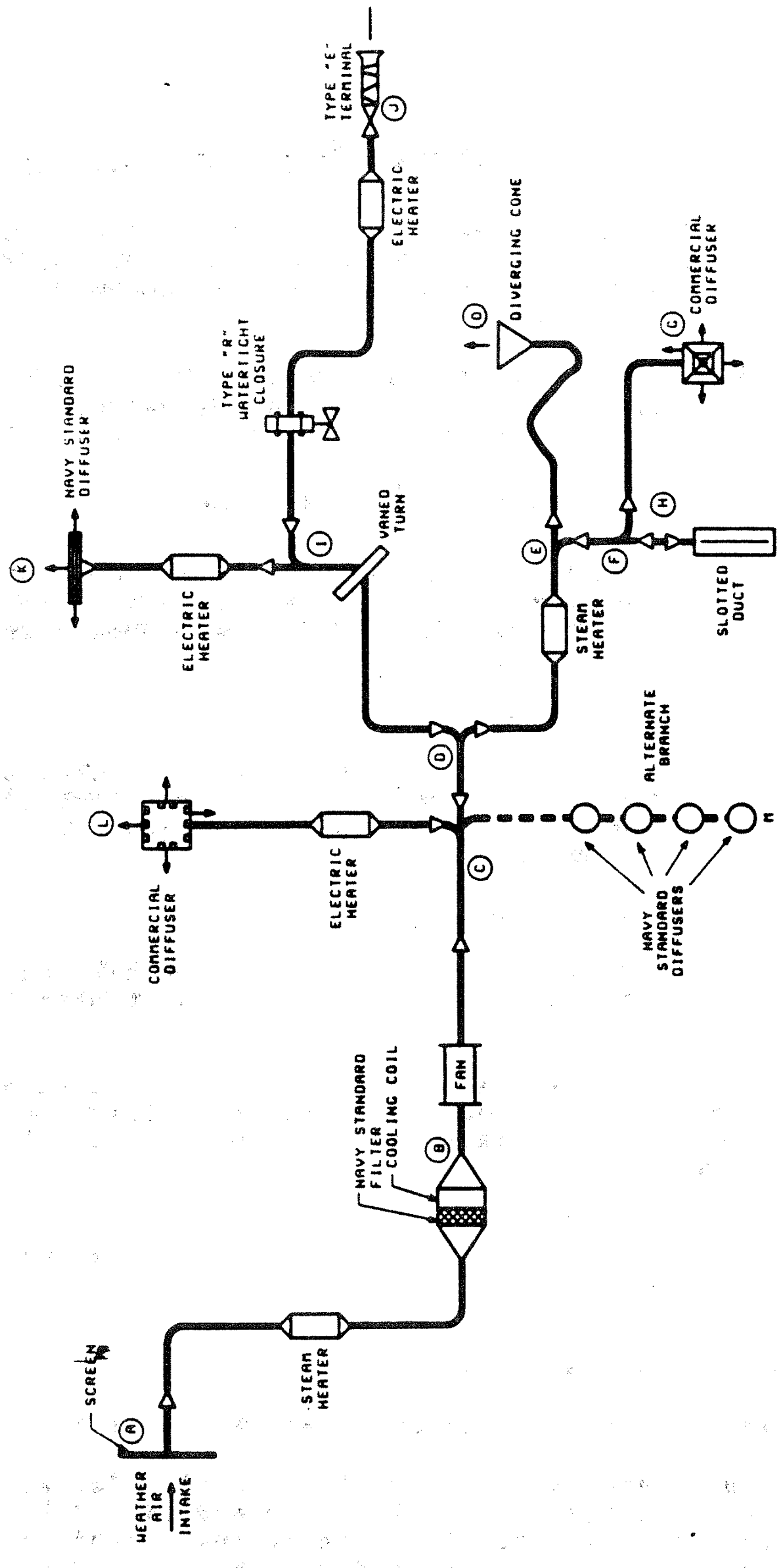

Figure 2. Example System

$$V = \frac{144Q}{A} = \frac{144(4100/4)}{333.1} = 443 \text{ fpm}$$

The total pressure loss (from figure 38 in reference (e)) for a Navy standard filter at the face velocity calculated above is 0.13 in wg.

(3) 58 DW Navy Standard Cooling Coil. The face velocity for the coil can be calculated by substituting into equation (3), as follows:

$$V = \frac{Q}{A} = \frac{4100}{10} = 410 \text{ fpm}$$

The total pressure loss (from figure 38 reference (e)) for a Navy standard cooling coil at the calculated face velocity is 0.50 in wg.

Section D-E:

(1) 27L Steam Heater. Reference (f) indicates that the average total pressure loss of a steam heater at 2050 cfm is equal to 0.24 in wg.

Section F-G:

(1) Commercial Diffuser. Based on catalog information, the pressure loss equals 0.08 in wg.

The available pressure factor, H, is calculated by subtracting the equipment pressure losses and a (10 percent) safety factor from the available fan pressure.

$$H = 5.20 \text{ in wg} - (0.38 + 0.13 + 0.50 + 0.24 + 0.08) \text{ in wg} - (0.10 \times 5.20) \text{ in wg}$$

$$= 5.20 - 1.33 - 0.52 \text{ in wg}$$

$$= 3.35 \text{ in wg}$$

This factor will be utilized in Step (i).

Step (d) Find pressure loss coefficients (K), for all of the fittings in the flow path and record these in column 3 of table 1. Some of these coefficients must be based on assumed duct sizes. Duct sizes can be assumed for the preliminary calculations by

TABLE 1. PRELIMINARY EVALUATION OF FLOW PATH A-G

SECTION & CFM	DUCT DEVICE OR FITTING	K PREL.	K FINAL	Cq	L (ft)	DUCT DIMENSIONS (n) PREL	DUCT DIMENSIONS (n) FINAL	f (in wg/ft)	Pv (in wg)	Pt (in wg)	Ttl. Press. avail. at pt. (in wg)
1	2	3	4	5	6	7	8	9	10	11	12
A TO B	Screen (1 1/2 inch mesh)	0.25	0.25								
4100	Inlet Loss	0.87	0.87								
	Transition (Converging)	0.05	0.05								
	90° Elbow (H/W=0.75) (r/W=0.60) 2 SPL	0.11	0.12								
	Transition (Diverging) (A1/Ao=1.91)	0.16	0.15								
	Steam Preheater (size 29M)									0.38	
	Transition (Converging)	0.05	0.05								
	90° Elbow (H/W=0.75) (r/W=0.60) 3 SPL	0.06	0.06								Point A 0.00
	Transition (Diverging) (A1/Ao=7.51)	0.30	0.30								
	4-15AF Navy Std Filters									0.13	
	Cooling Coil									0.50	
	Transition (Converging)	0.05	0.05								
	Straight Duct				30	14x12	16x12	0.0132		0.40	
	c =	1.90	1.90	39.7					0.59	1.12	
										2.53	
B TO C	Vaneaxial Fan										
4100	Transition (Converging) (Round to Rectangular)	0.05	0.05								
	Straight Duct				4.5 5.0	18.0 14x12	18.0 16x12	0.0057 0.0132		0.03 0.07	Point B -2.53
	c =	0.05	0.05	1.04					0.34	0.02	
										0.11	
C TO D	Rectangular Split Main	0.00	0.00								
3550	Transition (Converging)	0.05	0.05								
	Straight Duct				10	13x12	14x12	0.0137		0.14	Point C 2.56
	c =	0.05	0.05	0.99					0.58	0.03	
										0.17	

TABLE 1. PRELIMINARY EVALUATION OF FLOW PATH A-G (Continued)

SECTION & CFM	DUCT DEVICE OR FITTING	K PREL.	K FINAL	Cq	L (ft)	DUCT DIMENSIONS (n) PREL	FINAL	f (in wg/ft)	Pv (in wg)	Pt (in wg)	Ttl. Press. avail. at pt. (in wg)
1	2	3	4	5	6	7	8	9	10	11	12
D TO E 2050	Takeoff From Rectangular Split Main	0.25	0.27								
	Transition (Converging)	0.05	0.05								
	90° Elbow (H/W=0.91) (r/W=1.0) NO SPL	0.21	0.22								
	Transition (Diverging) (A1/Ao=1.85)	0.16	0.15								
	Steam Heater (size 27L)									0.24	Point D 2.39
	Transition (Converging)	0.05	0.05								
	Straight Duct				25.5	12x8	11x10	0.0132		0.34	
	c =	0.72	0.74	11.4					0.45	0.33	
										0.91	
E TO F 1600	Takeoff From Rectangular Split Main	0.67	0.56								
	Straight Duct				7	9x8	10x9	0.0133		0.09	Point E 1.48
	c =	0.67	0.56	9.60					0.41	0.23	
										0.32	
F TO G 1000	Takeoff From Rectangular Split Main	0.08	0.50								
	Transition (Diverging) (A1/Ao=1.00)	0.05	0.00								
	90° Elbow (H/W=1.00) (r/W=1.0) NO SPL	0.23	0.21								
	90° Elbow (H/W=1.00) (r/W=0.60) 2 SPL	0.11	0.11								
	Transition (Diverging) (A1/Ao=5.06)	0.28	0.28								Point F 1.16
	Commercial Diffuser (18 in. x 18 in.)									0.08	
	Straight Duct				16.5	8x8	8x8	0.0121		0.20	
	c =	0.75	1.10	8.9					0.32	0.35	
										0.64	

using a friction factor of .01 in wg/ft or by using the maximum air velocities given in reference (a). The estimated K-values, by section, are as follows:

Section A-B:

Component	K-Value
Screen at intake (20" x 16")	0.25
Inlet (in bulkhead)	0.87
Transition after inlet (20" x 16" to 14" x 12")	0.05
90° elbow	0.11
Transition before preheater (14" x 12" to 30" x 12-1/4")	0.16
Transition after preheater (30" x 12-1/4" to 14" x 12")	0.05
90° elbow	0.06
Transition before filters (14" x 12" to 39-1/2" x 36-1/2")	0.30
Transition after cooling coil (39-1/2" x 36-1/2" to 18" Dia.)	0.05
	c = 1.90

Section B-C:

Component	K-Value
Transition at fan discharge (18" Dia. to 14" x 12")	0.05
	c = 0.05

Section C-D:

Component	K-Value
Rectangular split main	0.00
Transition after takeoff (14" x 12" to 13" x 12")	0.05
	c = 0.05

Section D-E:

Component	K-Value
Takeoff from rectangular split main (12" x 8")	0.25
Transition after takeoff	0.05
90° elbow	0.21
Transition before heater (12" x 8" to 22" x 9-1/4")	0.16
Transition after heater (22" x 9-1/4" to 12" x 8")	0.05
	c = 0.72

Section E-F:

Component	K-Value
Takeoff from rectangular split main (9" x 8")	0.67
	c = 0.67

Section F-G:

Component	K-Value
Takeoff from rectangular split main	0.08
Transition (Diverging)	0.05
90° Elbow (no splitters)	0.23
90° Elbow (2 splitters)	0.11
Transition (Diverging)	0.28
	c = 0.75

Step (e) In the listings above, the pressure loss coefficients (K) for each section of the duct main have been summed to determine c-value for that section.

Step (f) The length of straight duct in each section of the main must be measured and the total length recorded in column 6 of table 1. Add the values in column 6 to find total length factor, L, for the duct main. In this example L = 98.5 feet.

Step (g) Using the back of the Duct Size Calculator, reference (b), determine the C_q values for each duct section. To do this, set the air flow rate (cfm) for the section opposite the value of c calculated for each duct section and read C_q. The value of c for the individual duct sections can be found in column 3 in table 1. Record the values of C_q in column 5 of table 1. Note: Values for c less than 0.1 do not appear on the calculator. If c is less than 0.1, use a value of 10c to enter the Duct Size Calculator. Read the value of C_q from the calculator, and divide that value by 10 to find the value of C_q for that duct section.

Step (h) Calculate the value of the fitting pressure loss factor, A, for the flow path by totaling the values of C_q determined in Step (g). In this example, A = 71.6.

Step (i) Using the values of A (step h), H (step c), and L (step f), determine the values of A/H and L/H.

$$\frac{A}{H} = \frac{71.6}{3.35} = 21.4$$

$$\frac{L}{H} = \frac{98.5}{3.35} = 29.4$$

Set A/H opposite L/H on the back of the Duct Size Calculator reference (b), and read friction factor per unit length of duct throughout the main. The friction factor, f, can also be determined using figure 1. The friction factor for the example problem, f, equals 0.0122 in wg/ft.

Step (j) Based on the f-value (determined in step (i)) and the air flow rate in each duct section, the ideal (or required minimum) diameter of each duct section can be determined from the Duct Size Calculator. These duct sizes are then recorded in column 7 of table 1. Duct diameters can also be calculated by using formula (5), derived in section DDS512-2-c.3.

Using formula (5), the ideal diameter for each duct section in the example problem has been calculated, as follows:

Section	Diameter
A-B	1.29 ft or 15.50 in
B-C	1.29 ft or 15.50 in
C-D	1.22 ft or 14.63 in
D-E	0.98 ft or 11.75 in
E-F	0.89 ft or 10.64 in
F-G	0.73 ft or 8.81 in

Step (k) Determine actual size and shape of duct to be used in the system. If round duct is required, use the nearest standard diameter then determine the f-value associated with that diameter. If rectangular or flat oval ducts are required, determine a size equivalent to calculated diameter using the front bottom scales on the Duct Size Calculator, reference (b), or tables 6-2 and 6-3 in reference (d). The actual size and shape of ducts used in this example are given in column 8 of table 1. The values of the actual friction factors for the selected ducts should be determined (using the Duct Size Calculator) and recorded in column 9 of table 1.

Note: the friction factor used to size rectangular and flat oval duct is the same as that used to determine the sizes of round ducts.

Based on the calculated, theoretical duct diameters and practical duct construction considerations, the duct sizes selected for the sample problem are as follows:

Section	Size	Area
A-B	16" x 12"	192 sq. in.
B-C	16" x 12"	192 sq. in.
C-D	14" x 12"	168 sq. in.
D-E	11" x 10"	110 sq. in.
E-F	10" x 9"	90 sq. in.
F-H	8" x 8"	64 sq. in.

Step (1) Determine the actual velocity in each duct section and each equipment item to make sure that the velocity limits stated in reference (a) have not been exceeded. The velocity in any duct section is calculated by using either formula (3) or formula (3A).

In the example system, the dimensions of duct section A-B are 16 inches by 13 inches and the air flow rate is 4100 cfm. The velocity is calculated as follows:

$$V = \frac{144(4100)}{16 \times 12} = 3075 \text{ fpm}$$

The velocity does not exceed the maximum velocity limit set in the general specifications (reference a) of 3500 fpm for rectangular duct.

After the air flow velocity has been calculated, the velocity pressure, p_v, developed in each section of duct must be calculated using equation (2).

Record the velocity pressure for each section of duct in column 10 of table 1. Velocity pressures should be calculated for the actual duct velocities calculated above.

The calculated velocities and velocity pressures for each section of duct in the example problem are as follows:

Section	Velocity	Velocity Pressure
A-B	3075 fpm	0.59 in wg
B-C	3075 fpm	0.59 in wg
C-D	3043 fpm	0.58 in wg
D-E	2684 fpm	0.45 in wg
E-F	2560 fpm	0.41 in wg
F-H	2250 fpm	0.32 in wg

Step (m) Determine the correct values for all pressure loss coefficients. These values should be based on the actual duct sizes listed in column 8 of table 1. The new values for pressure loss coefficients should be recorded in column 4 of table 1.

To facilitate the calculations the following assumptions were made:

(1) All diverging transitions (both rectangular and round) have an angle of 10°.

(2) All 90° rectangular elbows without splitters have radius ratio's (r/W) of 1.0.

(3) All 90° rectangular elbows with splitters have radius ratio's (r/W) of 0.60.

(4) All 90° round elbows have radius ratio's (r/D) of 1.5.

(5) The pressure loss coefficients for transitions from square-to-round (or vice versa) are assumed to be the average of the coefficients for round-to-round and square-to-square transitions with similar inlet and outlet areas.

Reference (e) provides some additional simplifications as follows:

(1) All rectangular or round converging transitions have pressure loss coefficient of 0.05 regardless of size.

(2) When a transition involves a change in shape only (areas equal), the pressure loss coefficient is 0.

(3) Calculations for elbow aspect ratio (H/W) are based on actual duct dimensions at the elbow.

Step (n) The total pressure loss for all equipment, duct, and fittings must be calculated and recorded in column 11 of table 1. The total pressure loss for a duct length is calculated as follows:

$$p = Lf \quad (9)$$

The total pressure loss for a fitting is calculated as follows:

$$p = Kp_v \quad (10)$$

This calculation can be simplified by totaling the pressure loss coefficients for all fittings in a duct section (of constant diameter) prior to multiplying by the velocity pressure, p_v. This is possible since the velocity pressure in a duct of constant diameter will also remain constant. Therefore, for ease in calculation, the total pressure loss due to fittings in a duct section will be:

$$p = cp_v \quad (11)$$

where c equals the sum of the pressure loss coefficients, K.

For section A-B the total pressure loss is:

Preheater	= 0.38 in wg
Navy standard filters	= 0.13 in wg
Cooling Coil	= 0.50 in wg
Straight duct = Lf = 30 ft x 0.0132 in wg/ft	= 0.34 in wg
Fittings = cp_v = 1.90 x 0.59 in wg	= 1.12 in wg
Total pressure loss for the section	= 2.53 in wg

All other sections of the sample problem are calculated in a similar manner. The resulting pressure losses are as follows:

Section	Total Pressure Loss
A-B	2.53 in wg
B-C	0.11 in wg
C-D	0.17 in wg
D-E	0.91 in wg
E-F	0.32 in wg
F-G	0.64 in wg

Step (o) Find the available pressure at each branch takeoff by successively deducting the total pressure loss in each section from the available pressure at the beginning of the section. Record these values in column 12 of table 1.

For example, since the pressure at the inlet to the system (point A) is atmospheric pressure and the pressure drop in the duct section A-B is 2.53 in wg, the pressure at point B will be (0 in wg - 2.53 in wg =) -2.53 in wg. Similarly, the pressure at point C will be: The pressure at point B, plus the total fan pressure, less the pressure losses in duct section B-C. That is, the pressure at point C will be (-2.53 in wg + 5.20 in wg - 0.11 in wg =) 2.56 in wg.

Step (p) Check the pressure loss in the final duct section against the available pressure at the beginning of that section to ensure that there is sufficient total pressure to force the supply air into the terminal space. In the example problem the available pressure at point F is 1.16 in wg, and the total pressure loss in section F-G, calculated on basis of $\bar{f}$ = 0.0121 in wg/ft, is 0.64 in wg. If the available pressure at point F was not equal to or greater than the total pressure loss in section F-G, it would be necessary to increase the branch duct diameter and check the validity of the new duct size by repeating steps (m) through (p) using the new dimensions.

The entire preliminary evaluation of flow path A-G is shown in table 1.

The preliminary evaluations of flow paths A-H, A-J and A-K must be performed in the same manner as the preliminary evaluation of flow path A-G just illustrated. The pressure losses calculated for each of these paths are summarized in table 2. Note that flow path A-G with a total pressure loss of 4.68 in wg is the critical flow path, or "longest leg." Therefore, the designs of all duct branches and branch mains will be based on the available pressures calculated for each point on flow path A-G, which can be calculated from the pressure losses given in table 2. Note, too, that the preliminary evaluations of flow paths A-H, A-J, and A-K should be discarded and each of these flow paths should be re-evaluated using the available pressures calculated for flow path A-G.

TABLE 2. TOTAL PRESSURE LOSS, BY FLOW PATH, FOR FOUR POSSIBLE MAIN DUCTS

Flow Path A-G

A = 71.6
L = 98.5 ft
H = 3.35 in wg

A/H = 21.4
L/H = 29.4
f = 0.0127 in wg/ft

SECTION:	Loss:
A-B	2.53 in wg
B-C	0.11 in wg
C-D	0.17 in wg
D-E	0.91 in wg
E-F	0.32 in wg
F-G	0.64 in wg
	4.68 in wg

Flow Path A-H

A = 92.7
L = 92.0 ft
H = 3.43 in wg

A/H = 27.0
L/H = 26.8
f = 0.0108 in wg/ft

SECTION:	Loss:
A-B	2.29 in wg
B-C	0.11 in wg
C-D	0.12 in wg
D-E	0.77 in wg
E-F	0.25 in wg
F-H	0.82 in wg
	4.36 in wg

Flow Path A-J

A = 84.1
L = 85.5 ft
H = 3.35 in wg

A/H = 25.1
L/H = 25.5
f = 0.0118 in wg/ft

SECTION:	Loss:
A-B	2.33 in wg
B-C	0.12 in wg
C-D	0.15 in wg
D-I	0.42 in wg
I-J	1.09 in wg
	4.11 in wg

TABLE 2. TOTAL PRESSURE LOSS, BY FLOW PATH, FOR FOUR POSSIBLE MAIN DUCTS (Continued)

Flow Path A-K

A = 56.2
L = 84.5 ft
H = 3.07 in wg

A/H = 18.3
L/H = 27.5
f = 0.0119 in wg/ft

SECTION:		Loss:
	A-B	2.33 in wg
	B-C	0.12 in wg
	C-D	0.15 in wg
	D-I	0.42 in wg
	I-K	0.92 in wg
		3.94 in wg

2. Designing the branch main.

Note that, by definition (see page 5), there is a branch main which splits from the main duct (longest leg) at point D. This branch main will be one of either flow path D-K or flow path D-J. Both paths must be evaluated to determine which one of the two is the branch main, i.e., the flow path with the greatest total pressure loss.

The procedure for designing the branch main is, in general, the same as that used to design the system main. Basically, steps (c) through (p) are regarded for each flow path to determine the duct sizes required and the total pressure loss of the critical flow path. Note, however, that in:

Step (e) A size 26H heater is required at point I. Total pressure loss for the heater is 0.24 in wg.

Step (h) The pressure available at point D for the branch main is 2.39 in wg. This value is taken from the evaluation of the duct main (flow path A-G), conducted previously.

Step (i) The value of H is the pressure available at point D, less a safety factor (again assumed to be 10 percent) less the pressure losses due to equipment in the duct run being investigated.

The evaluation of flow path D-K is shown in table 3. Flow path D-J should be evaluated similarly. The characteristics of flow paths D-J and D-K are listed in table 4. The total pressure drop through flow path D-J is 1.87 in wg. The pressure drop in rough flow path D-K is also 1.87 in wg. Because both flow paths have identical pressure drops, either flow path can be called the branch main.

3. Designing the branches.

Consider each of the branch ducts separately. The available pressure for each branch is the pressure at the branch entrance, which is given in column 12 of tables 1 and 3. The procedure for sizing branch ducts is similar to that used to design the branch main. Note that since each branch has only one C_g value, that value is also the value of A. Branch duct designs are summarized in table 5.

Note, also, that in some cases the friction factor, f, indicated by the analysis will cause the selection of duct size to be smaller than that allowable by reference (a). In such cases the additional pressure loss necessary for proper system operation will need to be provided through the use of an orificing device. The use of orificing devices is discussed in section DDS512-2-g.

TABLE 3. PRELIMINARY EVALUATION OF BRANCH MAIN D-J

SECTION & CFM	DUCT DEVICE OR FITTING	K PREL.	K FINAL	Cq	L (ft)	DUCT DIMENSIONS (in) PREL	DUCT DIMENSIONS (in) FINAL	f (in wg/ft)	Pv (in wg)	Pt (in wg)	Ttl. Press. avail. at pt. (in wg)
1	2	3	4	5	6	7	8	9	10	11	12
D TO I 1500	Takeoff From Rectangular Split Main	0.29	0.29								
	Transition (Converging)	0.05	0.05								
	90° Elbow (H/W = .80) (r/W=1.0) NOMSPL	0.23	0.23								Point D 2.39
	90° Vaned Turn (R=4.5, S=2.25 ,L=0)	0.15	0.15								
	Straight Duct				15	10x8	10x7	0.0222		0.33	
	c =	0.72	0.72	10.1					0.59	0.42	
										0.76	
I TO J 600	Takeoff From Rectangular Split Main	0.40	0.38								
	Transition (Diverging) (A1/Ao=1.05) (Rectangular to Round)	0.13	0.13								
	Type R Watertight Closure (size R8)									0.06	
	90° Elbow (r/D=1.5)	0.15	0.15								
	90° Elbow (r/D=1.5)	0.15	0.15								
	Transition (Diverging) (A1/Ao=1.74) (Round to Rectangular)	0.13	0.13								Point I 1.63
	Electric Heater (size 26M)									0.26	
	Transition (Converging) (Rectangular to Round)	0.05	0.05								
	Type E Terminal (8.0 in.)	2.33	2.33								
	Straight Duct				5 16	6x6 8.0	7x7 8.0	0.0085 0.0071		0.04 0.11	
	c =	3.34	3.32	32.3					0.19	0.63	
										1.11	

TABLE 4. TOTAL PRESSURE LOSS, BY FLOW PATH, FOR TWO POSSIBLE BRANCH MAIN DUCTS

Flow Path D-J

A = 42.4		A/H = 23.2	
L = 36.0 ft		L/H = 19.7	
H = 1.83 in wg		f = 0.0128 in wg/ft	
SECTION:	D-I	Loss:	0.76 in wg
	I-J		1.11 in wg
			1.87 in wg

Flow Path D-K

A = 14.5		A/H = 9.4	
L = 35.0 ft		L/H = 22.6	
H = 1.55 in wg		f = 0.0169 in wg/ft	
SECTION:	D-I	Loss:	0.76 in wg
	I-K		1.11 in wg
			1.87 in wg

TABLE 5. FINAL EVALUATION OF BRANCH DUCTS

SECTION & CFM	DUCT DEVICE OR FITTING	K PREL.	K FINAL	Cq	L (ft)	DUCT DIMENSIONS (n) PREL	DUCT DIMENSIONS (n) FINAL	f (in wg/ft)	Pv (in wg)	Pt (in wg)	Ttl. Press. avail. at pt. (in wg)
1	2	3	4	5	6	7	8	9	10	11	12
F TO H	Rectangular Split Main	0.10	0.00								
600	Transition (Converging)	0.05	0.05								
	Slotted Duct	2.90	2.64								Point F 1.16
	Straight Duct				10	8x8	8x6	0.0091		0.09	
	c =	3.05	2.69	29.5					0.20	0.54	
										0.63	
I TO K	Rectangular Split Main	0.00	0.00								
900	Transition (Diverging) (A1/Ao=2.29)	0.15	0.16								
	Electric Heater (Size 26H)									0.24	
	Transition (Converging)	0.05	0.05								
	90° Elbow (H/W = 1.67) (r/W=0.60) 3 SPL	0.06	0.06								Point I 1.63
	Transition (Diverging) (A1/Ao=1.31) (Rectangular to Round)	0.13	0.13								
	Navy Standard Diffuser (Size 10)									0.36	
	Straight Duct				5	10x8	10x8	0.0057		0.03	
					15	9x6	8x6	0.0204		0.31	
	c =	0.39	0.40	4.4					0.45	0.18	
										1.11	

TABLE 5. FINAL EVALUATION OF BRANCH DUCTS (Continued)

SECTION & CFM	DUCT DEVICE OR FITTING	K PREL.	K FINAL	Cq	L (ft)	DUCT DIMENSIONS (n) PREL	DUCT DIMENSIONS (n) FINAL	f (in wg/ft)	Pv (in wg)	Pt (in wg)	Ttl. Press. avail. at pt. (in wg)
1	2	3	4	5	6	7	8	9	10	11	12
C TO L 550	Takeoff From Rectangular Split Main	0.54	0.54								
	Transition (Diverging)	0.17	0.30								
	Steam Heater (size 25H)									0.18	
	Transition (Converging)	0.05	0.05								
	90° Elbow (H/W=01.20) (r/W=0.60) 1 SPL	0.25	0.26								Point C 2.56
	Transition (Diverging) (A1/Ao=4.8)	0.25	0.27								
	Commercial Diffuser (12 in. x 12 in.)									0.13	
	Straight Duct				29	6x6	6x4.5	0.0321		0.93	
	c =	1.26	1.42	11.8					0.54	0.77	
										2.01	
E TO O 450	Rectangular Split Main	0.31	0.31								
	Transition (Converging)	0.05	0.05								
	Compounds Elbows (60° to 60° Elbow)(Same Plane, Change in Direction Less Than 90°)(H/W=0.67)(L/D=0)	0.21	0.24								
	Elbow (120°)(H/W=1.0) (r/W=1.0) NO SPL	0.29	0.35								
	90° Elbow (H/W=1.0) (r/W=0.60) 2 SPL	0.11	0.12								Point E 1.48
	Transition (Diverging) (Rectangular to Round)	0.00	0.13								
	Diverging Cone at Discharge (O=16°)	0.28	0.28								
	Straight Duct				14	6x6	5x5	0.0258		0.36	
	c =	1.25	1.48	10.8					0.42	0.62	
										0.98	

DDS512-2-g. **Design Procedure for Multi-Terminal Branch Ducts of Constant Cross-Sectional Area.**

1. Design procedure for branch ducts of constant cross-se-tional area using multiple Navy standard diffusing termina .

(a) aboratory tests have established the following 5 uidelines for designing branch ducts of this type:

1) The optimum throat velocity for diffusers in branch ducts of this type is 1,800 fpm. Since it is rarely possible to obtain this optimum velocity, the ranges shown in figure 53 of reference (e) should be used.

2) The optimum air distribution is accomplished when the ratio of static pressure to velocity pressure at a point 2 feet upstream of the terminal nearest to the fan is greater than 0.5. Using this ratio, the following formula was developed for a system where each terminal is designed to deliver air at the same flow rate:

$$p_{vm} = \frac{p_t}{(1.05)(N)^2 + 0.45} \qquad (12)$$

here:

p_{vm} = maximum velocity pressure in branch preceding last terminal

p_t = required pressure 2 feet upstream of last terminal

N = number of terminals to be installed in the branch duct

This formula neglects duct friction loss between terminals. However, unless extreme savings of space or materials is desired, this formula will provide duct sizes which are satisfactory.

3) The pressure loss in the main due to a terminal connection is:

$$p = 0.45(p_{vu} - p_{vd}) \qquad (13)$$

where:

p_{vu} = velocity pressure in duct 2 feet upstream of terminal

p_{vd} = velocity pressure in duct 4 feet downstream of terminal

(4) A length of straight duct of no less than 6 feet is required between adjacent terminals.

(5) The height of the supply duct must be no less than 80 percent of the terminal throat diameter.

(b) Procedure for sizing the branch is as follows:

(1) Lay out the branch to scale. The total air flow rate, Q, to be delivered by the branch should be determined. The area and shape of the space served by the branch duct will generally determine N, the total number of terminals required. The total air flow rate should be divided by N to determine the air flow rate through each terminal. The curves in figure 53 of reference (e), should, then, be used to select the terminals' size.

(2) Determine p_t from figure 53, reference (e).

(3) Using formula (12), determine the maximum velocity pressure, p_m, in the branch at a point preceding the last terminal.

(4) Determine the size of the branch duct. Note that the terminal size limits the duct width, since that dimension cannot be less than the diameter of the terminal flange. Also, the minimum duct height must be no less than 80 percent of the terminal throat diameter. Care should also be taken to ensure that the duct velocity limits of reference (a) are not exceeded and that the duct width does not exceed 8.5 times the duct height.

(5) Determine the pressure required at a point 2 feet upstream of the next to last terminal. This pressure should equal the sum of: p_t, defined above; the pressure loss due to friction between the last and next to last terminals; and the dynamic loss due to the take-off for the next to last terminal. When calculating this value, it is important that the designer remember that the total pressure loss associated with a 6 foot length of duct (2 feet upstream and 4 feet downstream of any terminal) is included in the calculated value of the terminal pressure loss, i.e., the value

obtained from equation (13). Therefore, the length of duct between any two terminals is equal to the measured distance between the take-off centerlines, less 6 feet. The friction factor, f can be obtained from reference (b). Losses due to special fittings may be obtained from reference (e). The loss at the terminal take-off (due to the "air scoop" fitting and the reduction in air velocity) is calculated using equation (13). The values of p_{vu} and p_{vd} should be the actual upstream and downstream velocities, based on the actual duct dimensions.

(6) Steps (2) through (5) should be repeated for each successive terminal until the pressure required at a point 2 feet upstream of the terminal nearest to the main has been determined.

(7) The ratio of the static pressure to the velocity pressure at the point 2 feet upstream of the terminal nearest the duct main should, then, be calculated. This ratio must be greater than 0.5. Remember that the static pressure at any point is the difference between the total pressure (calculated in step (5), above) and the velocity pressure at the point.

(8) To ensure that the air flow rate through each terminal is equal, balancing orifices will need to be installed in the throat of all but the last supply terminal. Orifices are required to ensure that the total pressure within the neck of each terminal is identical. This will, in turn, ensure that the flow through each terminal will be identical. Orifice sizing data is given in figures 3 through 11.

The entrances to orifices should be faired. For throat diameter reductions up to 1/2 inch, it is recommended that a properly sized wire, secured to terminal throat be used as an orifice. For throat diameter reductions greater than 1/2 inch, orifice plates with rounded entrances or a combination of plates and 1/4-inch diameter wires may be utilized.

(9) The pressure in the duct main supplying multi-terminal branches must be greater than the sum of the calculated pressure at a point 2 feet upstream of the first terminal and the friction and dynamic losses developed in the branch ductwork between the branch take-off and that point. If the pressure in the main at the branch take-off is less than the required pressure, it will be necessary to redesign the branch duct.

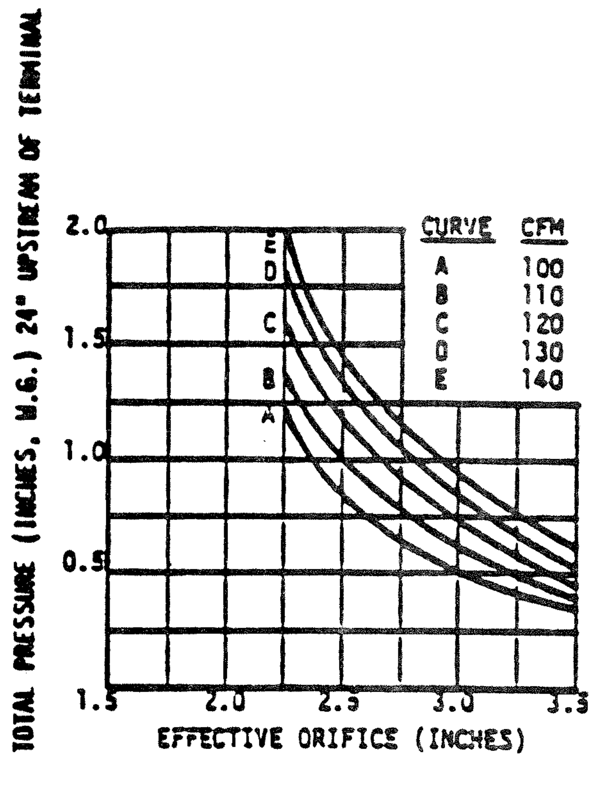

Figure 3. Effective Orifices for Size 3 1/2 Navy Standard Diffusing Terminals

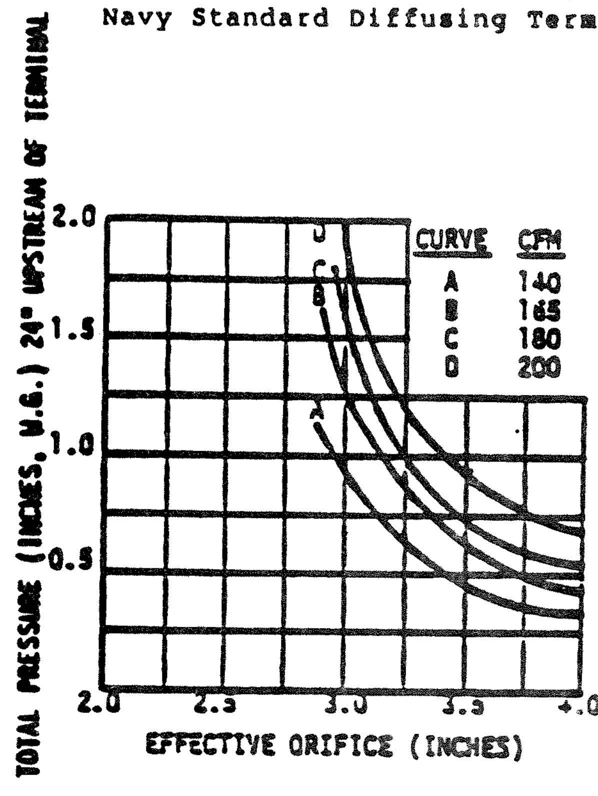

Figure 4. Effective Orifices for Size 4 Navy Standard Diffusing Terminals

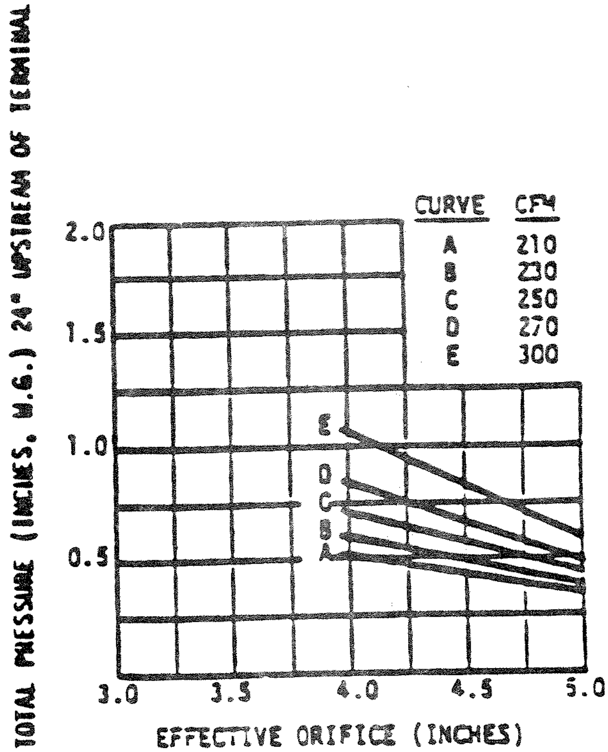

Figure 5. Effective Orifices for Size 5 Navy Standard Diffusing Terminals

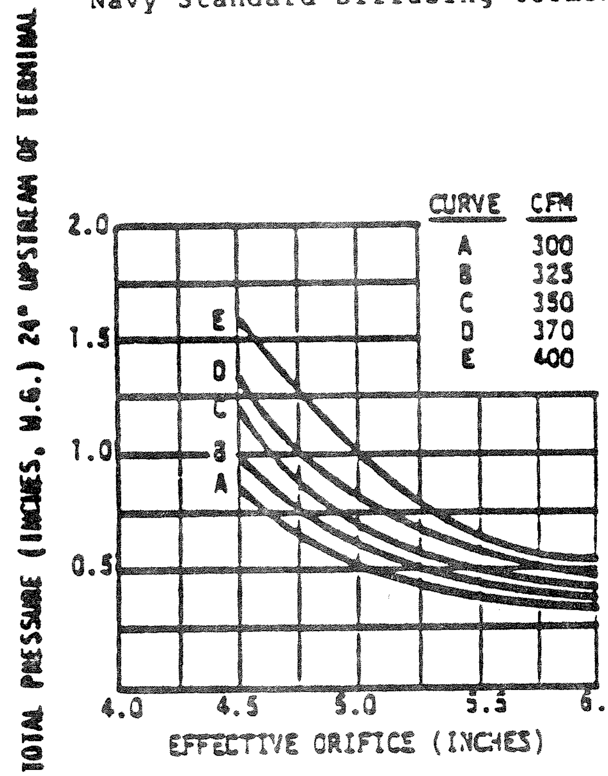

Figure 6. Effective Orifices for Size 6 Navy Standard Diffusing Terminals

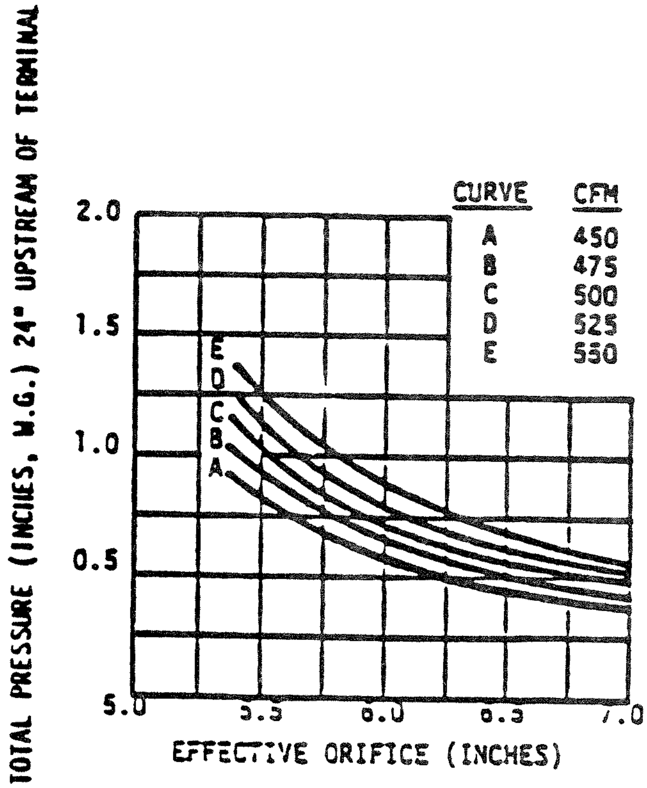

Figure 7. Effective Orifices for Size 7 Navy Standard Diffusing Terminals

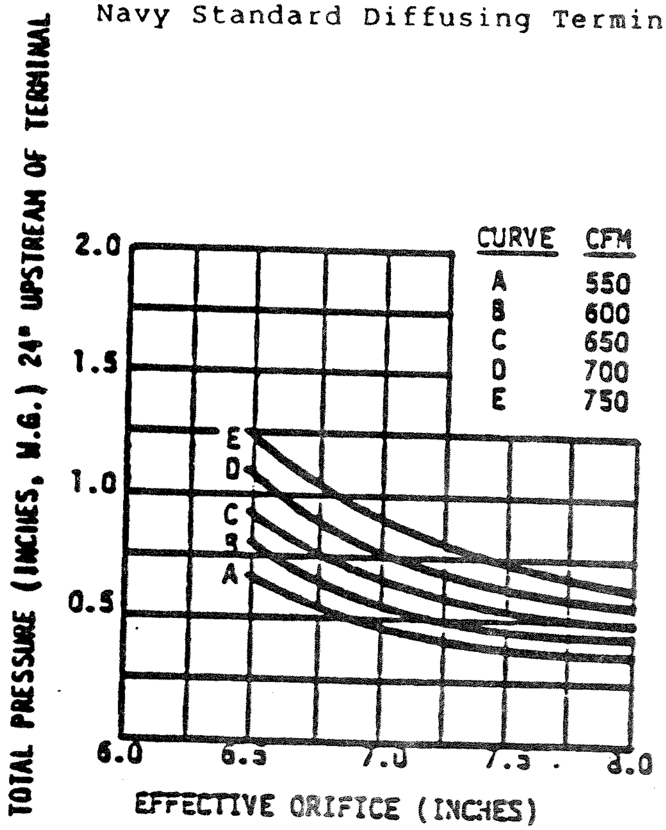

Figure 8. Effective Orifices for Size 8 Navy Standard Diffusing Terminals

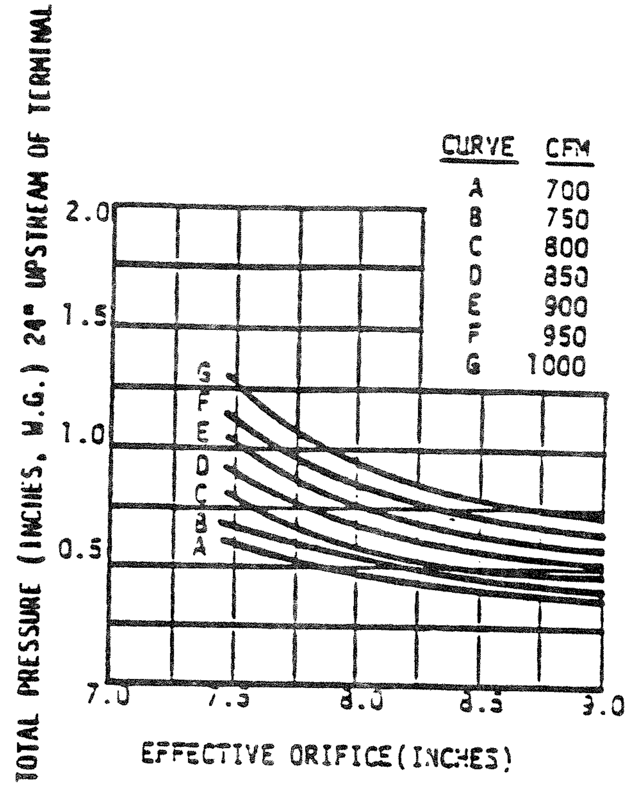

Figure 9. Effective Orifices for Size 9 Navy Standard Diffusing Terminals

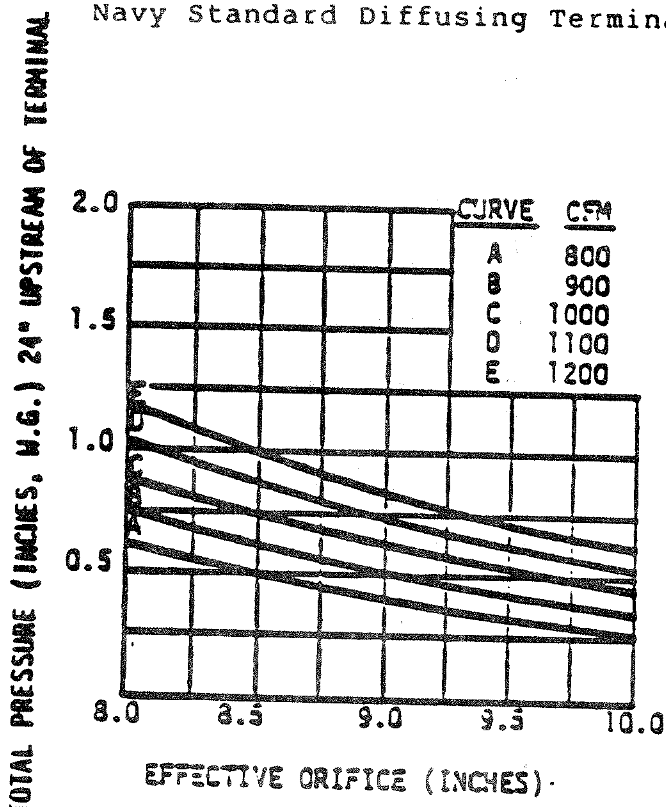

Figure 10. Effective Orifices for Size 10 Navy Standard Diffusing Terminals

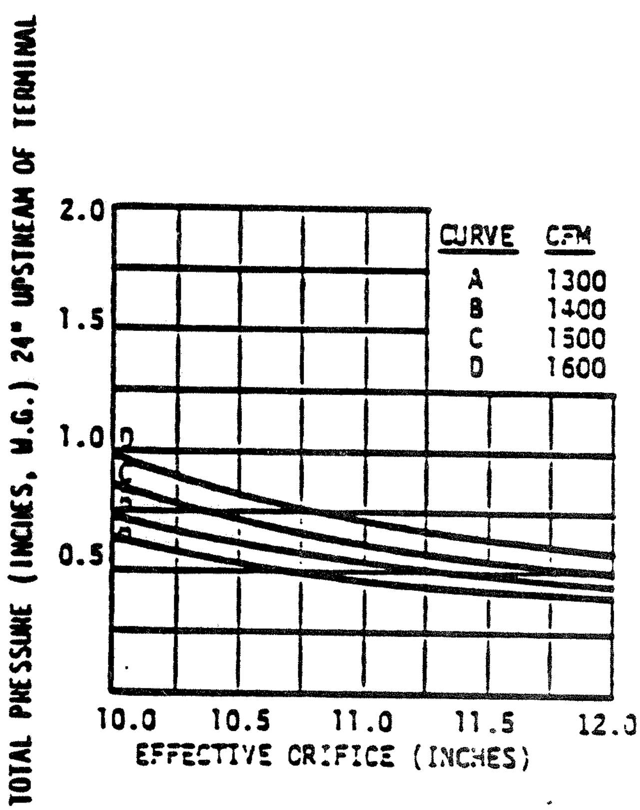

Figure 11. Effective Orifices for Size 12 Navy Standard Diffusing Terminals

DDS512-2-h. **Example Problem** - Design of a **Multi-Terminal** Branch **Duct**

1. The alternate branch duct, C-M, shown in figure 2 is designed as follows:

Assume that alternate branch C-M is installed at point C (figure 2) instead of branch C-L. Assume, also, that branch C-M will be designed to distribute 1,000 cfm through four Navy standard diffusing terminals attached to a duct with a constant cross section.

Step 1. A schematic of the branch duct indicating the locations of all terminals, fittings, etc., is shown in figure 12. The air flow rate through each of the four terminals will be:

$$\frac{1000 \text{ cfm}}{4 \text{ terminals}} = 250 \text{ cfm/terminal}$$

From figure 53, reference (e), a diffusing terminal of size No. 5 is selected. Because the width of the duct must not be less than the diameter of the terminal flange (which, in this case, is 6.5 inches) and because the depth of the constant cross-section portion of the duct must not be less than 0.8 times the throat diameter of the terminal (i.e., 0.8 x 5 = 4 inches), the constant cross-section portion of the branch duct must be at least 6.5" wide X 5" high.

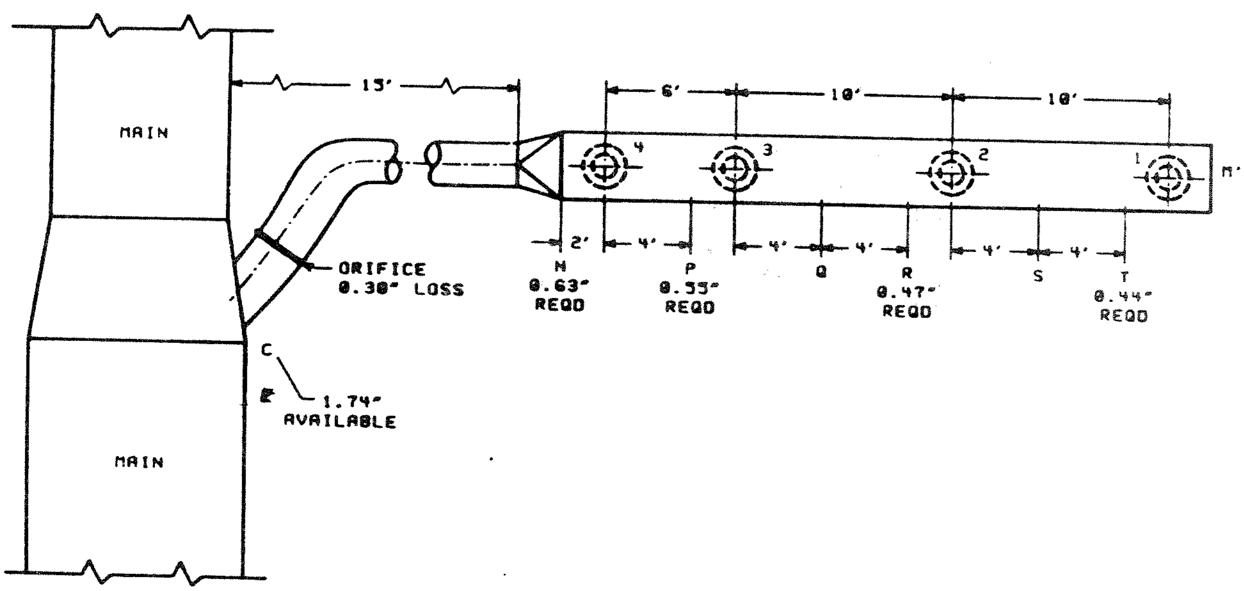

Figure 12. Alternate Branch C-M Using Navy Standard Diffusing Terminals

Step 2. Determine p_t (pressure required at point T, 2 feet upstream of last terminal). From figure 53, reference (e).

$$p_t = 0.44 \text{ in wg.}$$

Step 3. Determine the maximum velocity pressure at point T by substituting the known values into equation (12):

$$p_{vm} = \frac{p_t}{(1.05)(N)^2 + 0.45}$$

$$= \frac{0.44}{(1.05)(4)^2 + 0.45} = 0.025 \text{ in}$$

From reference (e) or equation (2), the velocity corresponding to a velocity head of 0.025 in wg. is 640 fpm. At this velocity, duct area can be calculated to be 56.2 in^2. Since the minimum duct dimensions (calculated in step (1)) are 6.5" x 5", a duct measuring 10" x 6" (60 in^2 > 56.2 in^2) is selected for the constant cross-section portion of the branch duct. The velocity, V, in this duct will be 600 fpm and the equivalent diameter, D_e, will be 8.4 inches.

Step 4. Determine the pressure required at point R, which is 2 feet upstream of the next to last terminal. This will be the sum of the following losses:

1. Pressure required 2 feet upstream of terminal No. 1 (from step 2) = 0.44 in wg

2. Pressure loss in straight duct due to friction. At 250 cfm, the friction factor, f, in a 10" x 6" duct will be f = 0.001 in wg/ft. The length, L, of the duct in example problem is 10 ft (center-line-to-centerline) distance, less 6 ft (see paragraph 512-2-g), or 4 ft. Therefore, the pressure loss due to straight duct from S to T, p_{ST} = Lf, or 4 ft x .001 in wg/ft = 0.00 in wg

3. Loss across terminal No. 2
(Equation (13))
$P = 0.45\ (0.090 - 0.025)$ $= 0.03$ in wg

The pressure required at point R is the sum of 1., 2., and 3., above. Therefore, $p_R = 0.47$ in wg.

Step 5. Determine required pressure at point P by adding losses for section P-R to the pressure required at R. Similarly, determine pressure required at point N. The pressure required at point N is calculated to be 0.63 in wg.

Step 6. Check to determine if the ratio of static pressure (p_s) to velocity pressure (p_v) at point N is greater than 0.50 as follows:

$$\frac{p_s}{p_v} = \frac{p_t - p_v}{p_v} = \frac{0.63 - 0.39}{0.39} = 0.62 > 0.50$$

Step 7. Using figure 5, determine the sizes of the orifices to be installed in the throats of terminals 2, 3, and 4. The selected orifices are listed in the four right-hand column of table 6.

Step 8. Size section C-N as follows:

Pressure available for branch = 2.56 both in wg. The known losses for the constant cross-section portion of the branch (N-M) = 0.63 in wg. Therefore, the available total pressure factor, H, for the branch duct section C-N will be: 2.56 - 0.63 = 1.93 in wg

Velocity pressure loss coefficient assumptions for fittings:
0.40 = 30° branch takeoff loss
0.12 = 60° elbow loss
0.20 = transition loss
0.72 = c

At 1,000 cfm, for c = 0.72, $C_q = 8.5 = A$

Straight duct = 15 ft = L

$$\frac{A}{H} = \frac{8.5}{1.93} = 4.4$$

$$\frac{L}{H} = \frac{15.0}{1.93} = 7.8$$

Therefore, the friction factor for section C-N should be:

f = 0.07 in wg/ft

For 1,000 cfm, f = 0.07 in wg/ft, the required duct size is 6.2 inches and, the nearest stock size duct is a 7-inch diameter duct. This results in a velocity of 3,800 fpm, which exceeds the velocity requirements (3,500 fpm maximum) of reference (a). At 3,500 fpm, the required diameter is 7.3 inches. For this example, assume the nearest standard size duct diameter, which is 7.5 inches, will be used. The pressure losses in section C-N can be calculated, as follows:

Known:

Size	= 7.5 inch diameter
Area	= 44.2 sq.in.
Air flow rate	= 1,000 cfm
Velocity	= 3,260 fpm
p_v	= 0.66 in wg.
f	= 0.027 in wg/ft
L	= 15 ft

Actual velocity pressure loss coefficients for fittings:

30° branch takeoff	= 0.26
60° elbow	= 0.12
Transition	= 0.15
	0.53

Loss for fittings = cp_v =	0.53 x 0.66	= 0.35 in wg
Loss for straight duct =	15 x 0.027	= 0.40 in wg
Loss for C-N		= 0.75 in wg

Since pressure available for C-N is 1.93 in wg, an orifice designed to dissipate (1.93 - 0.75 =) 1.18 in wg should be installed in the branch takeoff to balance the branch.

TABLE 6. BRANCH C-M

Feature	Straight duct			Loss across terminal							Pressure losses in wg	Pressure in wg req'd at point	Effective orifice req'd at-	
	cfm	L	f	cfm_u	V_u	P_{vu}	cfm_d	V_d	P_{vd}	$0.45 (P_{vu} - P_{vd})$			Terminal	Size
Pressure req'd for terminal No. 1 (from fig. 53, ref (e)											0.44	T=0.44	1	5.00"
Section T-M Straight duct (S-T) Terminal No. 2 (R-S)	250	4'0"	0.001	500	1,260	0.098	250	630	0.025	0.03	0.44 0.00 0.03 0.47	R=0.47	3	4.85"
Section R-M Straight duct (Q-R) Terminal No. 3 (P-Q)	500	4'0"	0.004	750	1,890	0.22	500	1,260	0.098	0.06	0.47 0.02 0.06 0.55	P=0.55	3	4.55"
Section P-M Terminal No. 4 (N-P)				1,000	2,520	0.39	750	1,890	0.22	0.08	0.55 0.08 0.63	N=0.63	4	4.25"

P_{vm} at point N=0.4
Duct size=5" x 11.5"
Equivalent diameter = 8.3"
Velocity upstream of last terminal=630 fpm
Branch size=7.5" diameter

APPENDIX A

ESTIMATED PRESSURE LOSS FOR EXAMPLE PROBLEM

From figure 2, the longest leg appears (by inspection) to be the flow path from A to J. The length of the path is:

A-B	30.0 ft
B-C	9.5 ft
C-D	25.5 ft
D-I	15.0 ft
I-J	21.0 ft
	101.0 ft

Assume that the equivalent length of this duct run is 2.0 times the length of the straight duct, and that the friction factor will be 0.010 in wg/ft of straight duct. Therefore, the estimated pressure loss due to ductwork will be:

$$2.0 \times 101 \text{ ft} \times 0.010 \text{ in wg/ft} = 2.00 \text{ in wg}$$

Next, add the estimated pressure losses for all devices along the selected flow path to the duct loss estimate.

Duct	2.00 in wg
Intake	0.50 in wg
Steam Header	0.50 in wg
Navy Standard Filter	0.50 in wg
Cooling Coil	0.50 in wg
Type "R" Closure	0.10 in wg
Electric Heater	0.10 in wg
Type "E" Terminal	0.20 in wg
	4.40 in wg

Therefore, the estimated total pressure loss for the system will be 4.40 in wg. For convenience, this figure can be rounded off to 4.5 in wg.