DESIGN DATA SHEETS

BUREAU OF ENGINEERING, NAVY DEPARTMENT

FORCED DRAFT BLOWERS, DUCT WORK FOR

1 June 1935

A. REFERENCES.

A-1. General Specifications for Machinery, Subsections S38-4 and S53-1.
A-2. "Fan Engineering" of Buffalo Forge Company.

B. GENERAL DESIGN.

B-1. Forced draft blowers are required to draw air from the upper decks through suitable ducts to the fire rooms or to the boiler casings.

B-2. As regards the design of the blower proper, it is very difficult to predict with any degree of accuracy, by means of calculation, the performance of a new design of blade. Such fans are generally developed by the manufacturers by means of a cut and try process. Also the effect of the intake housing and exit diffuser on fan performance is very great. However, once the performance of a given type of blade, intake and diffuser combination is known under a certain set of circumstances, performance under a different set of circumstances can be accurately predicted by means of calculation. Manufacturers always supply the Bureau with performance characteristics of their fans and it is a very simple matter to calculate the effect of changes in speed, pressure, etc. on this performance as will be shown under "Detail Design".

B-3. The bringing of air from the upper decks to the fire rooms or other points below decks involves considerable calculation centering around the design of the duct work so as to reduce the pressure drop through this space to a minimum. Space is at a premium as is power required to overcome air resistance, therefore, although duct design is not under the cognizance of the Bureau of Engineering, such design must be approved by the Bureau and it is the duty of this Bureau to calculate the resistance to airflow therein, in order that the blower may be properly applied, and the design should be carefully studied for faults.

B-4. All air intakes should be bell mouthed. All right angle bends should contain dividing guides. All sharp turns should be avoided if possible. Pipe sections should be gradually, not sharply enlarged or reduced. When the air enters the fire room or other space where the velocity is very low, a diffuser should always be used.

C. DETAIL DESIGN.

C-1. Air subjected to the pressures ordinarily used in ventilating work is compressed very little, therefore, the laws as apply to the flow of water will apply here, correction being made for the weight of air. For ordinary velocities, the flow will be turbulent and the friction and shock loss will vary as the square of the velocity.

C-2. In order to determine pressure drop through any sort of an air duct it must be realized that the pressure in the duct is made up of two components, one, velocity pressure which is dynamic and due to the Kinetic energy in the moving air; and the other, static pressure, possibly negative, which is due to the potential energy, of the air being confined in the duct at a higher or lower pressure than atmospheric. When air starts through a duct which has a tortuous path, and the section of which changes, the values of velocity and static pressure will change, rising and falling, as one is converted into the other depending on conditions, but the total will continually decrease due to the losses. The problem is to determine what the change in heads or pressures is in each portion of the duct. The pressures finally resulting at the fan indicate fan performance necessary.

NOTATION

v = velocity of air in feet per minute.
P_v = velocity head or pressure in inches of water.
P_s = static head or pressure in inches of water.
P_t = total head or pressure in inches of water.
w = weight of air in lbs. per cubic foot.
C.F.M. = cubic feet volume of air per minute.
R.P.M. = revolutions per minute.
H.P. = horsepower.

FORMULAS

$$v = 1096.5 \sqrt{\frac{P_v}{W}}$$

For "Standard Air", that is 68°F and 29.92 barometer, $W = .07488$ lbs. per cu.ft. and then ve becomes

$$v = 4005 \sqrt{P_v} \qquad \text{or} \qquad P_v = \frac{v^2}{1.61 \times 10^7}$$

$$P_t = P_v + P_s$$

As regards the performance of any fan:
Capacity or C.F.M. is directly proportional to speed or R.P.M.
Pressure, P_s, is proportional to the square of the speed or R.P.M.
Horsepower is proportional to the cube of the speed or R.P.M.
These three effects take place simultaneously.
Assuming the sub-letters to denote different speeds, then

$$\frac{(C.F.M.)_1}{(C.F.M.)_2} = \frac{(R.P.M.)_1}{(R.P.M.)_2}$$

$$\frac{P_{s1}}{P_{s2}} = \frac{(R.P.M.)_1^2}{(R.P.M.)_2^2}$$

$$\frac{(H.P.)_1}{(H.P.)_2} = \frac{(R.P.M.)_1^3}{(R.P.M.)_2^3}$$

Total pressure of fan equals	(Static pressure at fan outlet. (+ velocity pressure at fan outlet. (+ static pressure (draft) at fan inlet (- velocity pressure at fan inlet.
Static pressure of fan equals	(Static pressure at fan outlet. (+ static pressure (draft) at fan inlet. (- velocity pressure at fan inlet.

C-3. One of the greatest losses is due to the shock of the moving air entering the more or less stationary air of the fire room. To minimize this shock and energy loss due to the impact the air should be led into the fire room, or other space to be ventilated, at low speed. This speed reduction is accomplished most efficiently by means of a diffuser. The diffuser should be proportioned according to the correct rate of change of speed.

C-4. The following expression covers a diffuser of rectangular section. However, the area is the governing feature and, regardless of shape, circular or what not, the area can be made equivalent to that determined by the method following.

C-5. In a rectangular diffuser or length, ℓ, having one pair of sides parallel, if the half-breadths at the small and large ends are respectively y_1 and y_2, the half-breadth, y, at the distance, x, from the small end is given by

$$\frac{1}{4\sqrt{y}} = \frac{1}{4\sqrt{y_1}} - \frac{x}{\ell}\left[\frac{1}{4\sqrt{y_1}} - \frac{1}{4\sqrt{y_2}}\right]$$

Figures 1 to 5 show the losses to be expected in ordinary duct work. Converging nozzle loss is not shown because this loss is so small it may be neglected where the total angle of convergence is not greater than 45°.

It will be noted that Fig. 3 shows friction loss in circular pipes. In the case of rectangular pipes, the value of the expression $\frac{2ab}{a+b}$ where a and b are the two sides, may be substituted for the diameter on the curve.

C-6. This curve, Fig. 3, covers ordinary galvanized iron swedge piping or duct work.

C-7. Armored grating resistance is not shown but this may be taken as 50 per cent P_v loss for gratings with circular holes with rounded edges. In other words, the gain in velocity head which depends on the size and number of holes should be calculated and half of this taken as the loss.

C-8. The existing practice is for the boiler manufacturer to guarantee satisfactory operation of his equipment with a fire room pressure of a certain amount, generally eight inches of water. Of course part of this pressure is absorbed in the uptakes, armored gratings and smoke pipes, but this is generally of the order of 0.5 inches of water or less than ten per cent.

C-9. The amount of air necessary is calculated on the basis of 300 cu.ft. per pound of oil when the boilers are operating at the 20% overload condition and the corresponding air pressure. The weight of oil which must be consumed is obtained from ship's contract and boiler manufacturer's guarantee which is made on the basis of the evaporation rate per pound of oil. This appears on the boiler plans.

C-10. Knowing the volume or amount of air and the area of entrance to the air intake, the velocity and the corresponding velocity pressure can be calculated. It should be borne in mind that all of the energy imparted to this air is supplied by the fan and that this velocity pressure is part of the total pressure which the fan must exert. The static pressure at the intake entrance is, of course, atmospheric or zero inches of water (gauge). From this point on we may calculate changes in pressure and add or subtract them, as proper, until we reach the fan or fire room. This is shown in the sample calculation to follow.

D. SAMPLE CALCULATIONS.

D-1. SAMPLE CALCULATION NO. 1.

D-1a. It is desired to bring 56,500 cu.ft. of air to each of three fire rooms on an aircraft carrier. These fire rooms are situated adjacent to each other and all the air is brought through a common trunk to a point just above the three fire rooms where the air is distributed. The sketch, Fig. 6, shows the general arrangement and dimensions of the inlet duct. At the point, D, the division of air takes place. The furthest fire room is at G, some 40 feet away and this should consequently entail the greatest entrance loss. The required static pressure in the fire rooms is 6.8 inches of water.

D-1b. What should be the characteristics of the blower situated in each fire room?

D-1c. The calculation of the duct loss or resistance follows for each section. Under each letter heading, the loss is calculated up to that letter point of the sketch. It will be noted that under A, the entrance loss is given as velocity pressure. The energy required to accelerate the air up to this velocity must, of course, be supplied by the fan.

A.

$v_a = 1540$ ft./min., $\qquad P_{v_a} = \dfrac{\overline{1540}^2}{1.61 \times 10^7} = 0.147$ inches.

B.

Entrance Loss = $0.5\, P_{v_a} = 0.5 \times 0.147 = 0.0735$ inches.
See Fig. 1.
See Fig. 3, 10.5 ft. = 126 inches.
0.028 inches of water per 100 ft.
$0.12 \times 0.028 = 0.00336$ inches of water drop.
$P_{s_b} = 0 - 0.00336 - 0.0735 = -0.07686$
$P_{v_b} = 0.147$

C.

See Fig. 5. This is half of a right angle bend with a 50% radius.
Loss is one half of 95% or 47.5% velocity head
$0.475 \times P_{v_b}$ = loss.
$0.475 \times 0.147 = 0.07$ inches of water.
Loss through the duct is
$0.14 \times 0.028 = 0.00392$ inches of water.
$P_{sc} = -0.07686 - 0.07 - 0.00392 = -0.15078$
$P_{v_c} = 0.147$

D.

Half of right angle bend. This completes the other half so pressure drop is 0.07 inches of water.

Equivalent diameter of duct is

$$\frac{2\,ab}{a + b} = \frac{2 \times 10 \times 13}{10 + 13} = \frac{260}{23} = 11.3 \text{ feet.}$$

or 136 inches.

See Fig. 3. 0.022 inches water per 100 ft.

0.09 x 0.022 = 0.00198 inches water drop.

$$P_{v_d} = \frac{\overline{1270}^2}{1.61 \times 10^7} = 0.1 \text{ inches.}$$

Since there is no diffuser it is assumed that the change in velocity head is completely lost, there being no static regain.

This loss is $P_{v_c} - P_{v_d} = 0.147 - 0.1 = 0.047$

$$P_{s_d} = -0.15078 - 0.07 - 0.00198 - 0.047$$

$$P_{s_d} = -0.26976$$

E.

Assume the change in section plus the right angle bend to result in velocity head loss of the elbow effect only which would be 95% velocity head (refer to Fig. 5) then

0.95 x 0.1 = 0.095 loss.

Loss due to change in velocity head is difference between velocity heads at D and E.

New velocity head is $P_{v_e} = \dfrac{\overline{2350}^2}{1.61 \times 10^7} = 0.343$ inches.

Old velocity head was $P_{v_d} = 0.1$

Loss, which should be substracted from static head is $P_{v_e} - P_{v_d} =$

$0.343 - 0.1 = 0.243$ inches.

$$P_{s_e} = -0.26976 - 0.095 - 0.243 = -0.608$$

F.

In entering the large area, all velocity head is lost. The friction drop through down to the armor grating may be neglected. At the armor grating entrance the velocity is

$$v_f = \frac{56{,}500}{30} = 1885 \text{ ft./min.}$$

$$P_{v_f} = \frac{\overline{1885}^2}{1.61 \times 10^7} = 0.22 \text{ inches.}$$

$$P_{s_f} = -0.608 - 0.343 = -0.951$$

G.

Velocity through grating.

$$v = \frac{56{,}500}{17} = 3325 \text{ ft./min.}$$

Change in velocity head is as follows:

$$P_v = \frac{\overline{3325}^2}{1.61 \times 10^7} = 0.688 \text{ inches.}$$

$P_v - P_{v_f} = 0.688 - 0.22 = 0.468$ change.

Half of this is lost.

$$\frac{0.468}{2} = 0.234 \text{ loss in static.}$$

$P_{s_g} = -0.951 - 0.234 = -1.185$

$P_{v_g} = 0.22$ inches.

This is at entrance to blower.

$Pt_g = P_{s_g} + P_{v_g} = 1.185 + 0.22 = 1.405$ inches.

Static pressure at outlet of fan must be 6.8 inches from specifications.

Referring to "Detail Design" it is seen that

Static pressure of fan equals (Static pressure at fan outlet (+ static pressure (draft) at fan inlet (- velocity pressure at fan inlet.

Therefore, static pressure of fan must be

$6.8 + 1.185 - 0.22 = 7.765$ inches of water.

D-2. <u>SAMPLE CALCULATION NO. 2</u>.

D-2a. Referring to the problem under sample calculation No. 1, suppose it should be desired to increase the amount of air per boiler to 68,000 cfm. What would be the per cent increase in speed at which the blower would have to run? What would be the static pressure in the fire room? What percentage increase in horsepower would be required? Conditions of inlet and discharge of blowers remaining constant.

$$\frac{68{,}000 \text{ cfm}}{56{,}500 \text{ cfm}} = 1.205 \text{ or}$$

20.5% increase in speed.

$(1.205)^2 (6.8) = 9.86$ inches of water pressure in fire room.

$(1.205)^3 = 1.75$ or 75% increase in horsepower.

D-3. <u>SAMPLE CALCULATION NO. 3</u>.

D-3a. A vertical trunk 4 x 4 feet is used to conduct 75,000 C.F.M. of air. The air enters through a 4 x 4 ft. opening in the side of this trunk, making a right angle turn and thence downward.

D-3b. What would be the difference in loss of entrance and downward turn should a bell mouthed entrance be added together with three inside dividing guides, spaced one foot apart?

Area of entrance is 16 sq.ft.

$$\frac{75{,}000}{16} = 4690 \text{ ft./min.}$$

$$P_v = \frac{\overline{4690}^2}{1.61 \times 10^7} = 1.365 \text{ inches of water.}$$

Refer to Fig. 1.

Half of P_v is lost in static pressure.

$0.50 \times 1.365 = 0.683$ inches static loss.

Loss in the right angle turn.

See Fig. 5.

Radius is 50% of pipe width. Loss is 95% of P_v.

$0.95 \times 1.365 = 1.28$ inches.

Total static pressure loss.

0.683 + 1.28 = 1.963 inches of water.

When bell mouth and dividing guides are added.

Entrance loss is zero. See Fig. 1.

The dividing guides make the entrance into four parallel elbows with center line radii of 3.5 ft., 2.5 ft., 1.5 ft. and 0.5 ft. See Fig. 5.

Call the elbows A, B, C, D for the radii 3.5, 2.5, 1.5 and 0.5 ft.

A, $\frac{3.5}{1}$ = 350%, Head loss = 7.5%

B, $\frac{2.5}{1}$ = 250%, Head loss = 7.5%

C, $\frac{1.5}{1}$ = 150%, Head loss = 9%

D, $\frac{0.5}{1}$ = 50%, Head loss = 95%

Head loss varies as the square of the velocity and since these elbows are all parallel, the drop or loss must be the same for each. Let v_a, v_b, etc. be the velocities through each elbow. Then

$$0.95\, v_D^2 = 0.09\, v_c^2 = 0.075\, v_b^2 = 0.075\, v_a^2$$

$$12.67\, v_d^2 = 1.2\, v_c^2 = v_b^2 = v_a^2$$

$$3.56\, v_d = 1.096\, v_c = v_b = v_a$$

$$\frac{\frac{1}{3.56} + \frac{1}{1.096} + 1 + 1}{4} = 0.80\, v_a = \text{average velocity.}$$

4690 ft./min. = average velocity.

$v_a = \frac{4690}{0.80} = 5850$ ft./min.

$v_b = 5850$ ft./min.

$v_c = \frac{5850}{1.096} = 5350$ ft./min.

$v_d = \frac{5850}{3.56} = 1640$ ft./min.

$P_{v_a} = \frac{\overline{5850}^2}{1.61 \times 10^7} = 2.13$ inches of water.

0.075 x 2.13 = 0.16 inches static pressure drop.

P_{v_b} = ditto

$P_{v_c} = \frac{\overline{5350}^2}{1.61 \times 10^7} = 1.775$ inches of water.

0.09 x 1.775 = 0.16 inches water drop.

$P_{v_d} = \frac{\overline{1640}^2}{1.61 \times 10^7} = 0.168$ inches of water.

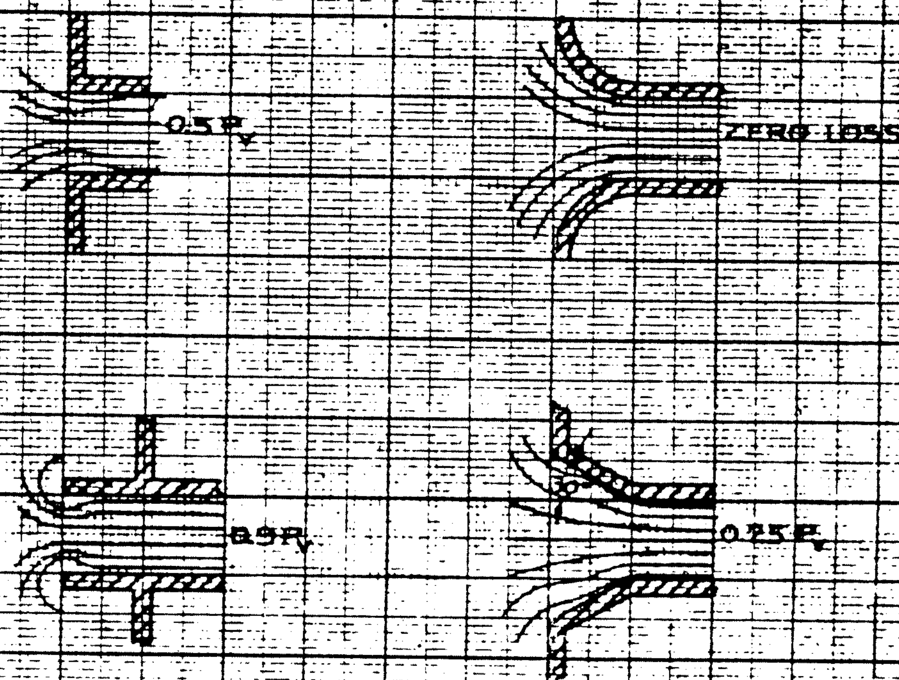
0.95 x 0.168 = 0.16 inches water loss.

Total static pressure loss

$0 + 0.16 = 0.16$ inches of water.

This is to be compred with the 1.963 inches loss when the pressure saving devices are absent.

ENTRANCE LOSS IN PIPES



LOSS SHOWN AS FACTOR OF P_v, WHERE P_v IS VELOCITY HEAD AT ENTRANCE.

FIGURE 1

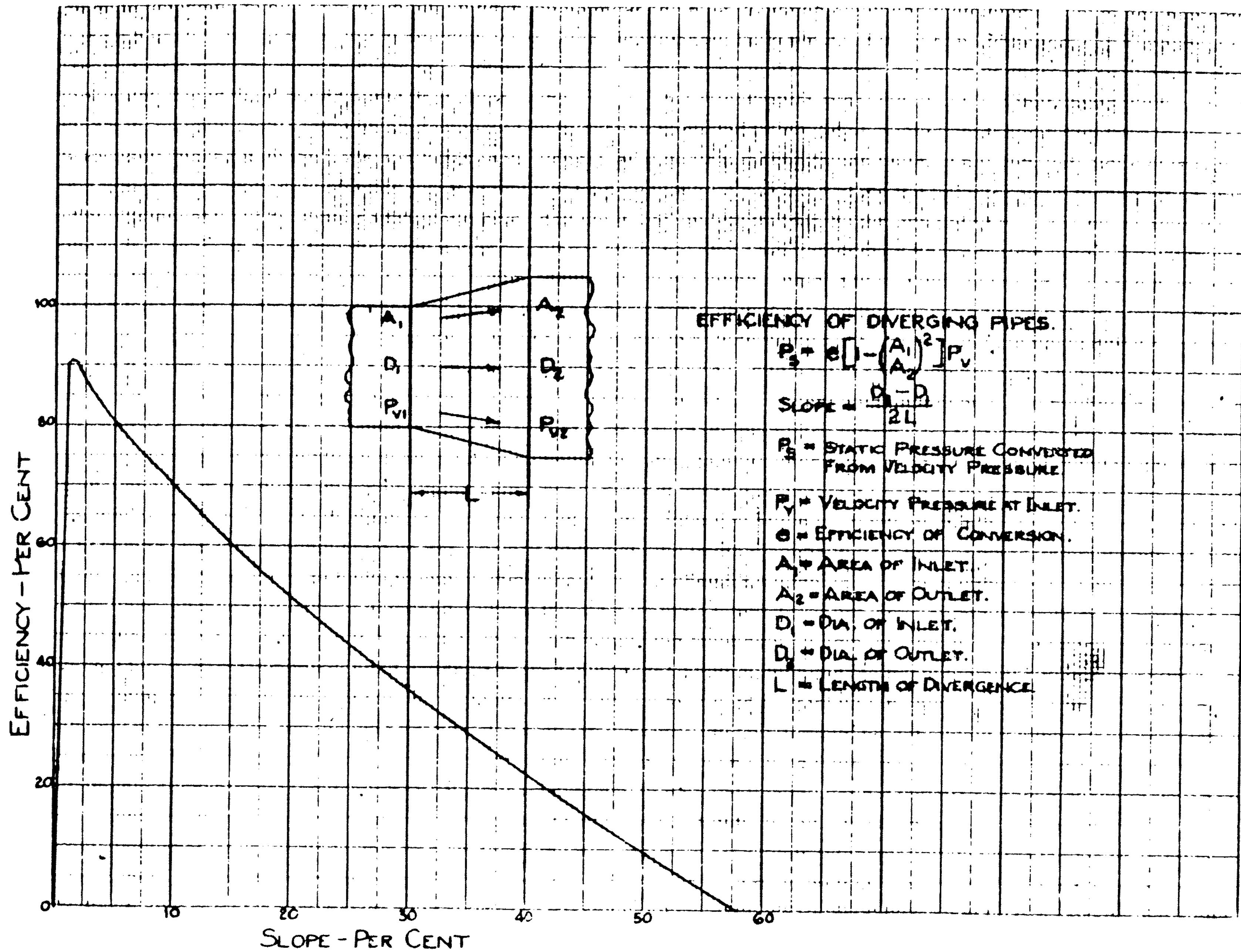
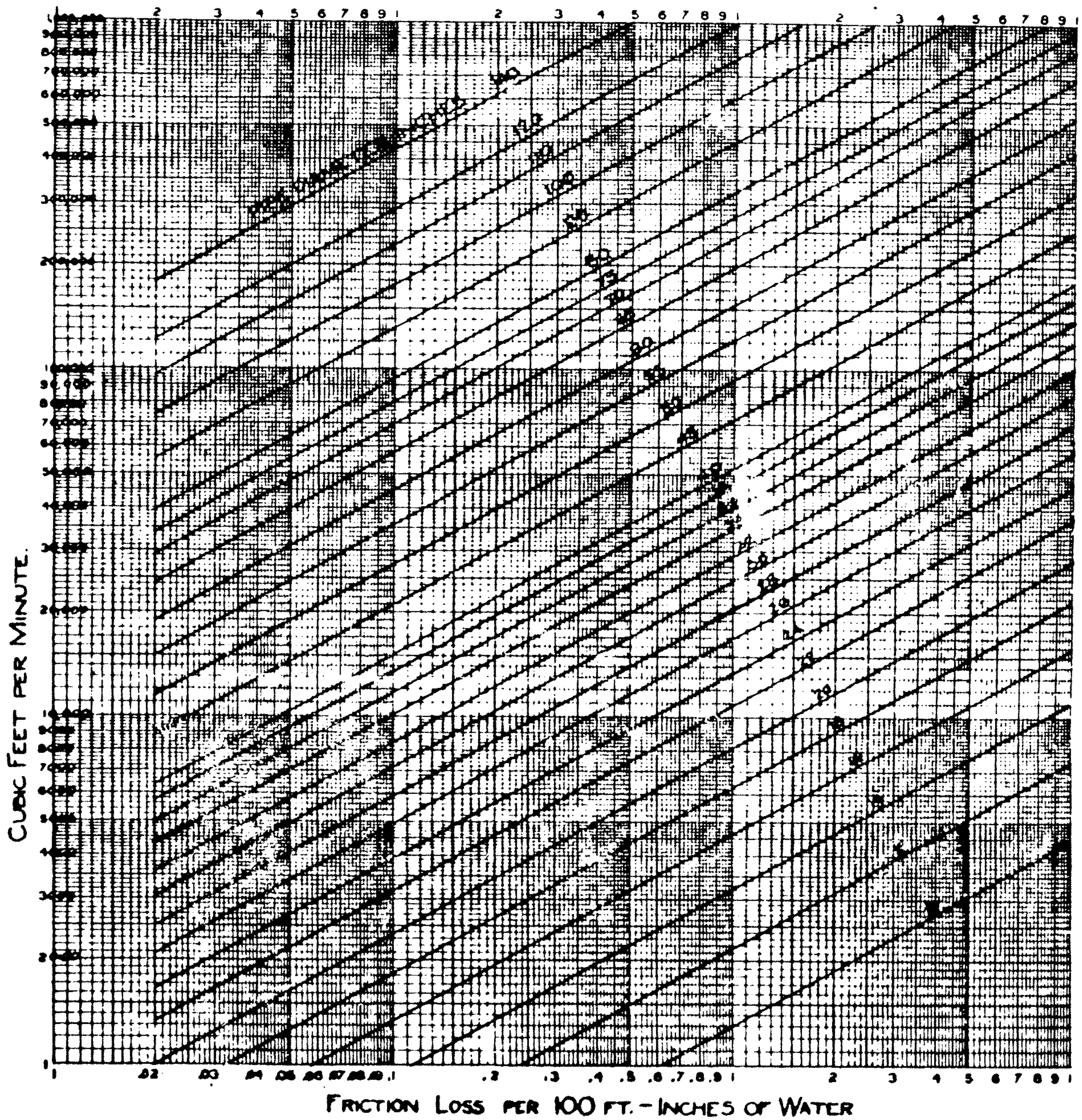


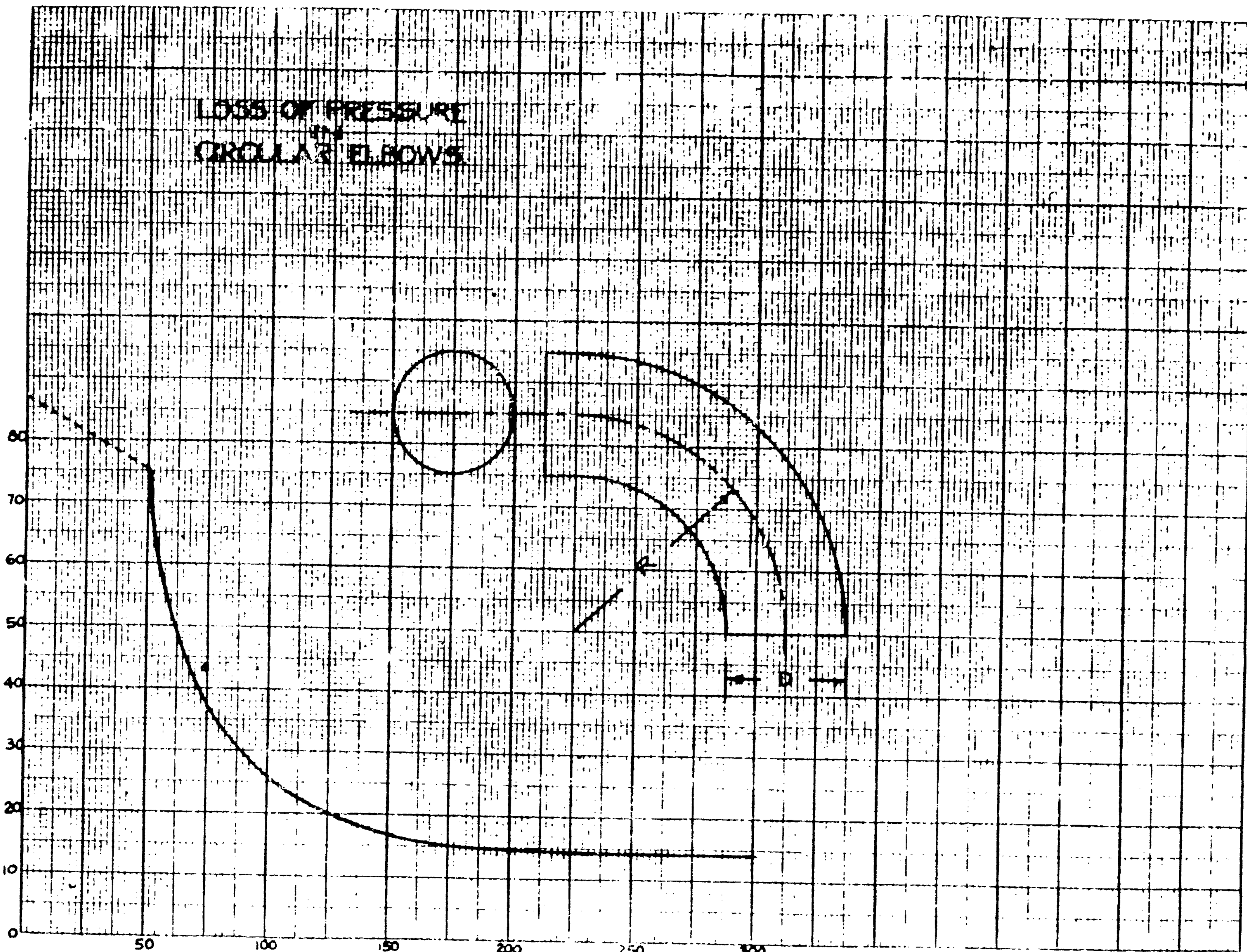
FIGURE 2

DDS53-1



Cubic Feet per Minute.
Friction Loss per 100 ft. - Inches of Water

Figure 3



CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER

FIGURE 4

DDS53-1

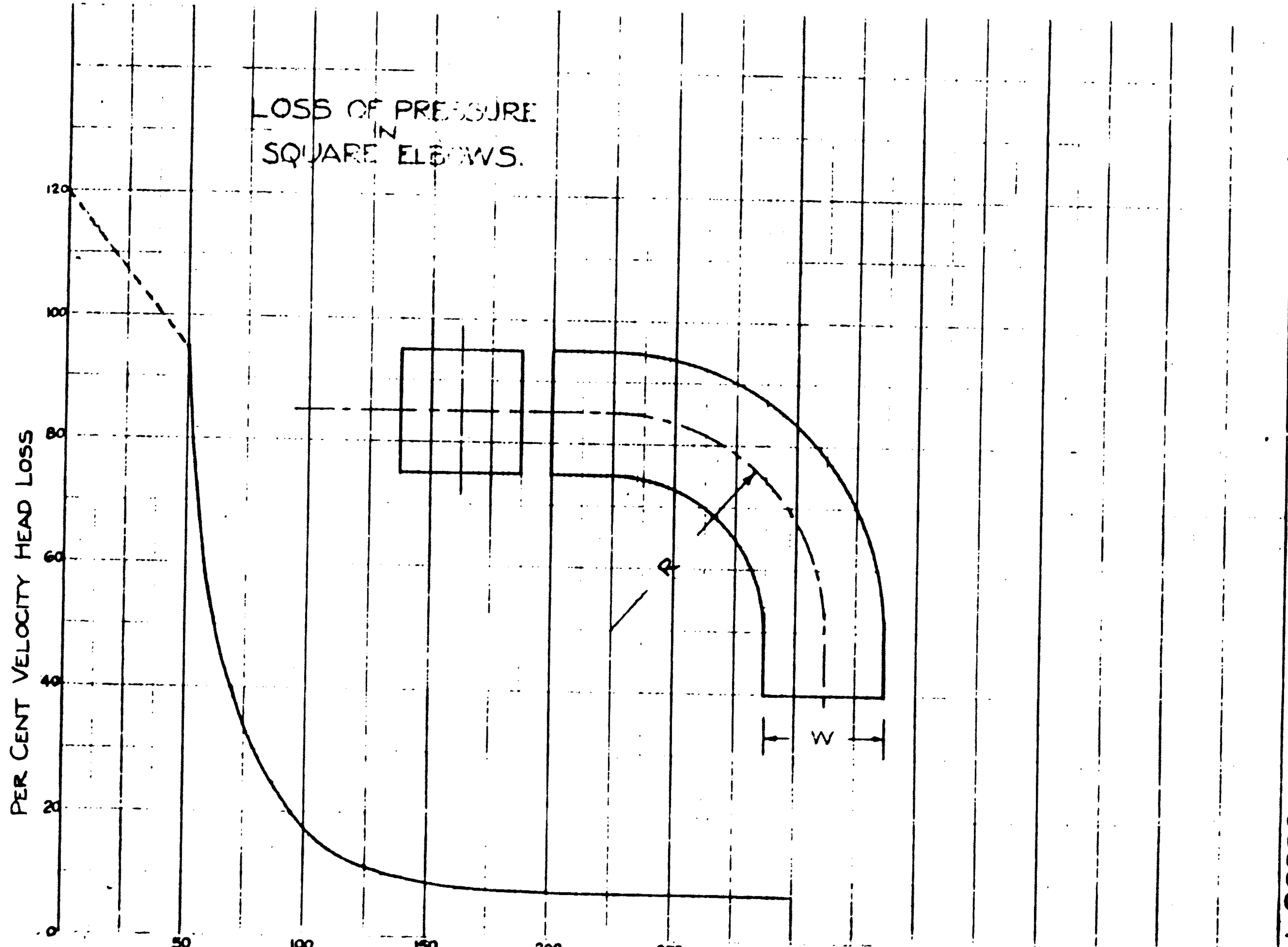


FIGURE 5

DD553-1

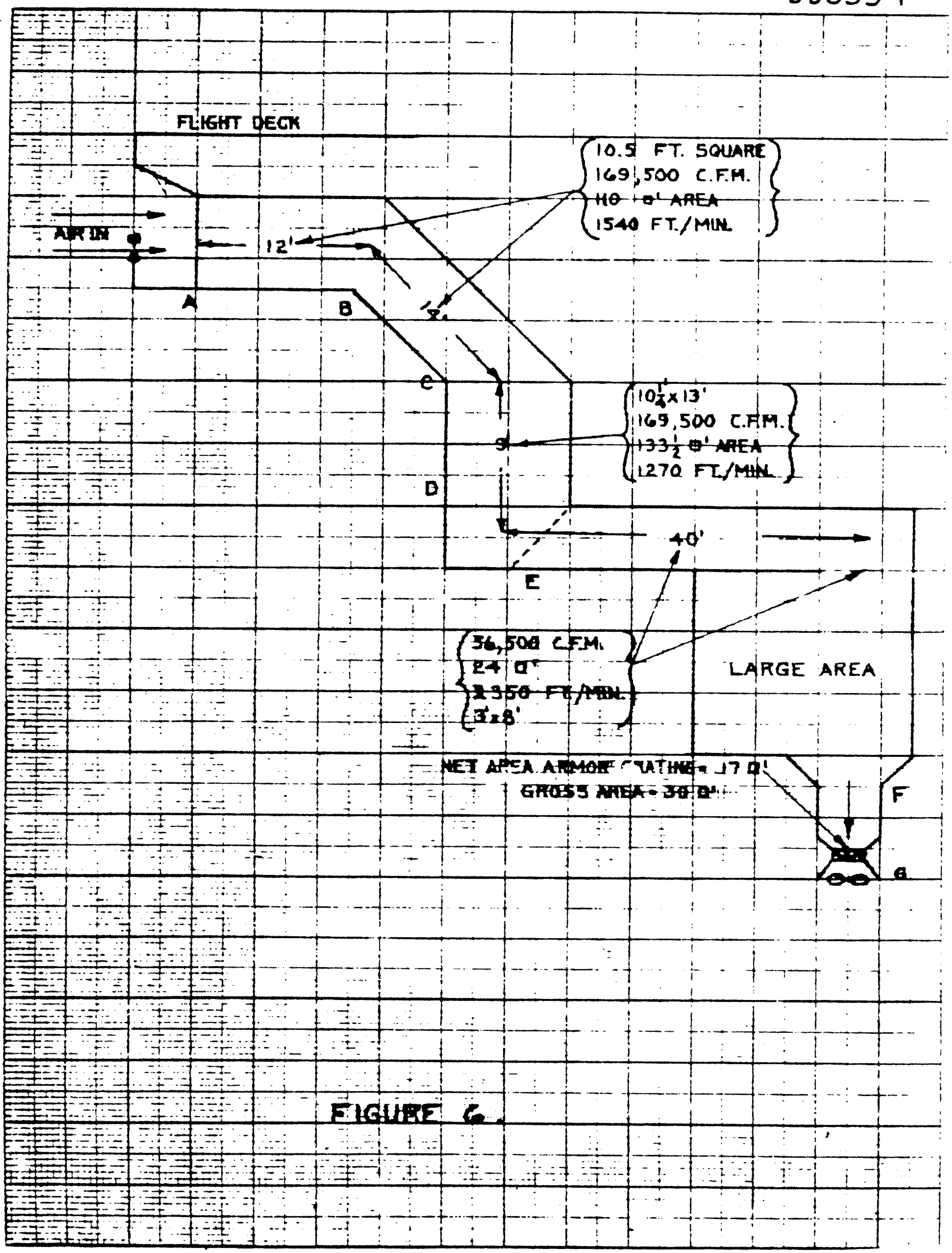

FLIGHT DECK
AIR IN
12'
10.5 FT. SQUARE
169,500 C.F.M.
110 □' AREA
1540 FT./MIN.
A
B
C
D
E
F
G
10½ x 13'
169,500 C.F.M.
133½ □' AREA
1270 FT./MIN.
40'
36,508 C.F.M.
24 □'
1350 FT./MIN.
3'x8'
LARGE AREA
NET AREA ARMOR GRATING = 17 □'
GROSS AREA = 38 □'

FIGURE 6.