MACHINERY'S REFERENCE SERIES

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NUMBER 39

FANS, VENTILATION AND HEATING

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CHAPTER I

CENTRIFUGAL AND DISK FANS

There are two types of fans in common use, known as the centrifugal fan or blower, and the disk fan or propeller. The former consists of a number of straight or slightly curved blades extending radially from an axis as shown in Fig. 1. When the fan is in motion the air in contact with the blades is thrown outward by the action of centrifugal force and delivered at the outer circumference or periphery of the wheel. A partial vacuum is thus produced at the center of the wheel, and air from the outside flows in to take the place of that which has been discharged. Fig. 3 illustrates the action of a centrifugal fan, the arrows showing the path of the air. This type of fan is usually enclosed in a steel plate casing of such form as to provide

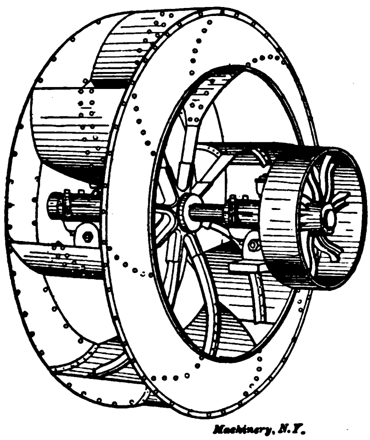


Fig. 1. Centrifugal Fan

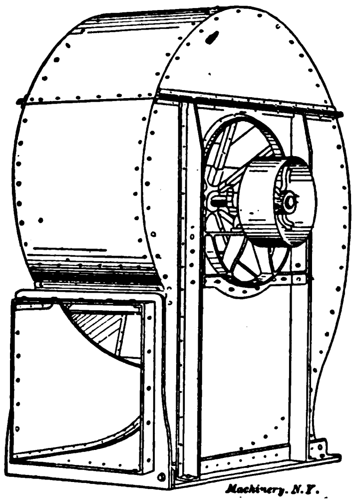


Fig. 2. Centrifugal Fan and Casing

for the free movement of the air as it escapes from the periphery of the wheel. An opening in the circumference of the casing serves as an outlet into the distributing ducts which carry the air to the various rooms to be ventilated, or to the furnaces in the case of mechanical draft

A fan with casing is shown in Fig. 2. The discharge opening can be placed in any position desired, either up, down, top horizontal, bottom horizontal, or at any angle. Where the height of the fan room is limited, a form called the three-quarter housing may be used, in which the lower part of the casing is replaced by a brick pit below the floor level (see Fig. 4). Another form of the centrifugal fan is shown in Fig. 6. This is known as the cone fan and is commonly placed in an opening in a brick wall and discharges air from its entire periphery into a room called a plenum chamber with which the various distributing ducts connect. This fan is often made double by placing two wheels back to back and surrounding them with a steel casing in a similar manner to the one shown in Fig. 2.

Cone fans are very efficient and are capable of moving large quantities of air at moderate speeds. Fig. 5 shows a form of small direct-connected exhauster ccmmcnly used for ventilating toilet rooms, chemical hoods, etc., and for furnishing a forced draft for forges and small boilers. Centrifugal fans are used almost exclusively for sup-

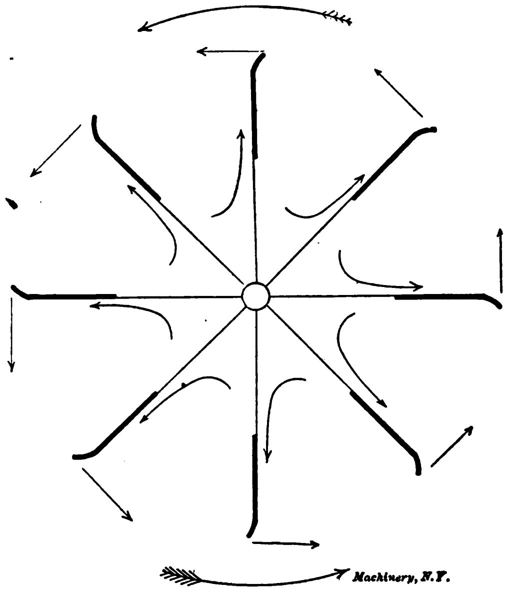


Fig. 3 Direction of Flow in Centrifugal Fan

plying air for the ventilation of buildings, for forced blast heating and for mechanical draft. They are also used as exhausters for removing the air from buildings when the resistance is considerable, and the quantity of air to be handled is large.

The disk fan is similar in construction to the propeller of a vessel and moves the air in lines parallel to its axis. This fan is made in various forms with both flat and curved blades. Fig. 7 shows one of the various designs arranged either for belted or direct-connected motor. This type of fan is light in construction, requires but little power at low speeds and is easily erected. It is especially adapted to exhaust ventilation when the resistance is small, being conveniently placed in the attic or upper part of a building and driven by an electric motor. Disk fans are largely used for the ventilation of public toilet rooms, smoking rooms, restaurants, etc., and are often connected with the main vent flues of large buildings, such as schools, halls, churches, theaters, etc. They are especially adapted for use in connection with gravity heating systems where the flow of air through vent flues is apt to be sluggish in mild weather.

Theory of Centrifugal Fans

The action of a fan is affected to such an extent by the various conditions under which it operates that it is impossible to give fixed rules for determining the exact results to be expected in any particular instance. This being the case, it seems best to take up the subject briefly from a theoretical standpoint, and then show what corrections are necessary in the case of a given fan under actual working conditions. As already stated, the rotation of a fan of this type sets in

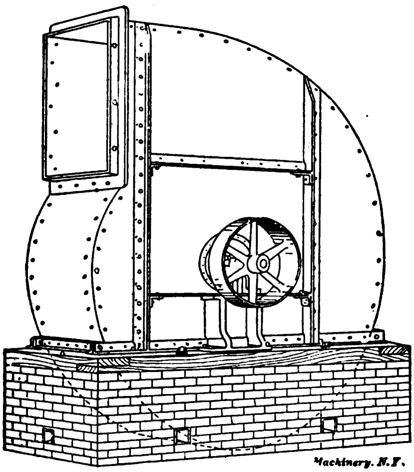


Fig. 4. Fan Partly Encased by Brick Wall

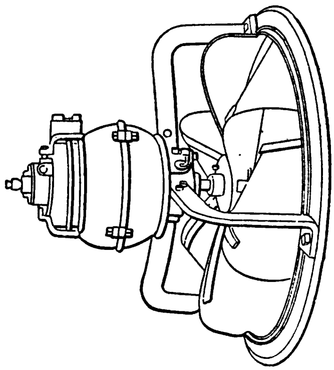


Fig. 5. Ventilator Wheel

motion the air between the blades, which by the action of centrifugal force is delivered at the periphery of the wheel into the casing surrounding it. As the velocity of flow through the discharge outlet depends upon the pressure or head within the casing, and this in turn upon the velocity of the blades, it becomes necessary to examine briefly into the relations existing between these quantities.

If a vessel as shown in Fig. 8 be filled with water, a certain pressure will be exerted upon the bottom, depending upon the depth and temperature of the water. If the weight of a cubic inch of water at a temperature of 50 degrees is 0.036 of a pound (called its *density* at that temperature) and the depth of the water in the vessel is 20 inches, then the pressure upon each square inch of the bottom will be the weight of a column of water having a sectional area of one square inch and a height of 20 inches, which in the above case is 20 ×

0.036 = 0.72 of a pound; so that for general use we may write $p = h \times d$, or $h = \frac{p}{d}$, in which

h = the height of the column of water, called the head,
d = the density of the water,
p = the pressure produced.

If h is taken in inches and d in ounces per cubic inch, then p will be in ounces per square inch. If h is in feet and d in pounds per cubic foot, p will be in pounds per square foot and so on, depending upon the units taken. When dealing with water pressure it is customary to take such units as will give p in pounds per square inch.

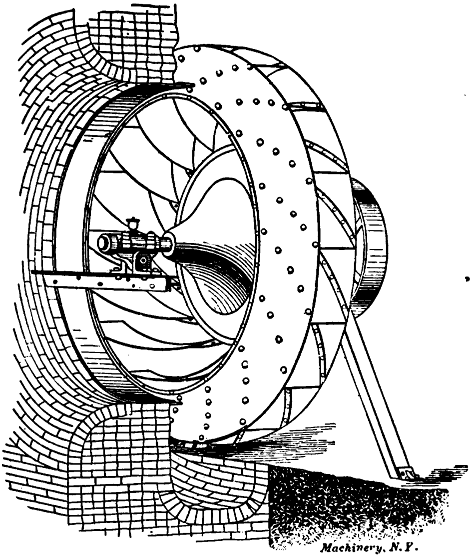


Fig. 6. Cone Fan Placed in Opening in Brick Wall

In the case of air pressure in connection with fans, ounces per square inch is the expression commonly used.

If a pipe be inserted in the side of the vessel (Fig. 9) at any given distance from the surface of the water, and a supply of water be provided sufficient to keep the level constant, a pressure of $h \times d$ will be exerted at the entrance to the pipe, causing the water to flow through it. The height h in this case is called the total head producing flow, and is divided into three parts, as follows: The *entry head,* that required to overcome the resistance to entry into the pipe; the *friction* or *pressure head,* that required to overcome the resistance due to the friction of the water in the pipe, and the *velocity head,* which is that used in giving motion or velocity to the water flowing through the pipe

The entry head depends upon the form of entrance to the pipe, and with smooth rounded edges is inappreciable. The friction head depends upon the length and size of the pipe, the interior surface, the number of bends and the quantity of water flowing through it. The velocity head is the same as the height through which a body must fall in a vacuum to acquire the velocity with which the water flows into the pipe. This is given by the formula for falling bodies.

$$h = \frac{v^2}{2g} \text{ in which}$$

h = the head in feet,
v = the velocity in feet per second,
g = the acceleration due to gravity, which is equal to 32.16.

This may also be written in the form of $v = \sqrt{2\,g\,h}$.

In applying this to the flow of air from a fan casing, it is customary to consider only the last two, and if the outlet is short and properly formed, the friction head may be neglected also. When the fan is

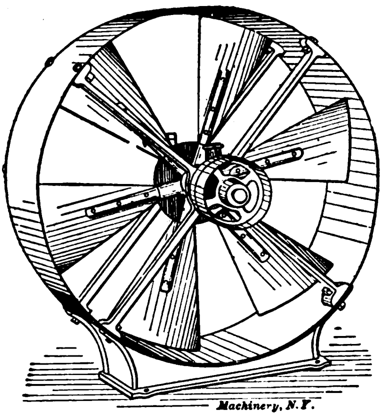


Fig. 7. Disk Fan

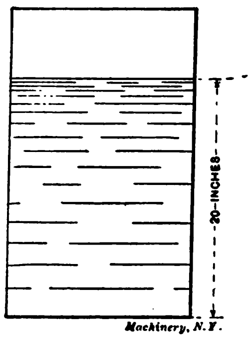


Fig. 8

to be used for moving air through ducts and flues in its practical application to ventilation, the effect of frictional resistance is added and must be provided for as stated later. We have seen that $v = \sqrt{2\,g\,h}$, and that $h = \frac{p}{d}$. Substituting $\frac{p}{d}$ for h in the above formula, we have $v = \sqrt{2\,g\,\frac{p}{d}}$ in which v is the velocity in feet per second of a liquid flowing from one chamber into another, where the difference in pressure is p and the density is d.

Applying this to the case of dry air at a temperature of 50 deg. F, and allowing for the change in density as it passes from a higher to a lower pressure, the formula becomes for the small differences in pressure employed in ventilating work,

$$v = \sqrt{\frac{1746659 \times p}{235 + p}}$$

in which v is in feet per second, and p in ounces per square inch.

Table I, computed from the above formula, is taken from "Mechanical Draft," published by the B. F. Sturtevant Co., and gives the velocity of dry air at a temperature of 50 deg. F. flowing into the atmosphere under different initial pressures.

A simple approximate formula giving very nearly the same results for air at 50 deg. is $v = 65.5\sqrt{h}$, in which v is the velocity in feet per

TABLE I. VELOCITY OF DRY AIR AT 50 DEGREES F. TEMPERATURE

Pressure in Ounces per square inch.	Feet per Second.	Feet per Minute.
¼	43 08	2585.0
⅜	52.75	3165.1
½	60.90	3653.8
⅝	68.07	4084.0
¾	74.54	4472.6
⅞	80.50	4829.7
1	86.03	5161.7
1¼	96.13	5768.0

second as before, and h the pressure expressed in inches of water as indicated by the balanced height of a column of that liquid in a water gage. Pressure in inches of water column may be reduced to ounces per square inch by multiplying by 0.58.

Example.—A pressure of ¼ of an ounce will produce a velocity

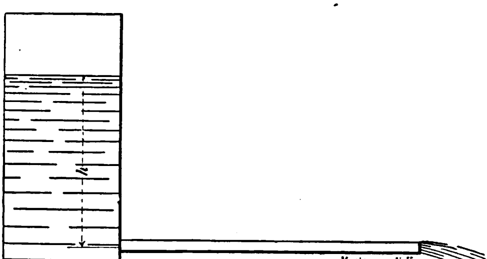


Fig. 9

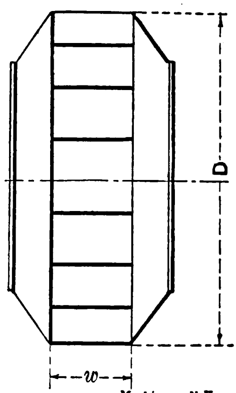


Fig. 10

of 43.08 feet per second, and a pressure of 1 ounce will produce a velocity of 86.03 feet per second, and so on.

The pressure within a fan casing is caused by the air being thrown from the tips of the blades, and varies with the velocity of rotation, that is, the higher the speed of the fan the greater will be the pressure produced.

When the various dimensions of a fan and casing are properly proportioned, the velocity of air-flow through the outlet will be the same as that of the tips of the blades, and the pressure within the casing will be that corresponding to this velocity. From this, it is evident that by knowing the diameter and speed of any given fan we can determine the peripheral velocity and find at once from Table I the pressure produced within the casing.

Blast Area

When the outlet from a fan casing is small, the air will pass out with a velocity equal to that of the tips of the blades, and the pressure within the casing will be that corresponding to the tip velocity. Now if the opening be slowly increased, while the speed of the fan remains constant, the air will continue to flow with the same velocity until a certain size is reached. The pressure in the casing will now begin to drop and the velocity of outflow become less than the tip

TABLE II. REVOLUTIONS PER MINUTE PRODUCING A GIVEN PRESSURE

Diam. of Fan Wheel in feet.	Pressure in Ounces per Square Inch.								
	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/4	1 1/2
2′	411	504	582	650	712	769	822	918	1005
2 1/2′	329	403	465	520	570	615	657	734	804
3′	274	336	388	433	475	513	548	612	670
3 1/2′	235	288	332	372	407	439	469	525	574
4′	206	252	291	325	356	384	411	459	502
4 1/2′	183	224	258	289	316	342	365	408	447
5′	164	202	232	260	285	308	329	367	402
6′	137	168	194	217	238	256	274	306	335
7′	117	144	166	186	203	220	235	262	287
8′	103	126	146	163	178	192	205	230	251
9′	92	112	129	144	158	171	183	204	223
10′	82	101	116	130	142	154	164	184	201

or peripheral velocity. The effective area of outlet at the point when this change begins to take place is called the *capacity area* or *blast area* of the fan. This varies somewhat with different types and makes of fans, but for the common form of blower it is approximately one-third of the projected area of the fan opening at the periphery, that is $\frac{D\,w}{3}$, in which D is the diameter of the fan wheel and w its width at the circumference (see Fig. 10).

Table II gives the speed of fans of different diameters necessary to maintain various pressures over an effective area equal to, or less than the blast area of the fan. The speeds given are in revolutions per minute, and are taken from "Mechanical Draft."

As a matter of fact the outlet of a fan casing is always made larger than the blast area, so that in actual practice the figures given in the following table must be corrected by certain factors as explained later.

Theoretical Capacity

If we assume the effective outlet area of a fan to be equal to the blast area, its capacity at any given speed can be computed as shown in the following example. A fan 5 feet in diameter has a width of 2 feet at the tips of the blades. What quantity of air will it discharge at a speed of 200 revolutions per minute?

Taking the blast area as $\frac{D\,w}{3}$, we find it to be $\frac{5 \times 2}{3} = 3.33$ square feet.

At a speed of 200 revolutions the tip velocity of the fan will be $3.1416 \times 5 \times 200 = 3{,}141$ feet per minute.

Therefore the air delivered by the fan is $3.33 \times 3{,}141 = 10{,}459$ cubic feet per minute.

What will be the capacity of the same fan working under a pressure

TABLE III. VOLUME OF AIR DISCHARGED AND HORSE-POWER REQUIRED

Pressure in Ounces per Square Inch	Cubic feet of Dry Air at 50 degrees Temperature which will be Discharged through an Orifice having an Effective Area of 1 square inch.	Horse Power required to move the Given Volume of Air under the Given Conditions.
$\frac{1}{4}$	17.95	.00122
$\frac{3}{8}$	21.98	.00225
$\frac{1}{2}$	25.37	.00346
$\frac{5}{8}$	28.36	.00483
$\frac{3}{4}$	31.06	.00635
$\frac{7}{8}$	33.54	.00800
1	35.85	.00978
$1\frac{1}{4}$	40.06	.01366

of ½ ounce, and what will be the required speed? Looking at Table I we find the velocity corresponding to ½ ounce pressure to be 3,653.8 feet per minute; therefore, the capacity of the fan is $3.33 \times 3{,}653.8 = 12{,}167$ cubic feet per minute.

The required speed may be taken directly from Table II, where it is found to be 232 revolutions per minute.

Power Required to Move Air

The work done by a fan in moving air is represented by the pressure exerted multiplied by the distance through which it acts. This is expressed in foot-pounds by the equation $W = PAV$, in which

W = work done, in foot-pounds per minute,
P = pressure at discharge opening, in pounds per square foot,
A = area in square feet, over which pressure P is exerted,
V = the velocity of flow, through discharge outlet, in feet per minute.

The horse-power required for moving air through any given area of discharge is given by the formula

$$\text{H. P.} = \frac{d\,a\,v^3}{5{,}100{,}480}$$ in which

d = density of the air at the given temperature,
a = effective area of discharge outlet, in square inches,
v = velocity of flow, in feet per second.

Table III (from "Mechanical Draft") gives the volume of air in cubic feet, which will be discharged per minute through an effective area of 1 square inch under different pressures; also the H. P. required for moving these quantities of air under the different conditions. This table gives only the power necessary for moving the air, and does not take into consideration the friction of the air in passing through the fan nor that of the fan itself. The additional power required to offset these losses will be taken up later.

Example: The effective area of a fan outlet is 480 square inches, and the pressure within the casing is ½ ounce per square inch. What volume of air will be discharged per minute, and what H. P. will be required, neglecting friction?.

From Table III we find that for ½ ounce pressure, 25.37 cubic feet of air will be discharged per minute through an area of 1 square inch, at an expenditure of 0.00346 H. P. Therefore, the total quantity discharged will be $480 \times 25.37 = 12{,}177$ cubic feet, requiring $480 \times 0.00346 = 1.66$ H. P.

Relation Between Volume, Pressure and Power

It can be shown mathematically that the following relations are true in the case of an ideal fan, and tests have shown them to be approximately correct for fans in actual operation. (1) The volume of air delivered varies directly as the speed of the fan, that is, doubling the number of revolutions doubles the volume of air delivered. (2) The pressure varies as the square of the speed; for example, if the speed is doubled, the pressure is increased $2 \times 2 = 4$ times. (3) The power required to run a fan varies as the cube of the speed; that is, if the speed is doubled the power required is increased $2 \times 2 \times 2 = 8$ times.

The value of a knowledge of these relations may be illustrated by the following example. Suppose for any reason it was desired to double the volume of air delivered by a certain fan. At first thought we might decide to use the same fan and run it twice as fast; but when we come to consider that the power would have to be increased eight times, it is probable that it would be much cheaper in the end to use a larger fan and run it at a lower speed.

Effect of Temperature

All computations and tables given thus far have been based on a temperature of 50 degrees F.

Raising the temperature of air causes it to expand and therefore reduces its density, or weight per unit of volume. This fact is of much importance where fans are used for induced draft, as the temperature of the gases commonly ranges from 300 degrees to 600 degrees. Table IV, also from "Mechanical Draft," shows the effect on the speed and power of a fan when the temperature of the air is increased. In the following example it is assumed for simplicity that the effective outlet area is equal to the blast area in each of the fans considered.

Example: From Table II we see that a 4-foot fan running at a speed of 411 revolutions per minute, will produce a pressure of 1 ounce, and looking in Table III we find that it will discharge 35.85 cubic feet of air per minute at a temperature of 50 degrees through an effective area of 1 square inch, with an expenditure of 0.00978 H. P. If the width of the fan is 18 inches at the tips of the blades, the blast area may be taken as $\frac{48 \times 18}{3} = 288$ square inches, from which the delivery will be $35.85 \times 288 = 10{,}324$ cubic feet per minute, requiring $0.00978 \times 288 = 2.8$ H.P.

Let us now assume the air to be heated to a temperature of 500 de-

TABLE IV. EFFECT OF TEMPERATURE ON SPEED AND POWER

Temperature in Degrees F.	Volume for same Weight.	Relative Velocity due to the same Pressure.	Speed of Fan to Handle same Weight.	Speed to Produce same Pressure.	Power for Speed Required to Handle same Weight.	Power Required to Handle same Weight at same Pressure with a Properly Proportioned Fan.
1	2	3	4	5	6	7
50	1.00	1.00	1.00	1.00	1.00	1.00
100	1.10	1.05	1.10	1.05	1.21	1.10
150	1.20	1.09	1.20	1.09	1.43	1.20
200	1.29	1.14	1.29	1.14	1.67	1.29
250	1.39	1.18	1.39	1.18	1.93	1.39
300	1.49	1.22	1.49	1.22	2.22	1.49
350	1.59	1.26	1.59	1.26	2.51	1.59
400	1.68	1.30	1.68	1.30	2.84	1.68
450	1.78	1.34	1.78	1.34	3.18	1.78
500	1.88	1.37	1.88	1.37	3.56	1.88
550	1.98	1.41	1.98	1.41	3.92	1.98
600	2.08	1.43	2.08	1.44	4.32	2.08

grees and see what conditions are necessary to handle the same weight per minute. Looking in Table IV for a temperature of 500 degrees we find that the volume becomes $10{,}324 \times 1.88 = 19{,}409$ cubic feet, and the speed of fan necessary to move this quantity is $411 \times 1.88 = 772$ revolutions per minute, requiring an expenditure of $2.8 \times 3.56 = 9.97$ H.P.

Suppose the above fan to be used for supplying a forced draft of 1 ounce to a battery of boilers, and it is desired to change to induced draft where the gases are to pass through the fan at a temperature of 500 degrees, what size and speed of fan will be required to produce the same intensity of draft (suction in this case) and what horsepower will be required to run it?

It is evident that the weight of air required will be the same in each case. This, for forced draft, we found to be 10,324 cubic feet per minute at a temperature of 50 degrees, and Table I shows that a peripheral velocity of 5,161.7 feet per minute was required to produce the pressure of 1 ounce at this temperature.

Referring to Table IV, columns 2 and 5, we find that after raising the air to a temperature of 500 degrees a volume of 10,324 × 1.88 = 19,409 cubic feet per minute is to be handled by the fan, requiring a peripheral velocity of 5,161.7 × 1.37 = 7,071.5 feet per minute to maintain the same pressure. As the velocity of flow through the discharge outlet is practically the same as that of the fan tips, the required blast area of fan will be 19,409 ÷ 7,071.5 = 2.74 square feet = 394 square inches. Assuming a fan 60 inches in diameter, we have 394 = $\frac{60}{3}$ — 19.7, or in round numbers, 20 inches as the required width at the periphery. This gives a fan of very nearly the same proportions as the 4-foot fan first used. The circumference of a 6-foot fan is 18.8 feet, therefore, 7,071.5, the required peripheral velocity, divided by 18.8 = 376 revolutions per minute, the required speed of the fan. The

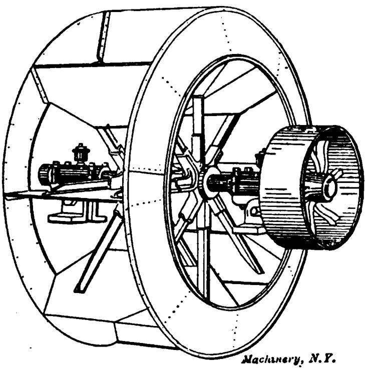


Fig. 11. Construction of Fan Wheel

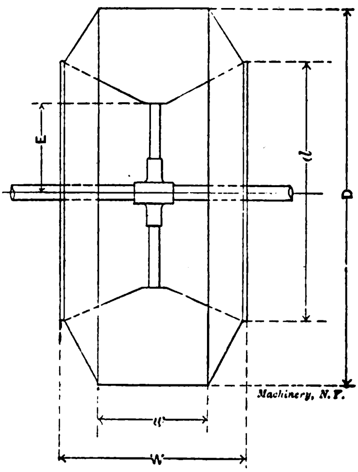


Fig. 12. Diagram of Fan Wheel

H. P. required by the 4-foot fan in handling the same weight of air at 50 degrees was 2.8. Referring to Table IV, column 7, we find that the power required to deliver the same weight of air at the same pressure at a temperature of 500 degrees is 1.88 times as great, or 2.8 × 1.88 = 5.2 H. P.

Having taken up the centrifugal fan from a theoretical standpoint and noted its action under ideal conditions, we will now consider it when working under the requirements of actual practice, and show what corrections must be made to the various rules and formulas previously given.

General Proportions

The general form of a fan wheel is shown in Fig. 11, which represents a double spider wheel with straight blades. Those over 4 feet in diameter usually have two spiders, while fans of large size are often provided with three or more. The number of blades or floats commonly varies from six to twelve, depending upon the size of fan. They are made both curved and straight; the former, it is claimed, run more quietly, but if curved too much will not work so well against a high pressure as the latter form. Fig. 12 represents a section

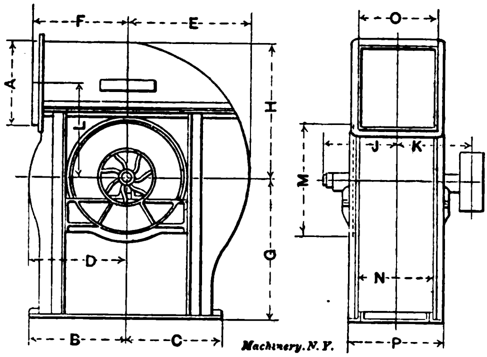


Diagram of Full Housing, for Use with Table V

through a fan wheel and shows the principal dimensions to be considered.

The following proportions are averages taken from fans of different sizes as made by several manufacturers for general ventilating and similar work and will be found to vary slightly from the proportions given by any one maker.

The diameter of the inlet (d) usually varies from 0.66 to 0.7 of the

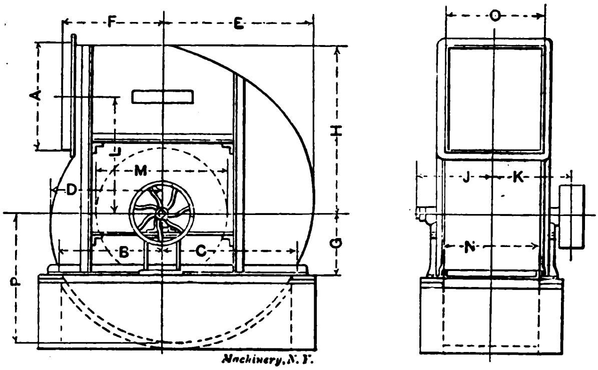


Diagram of Three-quarter Housing, for Use with Table VI

diameter of the wheel (D); 0.68 has been used in the following tables as a fair average. The distance from the center to the heel of the blades (E) is generally made 0.25 of the diameter of the wheel.

The width W varies somewhat in fans designed for different purposes. In the makes examined it averaged from 0.50 to 0.54 of the diameter, and 0.52 has been used in the tables following.

TABLE V. DIMENSIONS OF FULL HOUSING FANS.

Size of Wheel.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P
2½	18¾	20½	23½	21	25	24	28⅛	27	16¾	17⅛	17⅝	21	15	18¾	19¾
3	22¾	22	26	24	30	26¼	33¾	33	20¼	20⅜	21⅝	27½	18	22¾	22½
3½	25¾	24½	29½	27¼	34¼	28¾	39¼	37¾	22½	23⅜	24⅞	31⅞	21	25¾	27
4	28⅝	31	31	31	39	30½	41¼	43	22⅞	23½	29	34	24	29⅝	30¼
4½	31¼	34½	34½	34	44	34¼	41¼	49	24¾	25¾	34	38⅞	28	29½	34¼
5	33½	35½	35½	37	49	36⅛	47	55	27	28	39	42⅜	32	33½	38¼
5½	37¼	40½	40½	40½	53½	40½	51	60	29	30¼	42	47	36	37¼	44
6	42½	46	46	44	58	41½	55	65	33	34½	44	52¼	42	42½	51
7	48½	49	49	50	68	45½	64	77	36¼	38½	53	60½	48	48½	57
8	54½	57	63	57	77	49½	72	87	38	39¾	60	68½	48	48½	59
9	60¾	63	78	65	87	57½	80	98	41	42¾	68	77	54	54¾	65
10	72¾	66	84	72	96	68½	88	108	44⅜	47	72	85½	60	60¾	71

TABLE VI. DIMENSIONS OF THREE-QUARTER HOUSING FANS.

Size of Wheel.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P
4	28⅝	29½	36	31	39	30½	15	43	22⅞	23½	29	34	24	24⅝	35
4½	31¼	32	40	34	44	34¼	18	49	24⅞	25¾	34	38⅞	28	29½	39
5	33½	34½	44	37	49	36⅛	20	55	26¾	27¾	39	42⅜	32	33½	43
5½	37¼	38¼	48¾	40½	53½	40½	20	60	29	30¼	42	47	36	37¼	47
6	42½	40¾	52	44	58	41½	24	65	33	34½	44	52¼	42	42½	51
7	48½	47	62	50	68	45½	26¼	77	36¼	38½	53	60½	48	48½	59
8	54½	52½	69	57	77	49½	32¼	87	38	39¾	60	68½	48	48½	67
9	60¾	60	78	65	87	57½	36	98	41	42¾	68	77	54	54¾	76
10	72¾	66	85	72	96	68½	40	108	44⅜	47	72	85½	60	60¾	84

The width at the periphery should be, theoretically, such that the area of outlet around the entire wheel will be the same as the sum of the openings between the blades at the inlet, but in actual practice it is made somewhat greater, averaging from 0.7 to 0.8 of the width of the wheel. For convenience the relations between the different parts of the wheel may be expressed by the following equations:

$d = 0.68\,D,$ $W = 0.52\,D,$

$E = 0.25\,D,$ $w = 0.8\,W,$

in which

D = diameter of wheel,

d = diameter of inlet to wheel,

E = distance from center of fan to heel of blades,

W = width of fan at inlet,

w = width of fan at periphery.

These proportions, as already stated, do not represent those of any particular make, nor follow any fixed rule, but are general averages

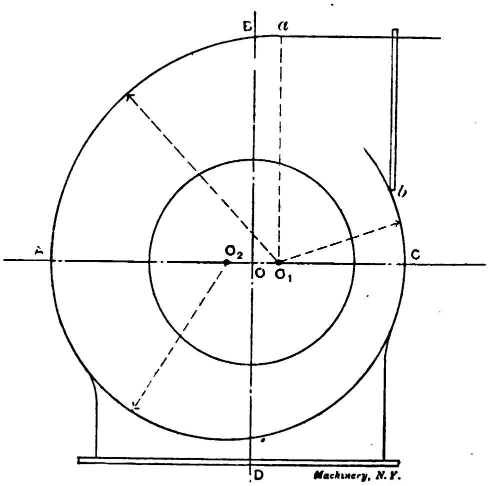


Fig. 13. Layout of Scroll Casing

as found from the catalogue dimensions of several well-known manufacturers.

Fans are made both with double and single inlets; the former being called "blowers" and the latter "exhausters."

The dimensions of the casing or housing vary somewhat with different makers and can be obtained from their catalogues. The dimensions given in Tables V and VI are taken from the catalogue of the B. F. Sturtevant Co. and will be found useful in the design of ventilating systems in approximating the space required. In case any particular make of fan is to be used, the exact dimensions should be obtained from the manufacturers, especially if the available space is limited.

The scroll of the casing is usually made up of the arcs of three circles of different radii, and the following method will be found useful in laying out a fan casing to scale. (See Fig. 13.)

First draw the center lines and lay off the distances *OA*, *OB*, *OC* and *OD* from the catalogue dimensions of the fan to be used. Then with a radius *R*, equal to *OB*, and a center at O_1 draw the arc *Aa*; with the same center and a radius R_1 equal to O_1C draw the arc *bC*; lay off $O\,O_2 = O\,O_1$ and with a radius R_2 equal to O_2A and a center at O_2 draw the arc *AC*. The other bounding lines can then be drawn in from dimensions taken from the catalogue table. The width of the casing is practically the width of the wheel with a small allowance for clearance.

Form of Orifice

The form of the opening through which the air passes when under pressure has a certain effect upon the quantity discharged, and makes it less than the theoretical amount which the size of opening and difference in pressures would indicate.

This reduction is due to two causes. First, to a contraction of the stream within, or just beyond the opening, depending upon its form; and second, to a certain amount of friction which tends to reduce the velocity somewhat.

The ratio between the actual quantity of air discharged and the theoretical quantity is called the "coefficient of discharge" and may be taken as about 0.8 for the short outlet from a fan casing, and as 0.56 for the inlet.

Example: The pressure within a fan casing is ½ ounce per square inch, and the area of the outlet is 4 square feet. What will be the actual discharge in cubic feet per minute?

From Table I the velocity corresponding to a pressure of ½ of an ounce is found to be 3,653.8 feet per minute, which gives a theoretical discharge of $3{,}653.8 \times 4 = 14{,}615.2$ cubic feet per minute. To find the actual quantity we must multiply this result by the coefficient of discharge, which gives us $14{,}615.2 \times 0.8 = 11{,}692.16$ cubic feet per minute.

The quantity of air discharged through any given opening, divided by the velocity of flow, is called the effective area. In the preceding example $11{,}692.16 \div 3{,}653.8 = 3.2$ square feet, which is the effective area of the outlet, while 4 square feet is the actual area.

Sometimes the conditions of the problem are such that it is stated as follows: The pressure within a fan casing is ½ ounce, and the outlet is 4 square feet, how much must the pressure be increased to make the actual discharge equivalent to the theoretical at the original pressure? It can be shown mathematically that the ratio of the theoretical pressure to the actual pressure required is $\frac{1}{K^2}$, in which *K* is the coefficient of discharge. Taking this as 0.8 for a fan outlet, the ratio becomes $\frac{1}{0.64} = 1.56$.

In the above example the theoretical discharge is 3,653.8 × 4 = 14,615.2 cubic feet per minute, in which it is assumed that 4 square feet is the effective area of the outlet. In order to make the actual discharge equal the theoretical, using the same sized outlet, it will be necessary to increase the pressure by 1.56, which gives us ½ × 1.56 = 0.78 of an ounce.

Blast Area

While the blast areas of fans of different diameters are slightly different for varying proportions, the formula $\frac{D w}{3}$ applied to standard fans will be found sufficiently accurate for ordinary use, in view of the approximations which must be made later in the assumption of pressures to be operated against, due to the friction of the air in ducts and flues.

Assuming $w = 0.8\ W$, and substituting for W its equivalent, $0.52\ D$, we have for the blast area $A = D \times 0.8 \times 0.52\,D \times 0.33 = 0.14\,D^2$.

TABLE VII

Dia. of Fan in Feet.	Blast Area, in sq. ft.
3	1.26
3½	1.72
4	2.24
4½	2.84
5	3.50
6	5.04
7	6.86
8	8.96
9	11.34
10	14.00

Table VII gives the blast areas for fans of different diameters, computed by the above method, which will be found to correspond very closely with those calculated by more complex methods for fans of approximately the same proportions.

Actual Capacity

In the examples given under *Theoretical Capacity*, it was assumed that the effective area of outlet was equal to the blast area, so that the velocity of outflow could be taken the same as the tip velocity. In actual practice the effective area of outlet is always made greater than the blast area and consequently the actual volume of air discharged is greater than the theoretical. On the other hand, the pressure drops below that due to the tip velocity and the velocity of flow through the outlet is correspondingly less.

The size of discharge outlet varies somewhat for different makes, but for a large number of fans examined it was found to average about 2.23 times the blast area as computed by the preceding method. Assuming a coefficient of discharge of 0.8, it gives as the effective area of discharge, 0.8 × 2.23 = 1.78 times the blast area.

A series of carefully conducted tests made some time ago upon an enclosed fan of practically the proportions taken, showed the pressure producing the flow of air through the outlet to be about 0.7 of that due to the peripheral velccity, when the effective area of outlet was made 1.78 times the blast area as computed above. Calculations based upon tests made by one of the leading manufacturers of fans of similar proportions give practically the same result.

We have seen that the velocity corresponding to any given pressure, or in other words, the peripheral velocity necessary to produce any desired pressure may be found by the formula:

$$v = 65.5 \sqrt{h}$$

for air at 50 degrees temperature, when the effective area of outlet is equal to or less than the blast area.

If increasing the effective outlet area to 1.78 times the blast area causes the pressure to drop to 0.7 that due to the peripheral velocity, we must, in order to again bring the pressure up to its original point, increase the velocity of the fan tips to a speed given by the equation

$$v = 65.5 \sqrt{\frac{h}{0.7}}$$

Assuming an original pressure (h) of 1, it is found that the tip velocity must be multiplied by 1.2 in order to produce this result.

This may be made clearer by an illustration: A 6-foot fan running

TABLE VIII. VOLUME OF AIR DISCHARGED PER MINUTE IN CUBIC FEET

Dia. of Fan.	¼ oz.	⅜ oz.	½ oz.	⅝ oz.	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	5 690	6,960	4,380	8,980	9,840	10,600	11,350	12,700
3½	7,750	9,490	10,950	12,250	13,400	14 500	15,500	17,300
4	10.350	12.650	14,600	16,350	17,900	19,300	20.650	23,100
4½	12,950	15,850	18,250	20,400	22,350	24.150	25,800	28,850
5	16,050	19,600	22,650	25,500	27,750	29,950	32,000	35,750
6	23,250	28,600	32,900	36,750	40,250	43.450	46,450	51,900
7	31,550	38,600	44.600	49,800	54,550	58,950	62,950	70,350
8	40.850	50.000	57,750	64,550	70,650	76.300	81,550	91,150
9	53,200	63,950	73 750	82.500	90,350	97,560	104 250	116,500
10	64,100	78,500	90,600	101,300	110,950	119.800	128,000	143,050

at a speed of 194 revolutions per minute produces a pressure of ½ ounce with a discharge outlet having an effective area equal to the blast area. If the effective discharge outlet is made 1.78 times the blast area, at what speed must the fan be run to maintain the same pressure, that is ½ ounce?

$$194 \times 1.2 = 233 \text{ revolutions per minute.}$$

Table VIII gives the cubic feet of air discharged per minute by fans of different diameters when run at such speeds as will produce the pressures indicated at the head of each column. These results were obtained by assuming the effective area of discharge outlet equal to 1.78 times the blast area, and multiplying this area in square inches by the quantities for the corresponding pressures as given in Table III. The results are given to the nearest ten, for quantities less than 10,000, and to the nearest fifty for those above 10,000.

Table IX gives the speeds of fans of different sizes necessary to maintain various pressures over effective discharge areas equal to 1.78 times the blast areas. These results are obtained by multiplying the speeds given in Table II by 1.2.

Horse-power Required

The power required for moving a given quantity of air under different conditions is given in Table III. This, however, does not include that necessary for overcoming the friction of the fan or the passage of the air through it.

The efficiency of a fan varies with the speed, the size of outlet, and the pressure against which it is working. Under favorable conditions, properly proportioned fans should have an efficiency of about 40 per cent, although they often fall considerably below this. The horsepower given in Table X for different sized fans is obtained by multiplying the effective area of outlet, in square inches (blast area × 1.78) by

TABLE IX. REVOLUTIONS PER MINUTE OF CENTRIFUGAL FANS

Dia. of Fan.	¼ oz.	⅜ oz.	½ oz.	⅝ oz	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	328	403	465	531	570	615	657	734
3½	282	345	398	446	488	526	562	630
4	247	302	349	390	427	460	493	550
4½	219	268	309	346	379	410	438	489
5	196	242	278	312	342	369	394	440
6	164	201	232	260	285	307	328	367
7	140	172	199	223	243	264	282	314
8	123	151	175	195	213	230	246	276
9	110	134	154	172	189	205	219	244
10	98	121	139	156	170	184	197	220

the quantities given in column three of Table III, and dividing the result by 0.4 (the efficiency).

Effect of Resistance

The effect of adding resistance to the flow of air from a fan by connecting it with a series of ventilating ducts or the furnace of a boiler is the same as would result from partially closing the discharge outlet.

Carefully conducted tests upon this type of fan have shown that the reduction of air flow is very nearly in proportion to the reduction of discharge area. That is, if the outlet of the fan is closed to one-half its original area the quantity of air discharged will be practically one-half that delivered by the fan with a free opening. Tests also show that the required horse-power varies approximately as the quantity of air discharged. The effect of attaching a fan to the ventilating flues of a building like a schoolhouse, church, or hall, where the ducts have easy bends and the velocity of air flow through them is not over 1,000 to 1,200 feet per minute, is about the same as closing the outlet from 10 to 20 per cent. For factories, with deep heaters and smaller ducts, where the velocity of air flow runs up to 1,500 or 1,800 feet per minute, the effect is equivalent to closing the outlet from 20 to 30 per cent, or even more in very large buildings.

For the average mechanical draft plant, the effect is about the same as for factory ventilation, that is, a reduction of area from 20 to 30 per cent.

In buildings similar to schoolhouses and churches, it has been found in practice that fans of the blower type, having curved floats, operate quietly and give good results when run at a speed corresponding to one-half ounce pressure at the discharge outlet; this gives a speed of about 3,600 feet per minute at the circumference of the wheel. Higher speeds are accompanied with greater expenditure of power and are likely to produce a roaring noise or cause vibration. A much lower speed does not provide sufficient pressure to give proper control of the air distribution during strong winds. For factories and similar buildings, a higher pressure of three-fourths ounce or more is generally employed. In the case of mechanical draft, a pressure of three-fourths to one ounce is usually required for forced draft, and from one-half to three-fourths ounce for induced draft.

Example: A schoolhouse requires an air supply of 52,000 cubic feet

TABLE X. HORSE-POWER REQUIRED FOR CENTRIFUGAL FANS

Dia. of Fan.	H. P. Required for Different Pressures.							
	¼ oz.	⅜ oz.	½ oz.	⅝ oz.	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	1.0	1.9	2.8	3.8	5.0	6.4	7.9	10.8
3½	1.5	2.5	3.8	5.4	7.0	8.8	10.8	15.0
4	1.8	3.3	5.0	6.9	9.1	11.5	14.0	19.6
4½	2.3	4.0	6.3	8.8	11.5	14.5	17.8	24.8
5	2.8	5.0	7.8	10.3	14.3	17.9	21.9	30.5
6	3.9	7.3	11.1	15.3	20.4	25.8	35.1	44.0
7	5.3	10.0	15.8	21.3	27.9	35.1	42.8	60.0
8	7.0	13.0	20.0	28.0	36.7	46.4	56.8	81.0
9	8.8	16.1	25.0	34.8	45.6	57.5	70.4	98.5
10	11.0	20.3	31.1	43.5	57.1	72.0	88.0	122.9

of air per minute. What size and speed of fan will be required, and what will be the necessary horse-power of engine, assuming the resistance to be equal to reducing the discharge outlet 20 per cent?

As the effect of the resistance is the same as reducing the discharge opening, and consequently the air quantity, 20 per cent, we must look in Table VIII for a fan capable of delivering $\frac{52,000}{0.8} = 65,000$ cubic feet of air per minute. In the column headed ⅝ ounce pressure we find that an 8-foot fan will deliver 64,550 cubic feet, and is the size we should use.

Table IX shows us that a fan of this size must be run at a speed of 195 revolutions per minute to produce a pressure of ⅝ ounce. In Table X we find that 28 horse-power is required to run the fan at a speed to produce this pressure when discharging into free air. Under the conditions of the problem, with the outlet restricted 20 per cent, the required power would be $28 \times 0.8 = 22.4$ H. P.

Actually the pressure is increased slightly by restricting the outlet at constant speed, but this is seldom taken into account in ventilating work as air volume and power required are the quantities sought.

Mechanical Draft

Theoretically, about 12 pounds of air are required to burn one pound of average coal or coke. In practice, however, with natural draft, it is necessary to supply about twice this amount, to secure complete combustion, owing to the difficulty of bringing the air into contact with the entire body of coal. With mechanical draft and deeper fires, this quantity may be reduced to about 18 pounds per pound of coal, which is equivalent in round numbers to 230 cubic feet at a temperature of 50 degrees F. Assuming a coal consumption of 5 pounds per horse-power per hour, calls for an air supply of $5 \times 230 = 1{,}150$, or practically 1,200 cubic feet of air per boiler horse-power per hour.

The size of fan and power required for supplying a given quantity of air at a stated pressure may be determined in the same manner for forced draft as illustrated in the preceding example, except that the resistance may be taken as equivalent to a reduction of about 25 per cent instead of 20 per cent in air volume and horse-power, at a given speed or pressure.

In the case of induced draft a correction must be made for the effect of higher temperature of gases passing through the fan. The proper size and speed of fan in this case, together with the power of engine, may be found by first computing them for an air temperature of 50 degrees and then making corrections as shown below.

Let us assume the average temperature of the gases from the furnace to be 550 degrees. Referring to Table IV, we find that the volume of a given weight of air at this temperature becomes 1.98 times that at 50 degrees (column 2), and that the peripheral speed of fan required to produce a given pressure is 1.41 times as great (column 5), also that the necessary power is increased 1.98 times when a properly proportioned fan is used (column 7). The horse-power can evidently be obtained at once by multiplying that already computed by 1.98.

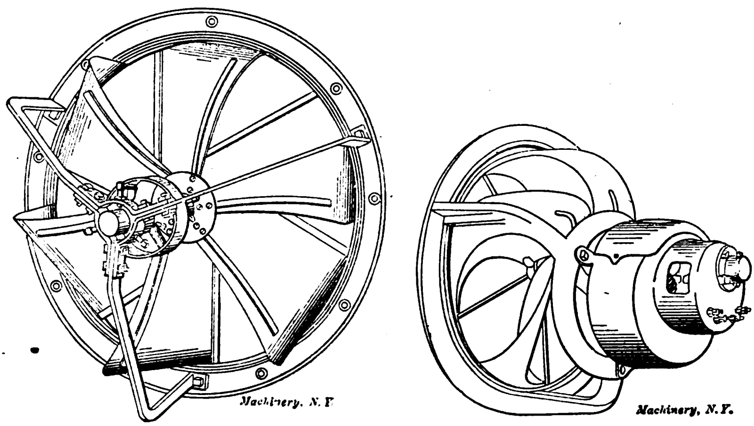
As we wish to keep the pressure the same, and to move the same weight of air, it will be necessary to increase the peripheral velocity of the fan by 1.41; the speed required for doing this can be fixed after determining the size of fan. From column 3 we find that the relative velocity of flow due to the same pressure at this higher temperature is 1.41. Therefore, if the pressure is to remain constant, the volume of air moved 1.98 times as great, and the velocity of flow through the outlet 1.41 times that of air at 50 degrees, the effective area of outlet must be $\frac{1.98}{1.41} = 1.4$ times that of the fan used for delivering the air at the lower temperature.

We may then select a fan having approximately this size of outlet and obtain the necessary speed in revolutions by dividing the required peripheral velocity by the circumference of the fan wheel.

For other temperatures, the method will be the same, although the ratios from Table IV will vary.

Example: Determine size and speed and power required for driving a fan used in producing an induced draft of ¾ ounce for a battery of boilers aggregating 670 horse-power. We find first 670 × 1,200 = 804,000 cubic feet of air per hour, or 804,000 ÷ 60 = 13,400 cubic feet per minute. Assuming the resistance as equivalent to restricting the area of outlet 0.25, we must look in Table VIII under ¾ ounce pressure for a fan delivering 13,400 ÷ 0.75 = 17,866 cubic feet, where we find that a 4-foot fan will deliver 17,900 cubic feet.

Table IX shows us that a speed of 427 revolutions per minute is required to produce this pressure, and Table X gives the necessary power as 9.1, which multiplied by 0.75 is 6.8, or in round numbers, 7 horse-power. Assuming the temperature of the gases to be 550 de-



Figs. 14 and 15. Types of Disk Fans

grees, and referring to Table IV, we find the following ratios or multipliers: Volume, 1.98; velocity due to same pressure, 1.4; peripheral speed to produce same pressure, 1.4; power, 1.98.

The effective area of outlet of a 4-foot fan is 2.24 × 1.78 = 4 square feet; therefore the required area is $\frac{1.98}{1.4} \times 4 = 5.6$ square feet. The fan having an outlet area corresponding most nearly to this is a 5-foot, whose outlet is 3.5 × 1.78 = 6.2 square feet. Although this is slightly larger than called for, we can depend upon the automatic regulator to keep the speed at the required point to supply the necessary air volume. The required peripheral speed is 427 × 12.6 (circumference of 4-foot wheel) × 1.4 = 7,532 feet per minute. The circumference of a 5-foot fan is 15.7 feet; therefore, 7,532 ÷ 15.7 = 480 revolutions per minute, which is the required speed of the larger fan. The final horse-power is 7 × 1.98 = 13.86, or 14 in round numbers.

Disk Fans

The capacity of disk fans varies greatly with the type and conditions under which they operate. The rated capacities given in catalogues are for fans revolving in free air; that is, mounted in an opening without being connected with ducts or working against a resistance. Disk fans of the type shown in Figs. 14 and 15, when working against a low resistance such as is commonly encountered in ventilating work where the air is drawn or forced through ducts of medium length at velocities not exceeding 600 or 800 per minute, propel the air in a direction parallel to the shaft, a distance equal to about 0.6 of the diameter for each revolution. From this we have the equation $C = 0.6\,D \times R \times A$, in which

C = cubic feet of air delivered per minute,
D = diameter of fan in feet,
R = revolutions per minute,
A = area of fan in square feet.

TABLE XI

Dia. of Fan, Inches	Cubic ft. per Revolution
12	0.5
18	1.7
24	4.0
30	7.8
36	13.0
42	21.0
48	32.0
60	62.0
72	108.0

In order to obtain the best results the linear velocity of air-flow through the fan should not average over 1,000 feet per minute, nor exceed 1,200 feet as a maximum.

Table XI gives the volume of air delivered per revolution for fans of different diameter based upon the above formula.

TABLE XII

Velocity through Fan in feet per minute	Horse-power required per 1,000 cubic feet of air removed
600	0.12
700	0.20
800	0.30
900	0.35
1,000	0.40
1,100	0.45
1,200	0.55

The power required per cubic foot of air delivered, depends upon the velocity of flow through the fan.

Table XII gives the horse-power required per thousand cubic feet of air for different velocities moved through the fan.

CHAPTER II.

HEATERS FOR HOT BLAST AND VENTILATION

The best type of heater for any particular case will depend upon the volume and final temperature of the air, the steam pressure and the available space. When the air is to be heated to a high temperature for both warming and ventilating a building as in the case of a shop or mill, or for drying purposes, heaters of the general form shown in Figs. 16, 18, and 19 are used. These may also be adapted to all classes of work by varying the proportions as required. They can be made shallow and of large superficial area for the comparatively

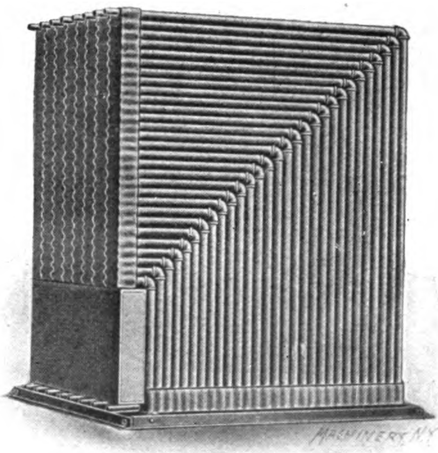


Fig. 16. Sturtevant Miter Type Heater

low temperatures used in purely ventilating work, or deeper, with less height and breadth, as higher temperatures are required.

Description of Types of Heaters

Fig. 18 shows the general construction of the standard hot blast heater of the B. F. Sturtevant Company. This consists of several sectional cast iron bases with loops of wrought iron pipe connected as shown. The steam enters the upper part of the bases or headers and passes up one side of the loops, then across the top and down on the other side, where the condensation is taken off through the return drip, which is separated from the inlet by a partition. These heaters are made up in sections of 2 and 4 rows of pipes each, and can be made any depth desired by adding more sections. The height varies from 3½ to 9 feet and the width from 3 feet to 7 feet in the standard sizes. They are usually made up of one-inch pipe, although 1¼ inch is commonly used in the larger sizes.

For convenience in estimating the approximate dimensions of a heater, Table XIII is given. The standard heaters made by different manufacturers vary somewhat, but the dimensions given in the table represent average practice. Column 3 gives the square feet of heating surface in a single row of pipes of the dimensions given in columns 1 and 2, and column 4 gives the free area between the pipes.

In calculating the total height of the heater add 1 foot for the base. These sections are made up of 1-inch pipe except the last, or 7-foot sections, which are made of 1¼-inch pipe.

Fig. 16 shows the miter type of the Sturtevant heaters, with single-chambered inlet and outlet sections. This arrangement provides abso-

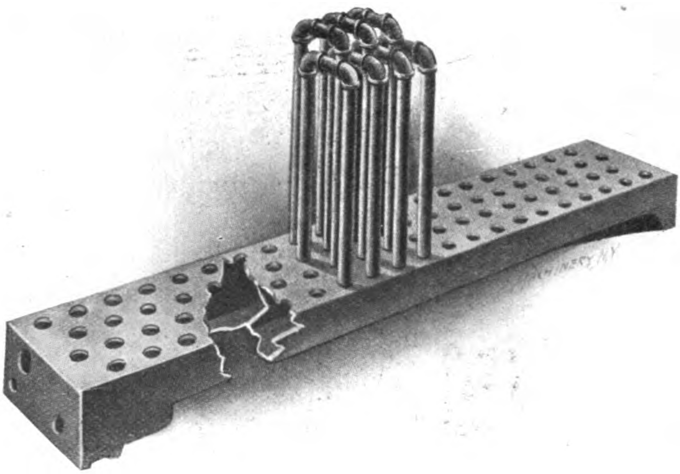


Fig. 17. American Blower Co.'s Hot Blast Heater, Four-pipe Section

lute freedom of expansion and perfect circulation. Steam is admitted at the top of the inlet section, and the drips removed from the end of the outlet section. Heaters of this type are usually enclosed in a steel casing as shown in Fig. 29, although brick walls are often used for heaters of large size.

Fig. 17 illustrates the construction of a 4-pipe section of the heater made by the American Blower Company, and Fig. 19 the same heater complete, without its steel plate casing. This heater is similar in appearance to the one just described, but differs somewhat in its construction. The base is divided lengthwise by an inside partition, so that the two pipes or legs of each loop connect with different chambers, one of which connects with the steam supply and the other with the return.

Fig. 20 shows a special form of heater particularly adapted to ventilating work where the air does not have to be raised above 75 or 80 degrees. It is made up of 1-inch wrought iron pipe connected with supply and return headers; each section contains 14 pipes, that is, 2 pipes wide and 7 pipes deep, and they are usually made up in groups of 5 sections each. These coils are supported upon T-irons resting upon a brick foundation. Heaters of this form are usually made to

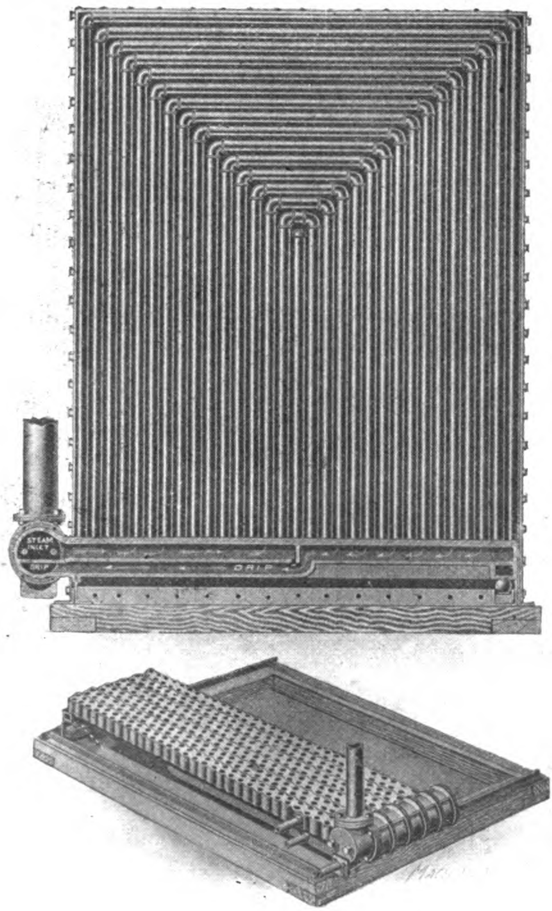


Fig. 18. Sturtevant Hot Blast Heater

extend across the side of a room with brick walls at the sides instead of being encased in steel housings.

Figs. 21, 22, and 24 show the "Vento" cast iron hot blast heaters made by the American Radiator Company. This type of heater is to be used under the same conditions as the pipe heaters already described. Fig. 21 shows a group of sections and illustrates the general construction and method of connection. Fig. 22 shows the sections arranged in a stack, five rows deep; and Fig. 24 the same stack with its steel casing and the supply and return connections.

Cast iron indirect radiators of the pin pattern shown in Fig. 23 are well adapted for use in connection with mechanical ventilation, and also for heating where the air volume is large and the temperature not too high, as in churches and halls. They make a convenient form of heater for schoolhouse and similar work, for being shallow, they can be supported upon I-beams at such an elevation that the con-

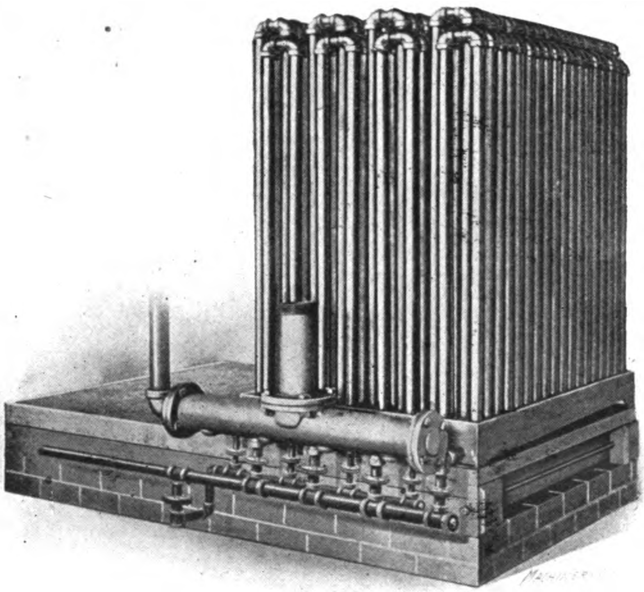


Fig. 19. American Blower Company's Heater Complete without Casing

densation may be returned to the boilers by gravity. In the case of vertical pipe heaters the bases are below the water-line of the boilers, and the condensation must be returned by the use of traps and pumps.

Efficiency of Pipe Heaters and Calculation of Sizes Required

The efficiency of the heaters used in connection with forced blast varies greatly, depending upon the temperature of the entering air, its velocity between the pipes, the temperature to which it is raised, and the steam pressure carried in the heater. The general method in which the heater is made up is also an important factor.

In designing a heater of this kind, care must be taken that the free area between the pipes is not contracted to such an extent that an excessive velocity will be required to pass the given quantity of air through it. In ordinary work it is customary to assume a velocity of 800 to 1,000 feet per minute; higher velocities call for a greater pressure on the fan which is not desirable in ventilating work.

TABLE XIII

Width of Section	Height of Pipes Ft.	In.	Heating Surface, Square Feet	Free Area through Heater, Square Feet
3	3	6	20	4.2
3	4	0	22	4.8
3	4	6	25	5.4
3	5	0	28	6.0
4	4	6	34	7.2
4	5	0	38	8.0
4	5	6	42	8.8
4	6	0	45	9.6
5	5	6	52	11.0
5	6	0	57	12.0
5	6	6	62	13.0
5	7	0	67	14.0
6	6	6	75	15.6
6	7	0	81	16.8
6	7	6	87	18.0
6	8	0	92	19.2
7	7	6	98	21.0
7	8	0	103	22.4
7	8	6	109	23.8
7	9	0	116	25.2

In the heaters shown, about 0.4 of the total area is free for the passage of air; that is, a heater 5 feet wide and 6 feet high would

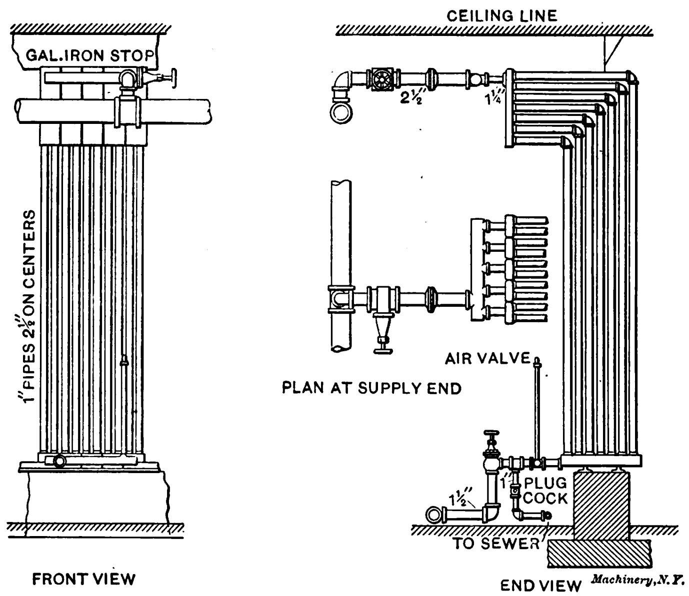


Fig 20 Special Heater used in Ventilating Work

have a total area of $5 \times 6 = 30$ square feet, and a free area between the pipes of $30 \times 0.4 = 12$ square feet. The depth or number of rows of pipe does not affect the free area, although the friction is increased and additional work is thrown upon the fan. The efficiency in any given heater will be increased by increasing the velocity of the air through it, but the final temperature will be diminished, that is, a larger quantity of air will be heated to a lower temperature in the second case, and while the total heat given off is greater, the air quantity increases more rapidly than the heat quantity, which causes a drop in temperature.

Increasing the number of rows of pipe in a heater with a constant

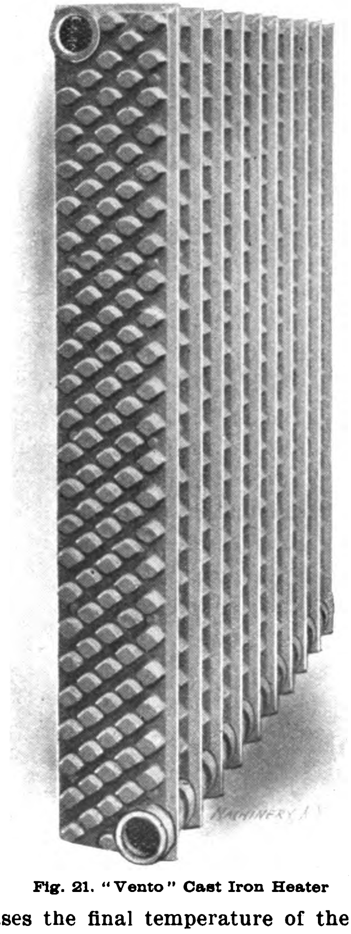


Fig. 21. "Vento" Cast Iron Heater

air quantity increases the final temperature of the air but diminishes the efficiency of the heater, because the average difference in temperature between air and steam is less. Increasing the steam pressure in the heater (and consequently its temperature) increases both the final temperature of the air and the efficiency of the heater. Table XIV has been prepared from different tests and may be used as a guide in computing probable results under ordinary working conditions. In this table it is assumed that the air enters the heater at a temperature of zero and passes between the pipes with a velocity of 800 feet per minute. Column 1 gives the number of rows of pipe in the heater and columns 2, 3, and 4 the final temperature of the air for different steam pressures. Columns 5, 6, and 7 give approximately the corresponding efficiency of the heater.

Example: Air passing through a heater 10 pipes deep and carrying 20 pounds pressure will be raised to a temperature of 90 degrees and

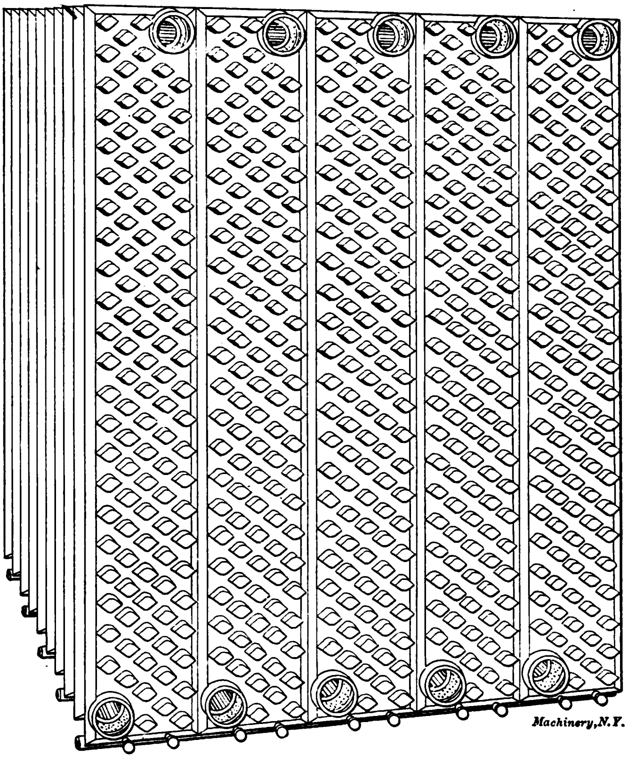


Fig. 22. "Vento" Cast Iron Heater

the heater will have an efficiency of 1,650 B. T. U. per square foot of surface per hour.

For a velocity of 1,000 feet, multiply the *temperatures* given in the table by 0.9 and the *efficiencies* by 1.1.

Example: How many square feet of radiation will be required to raise 600,000 cubic feet of air per hour from zero to 80 degrees, with a velocity through the heater of 800 feet per minute and a steam pressure of 5 pounds? What must be the total area of the heater front and how many rows of pipes must it have?

The B. T. U. required is found by multiplying the volume of air by the desired rise in temperature and dividing the result by 55; hence $600{,}000 \times 80 \div 55 = 872{,}727$ B. T. U. are required.

Referring to Table XIV we find that for the above conditions a heater 10 pipes deep is required, and that an efficiency of 1,500 B. T. U. will be obtained. Then 872,727 ÷ 1,500 = 582 square feet of surface is required, which may be taken as 600 in round numbers. 600,000 ÷ 60 = 10,000 cubic feet of air per minute, and 10,000 ÷ 800 = 12.5 square feet of free area required through the heater. If we assume

TABLE XIV

Temperature of entering air, zero.
Velocity of air between the pipes, 800 feet per minute.

Rows of Pipe Deep	Temp. to which the Air will be raised from zero. Steam Pressure in Heater. 5 lbs.	20 lbs.	60 lbs.	Efficiency of the Heating Surface in B. T. U. per sq. ft. per hour. Steam Pressure in Heater. 5 lbs.	20 lbs.	60 lbs.
4	30	35	45	1600	1800	2000
6	50	55	65	1600	1800	2000
8	65	70	85	1500	1650	1850
10	80	90	105	1500	1650	1850
12	95	105	125	1500	1650	1850
14	105	120	140	1400	1500	1700
16	120	130	150	1400	1500	1700
18	130	140	160	1300	1400	1600
20	140	150	170	1300	1400	1600

0.4 of the total heater front to be free for the passage of air, then 12.5 ÷ 0.4 = 31.25 square feet, total area required.

The general method of computing the size of heater for any given

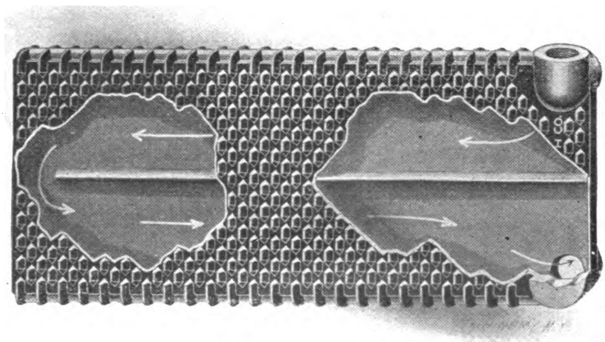


Fig. 23. Pin Type Heater

building which is to be both ventilated and warmed by a hot-blast system, is the same as in the case of indirect heating. First obtain the B. T. U. required for ventilation, and to that add the heat loss through walls, etc., and divide the result by the efficiency of the heater under the given conditions.

Example: An audience hall is to be provided with 400,000 cubic feet of air per hour. The heat loss through walls, etc., is 250,000 B. T.

U. per hour in zero weather. What will be the size of heater, and how many rows of pipe deep must it be, with 20 pounds steam pressure?

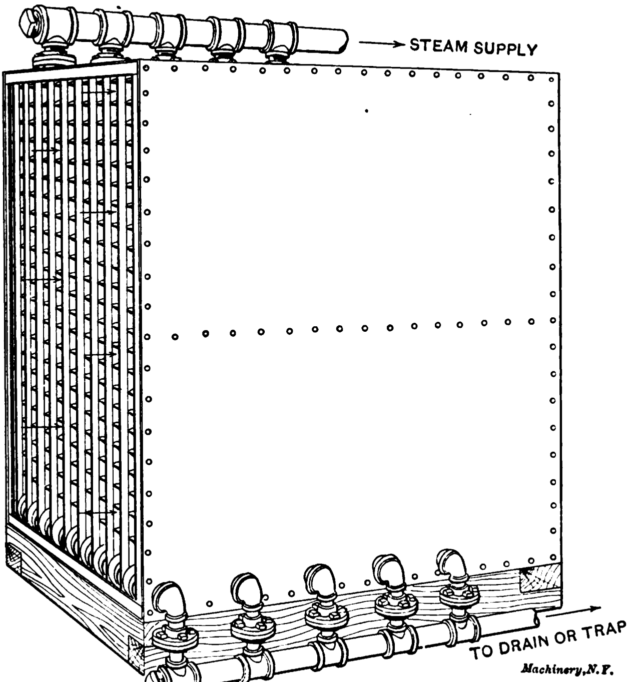


Fig. 24. "Vento" Cast Iron Heater

$400{,}000 \times 70 \div 55 = 509{,}090$ B. T. U. for ventilation. Therefore $250{,}000 + 509{,}090 = 759{,}090$ B. T. U., total to be supplied.

We must next find to what temperature the entering air must be

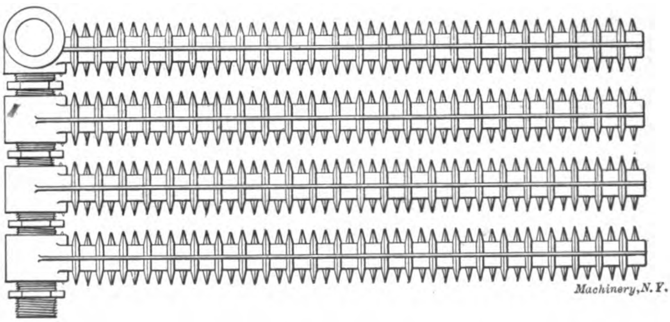


Fig. 25. Diagram of Pin Type Heater

raised in order to bring in the required amount of heat, so that the number of rows of pipe in the heater may be obtained and its corresponding efficiency determined. We have entering the room for purposes of ventilation, 400,000 cubic feet of air every hour at a temperature of 70 degrees, and the problem now becomes, to what temperature must this air be raised to carry in 250,000 B. T. U. additional for warming?

We know that 1 B. T. U. will raise 55 cubic feet of air 1 degree.

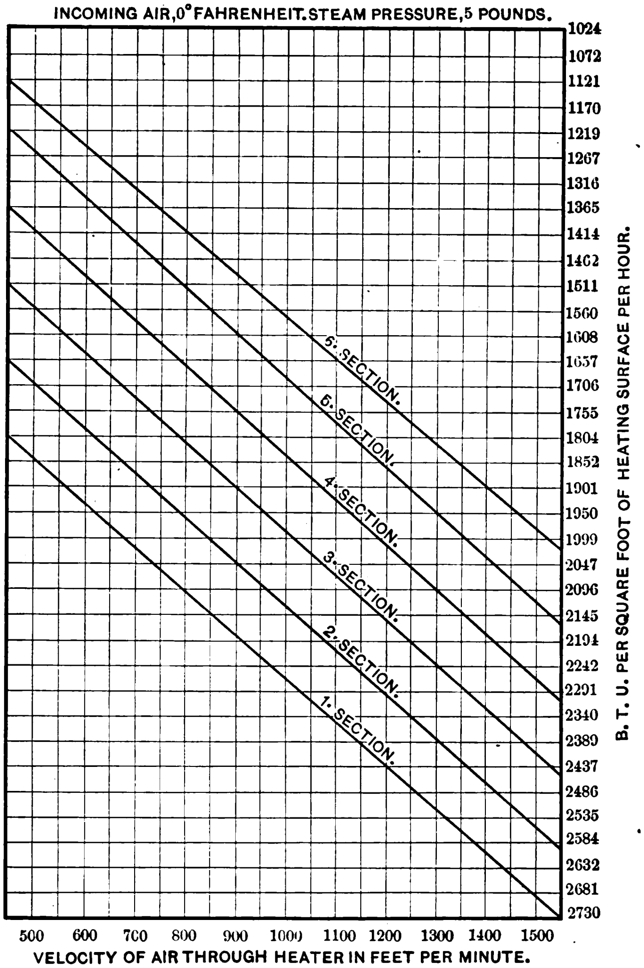


Fig. 26. Condensation Chart

Then 250,000 B. T. U. will raise 250,000 × 55 cubic feet or air 1 degree. Thus 250,000 × 55 ÷ 400,000 = 34 degrees, required excess temperature. The air in this case must then be raised to 70 + 34 = 104 degrees to provide for both ventilation and warming. Referring to Table XIV we find that a heater 12 pipes deep will be required, and that the corresponding efficiency of the heater will be 1,650 B. T. U. Then 759,090 ÷ 1,650 = 460 square feet of surface required.

Heating Surface Required for Factories

The proportional heating surface for factory heating is generally expressed in the number of cubic feet in the building for each linear foot of 1-inch steam pipe in the heater. On this basis, in factory practice, with all of the air taken from out of doors, there are generally allowed from 100 to 150 cubic feet of space per foot of pipe,

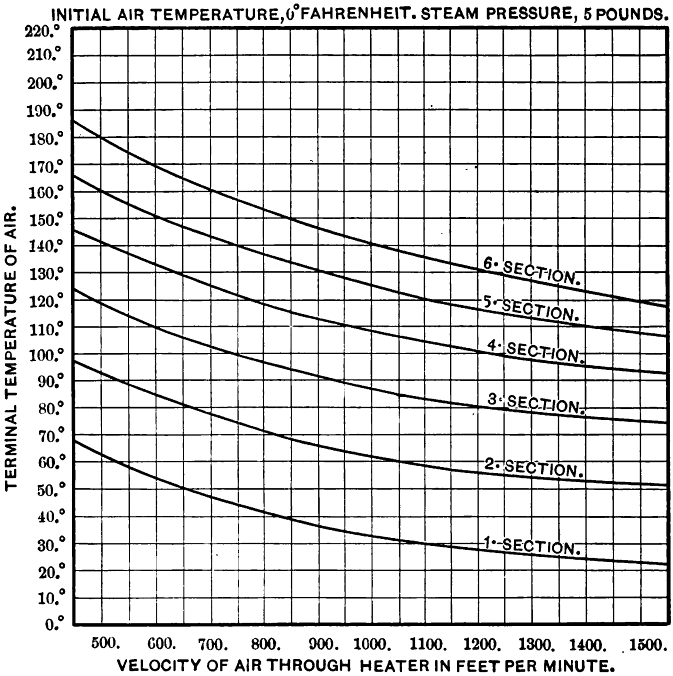


Fig. 27. Temperature Chart

according to whether exhaust or live steam is used, live steam here indicating steam of about 80 pounds pressure. If practically all of the air is returned from the buildings to the heater, these figures may be raised to about 140 as a minimum, and possibly 200 as a maximum, per foot of pipe.

Temperature and Condensation Charts

The accompanying "temperature" and "condensation" charts, Figs. 26 and 27, show the results obtained with the "Vento" cast iron heater, and the data given therein correspond to that found in Table XIV for pipe heaters. These charts explain themselves and require no further description.

Indirect Pin Radiators

Heaters made up of indirect pin radiators of the usual depth have an efficiency of at least 1,500 B. T. U. with steam at 5 pounds pressure, and are easily capable of warming air from zero to 80 degrees or over when computed on this basis. The free space between the sections bears such a relation to the heating surface that ample area is provided for the flow of air through the heater without producing an excessive velocity.

Pipe Connections

Hot blast heaters, commonly called main heaters, are usually divided into several sections, the number depending upon their size, and each provided with a separate valve in the supply and return. In

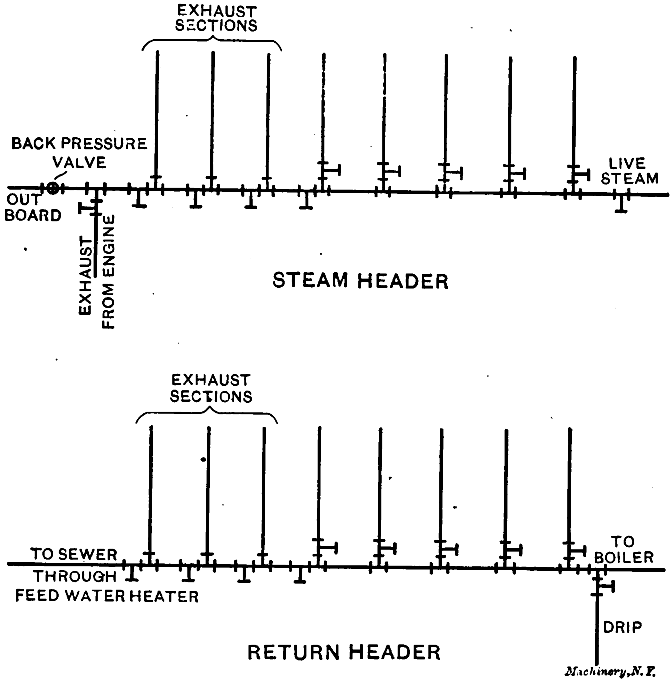


Fig. 28. Diagram of Heater Pipe Connections

making these divisions, special care should be taken to arrange for as many combinations as possible.

Example: A heater, 10 pipes deep, may be made up of three sections, one of 2 rows and two of 4 rows each. By means of this division, 2, 4, 6, 8 or 10 rows of pipe can be used at one time, as the outside weather conditions may require.

In making the pipe connections to a heater of this kind, a main or header is usually run along one side, from which branches of the proper size are carried to the different sections. The arrangement of the returns should correspond in a general way with the supplies. The main header should be properly drained, and the condensation from the heater tapped to a receiving tank, or returned to the boilers by gravity if the heater is overhead. If possible, the return from each section should be provided with a water-seal two or three feet in depth. This is because condensation is greater in the outer sections, resulting in a slight difference in pressure which causes the

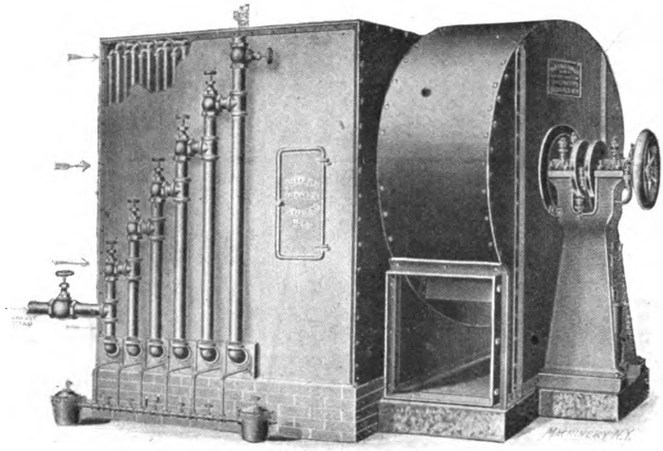


Fig. 29. Buffalo Forge Company's Heater

return water from the inner sections to be drawn into the outer ones, thus producing water-hammer and imperfect circulation of steam.

In the case of overhead heaters, the returns may be sealed by the water-line of the boiler or by the use of a special water-line trap, but vertical pipe heaters resting on foundations near the floor are usually provided with siphon loops, extending into a pit. If this arrangement is not convenient, a separate trap should be placed on the return from

TABLE XV

Square feet of Surface	Diameter of Steam Pipe	Diameter of Return
150	2″	1¼″
300	2½″	1½″
500	3″	2″
700	3½″	2″
1000	4″	2½″
2000	5″	2½″
3000	6″	3″

each section. The main return, in addition to its connection with the boilers or pump receiver, should have a connection with the sewer for blowing out when steam is first turned on. Sometimes each section is provided with a connection of this kind.

Large automatic air valves should be connected with each section, and it is well to supplement these with a hand pet-cock, unless individual blow-off valves are provided as described above. If the fan is driven by a steam engine, provision should be made for using the exhaust in the heater, and part of the sections should be so valved that they may be supplied with either exhaust or live steam as desired.

Fig. 28 shows in diagram a method of making the connections for a heater in which three of the sections may be used in this way. Another way of accomplishing the same result is shown in Fig. 29, which shows a heater made by the Buffalo Forge Company. In this arrangement all of the sections are interchangeable.

The sizes of the mains and branches are often fixed by the tapping of the heater sections. Table XV, based on experience, has been found to give satisfactory results where the apparatus is near the boilers.

From 50 to 60 square feet of radiating surface should be provided in the exhaust portion of the heater for each engine horse-power, and should be divided into at least three sections, so that it can be proportioned to the requirements of different outside temperatures.

The condensation from the exhaust sections contains oil from the engine and should not be returned to the boilers; much of its heat, however, can be saved by passing it through a feed water heater. A simple heater for this purpose may be made of a piece of 8-inch pipe, 7 or 8 feet in length, with flanged heads, and containing a coil made up of four lengths of 1-inch brass pipe. The feed to the boilers is made to pass through the coil, while the space around it is filled with hot condensation. A similar heater is sometimes placed in the exhaust pipe from the engine, for use when exhausting outboard in mild weather. After passing through the feed water heater the condensation should be trapped to the sewer.

CHAPTER III.

HEATING AND VENTILATING MACHINE SHOPS

Methods of Heating

The older method of heating a shop was by means of steam coils, either run along the walls under the windows, or supported overhead as most convenient. This arrangement necessitates a large amount of heating surface together with an extended system of supply and return piping, thus greatly increasing the liability to leaks and freezing. This method provides no fresh air for ventilation, and the distribution of heat is not of the best. When the coils are placed along the walls, under benches, it is uncomfortably warm for those working near them, and if supported overhead, the heat rises directly to the ceiling or roof, thus leaving the lower portion of the room too cold.

The most satisfactory arrangement is where the heating is done by hot air properly distributed through suitable ducts and flues. The heating surface in this case is very compact, only about one-fifth of that required for direct heating being necessary; and as the surface is grouped in a single heater, even in buildings of large size, long runs of piping are avoided. In the largest plants, or where the buildings are more or less detached, it becomes necessary to increase the number of units, but even then the pipe runs are simple compared with those necessary for direct heating. A better distribution of heat is obtained, resulting in a more uniform temperature throughout the rooms. As heating systems of this kind are usually arranged for taking a portion of their air supply from out of doors, it is possible to secure any degree of ventilation required.

General Arrangement

The location of the fan and heater and the general arrangement of the distributing ducts will depend largely upon the construction and plan of the building. One of the simplest arrangements for a building of small size is that shown in Fig. 31. In this case a single galvanized iron uptake is carried from the mouth of the fan directly upward through the different stories of the building. At each floor the requisite number of outlets are provided at or near the ceiling level, and the air discharged toward the outer walls. In the case of a larger building it would be necessary to extend the distributing ducts horizontally from the main uptake, as shown in Fig. 33.

Another typical arrangement is that shown in Fig. 30, which represents the plan and elevation of the heating and ventilating system installed in the shops of the Ashcroft Manufacturing Company, of Bridgeport, Conn. In this arrangement the fan and heater are centrally located in the basement near one of the side walls. Main distributing ducts are carried in both directions near the floor, and from these, vertical risers are taken off at frequent intervals and carried up to the different stories. The air is discharged into the rooms horizontally at an elevation of about eight feet from the floor. Regulating dampers are provided in each uptake for proportioning the air flow through each outlet. The fan is of the centrifugal type, with bottom horizontal discharge. The wheel is 6½ feet in diameter, and is driven by a 7-inch by 7-inch vertical direct-connected engine. The heater is made of 1-inch pipe, twenty-two rows deep. The sections are 6½ feet wide by 8 feet high. The building is warmed by air-rotation, no connection being made with the outside air.

Figs. 34 and 35 show plan and section elevation of a different arrangement, as installed in the shops of the Houston, Stanwood & Gamble Co., at Covington, Ky. In this case the fan and heater are placed in one corner of the building, and the air carried across one

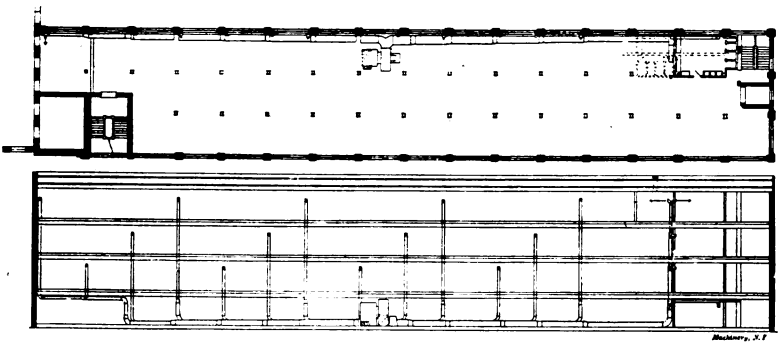


Fig. 30. Heating and Ventilating System of the Ashcroft Manufacturing Company, Bridgeport, Conn.

end in an underground concrete duct. Two uptakes connect with this, running up beside columns supporting the roof; horizontal branches are then carried the entire length of the building, one on each side, passing through the roof trusses as shown. The air is discharged through short mouthpieces set at an angle to give it a downward direction, and arranged to deliver it both to the central portion and the side aisles or bays. This particular arrangement of the distributing ducts is made necessary on account of the traveling cranes which pass through the entire length of the building in each of the three sections as indicated by tracks at the sides in Fig. 34. This is a typical illustration of an overhead distribution, the air being discharged at an elevation of about 18 feet above the floor. The apparatus in this case also takes its air supply from the building, leakage being depended upon for ventilation. The fan is of the three-quarter housed type, with a bottom horizontal discharge connecting with the underground duct. The wheel is 9 feet in diameter. The heater is made up of two groups, with a supply and return header at each side. It is 20 pipes deep, and has an exposed front 8 feet high by 12½ feet wide.

Fig. 36 illustrates a somewhat similar arrangement, although in this case two overhead units are used, each made up of two fans and a heater, and the air is carried downward to a point about 8 feet from the floor before being discharged. The system is installed in the machine and erecting shops of the Pennsylvania Railroad Company at Trenton, N. J. All parts of the system are symmetrically arranged.

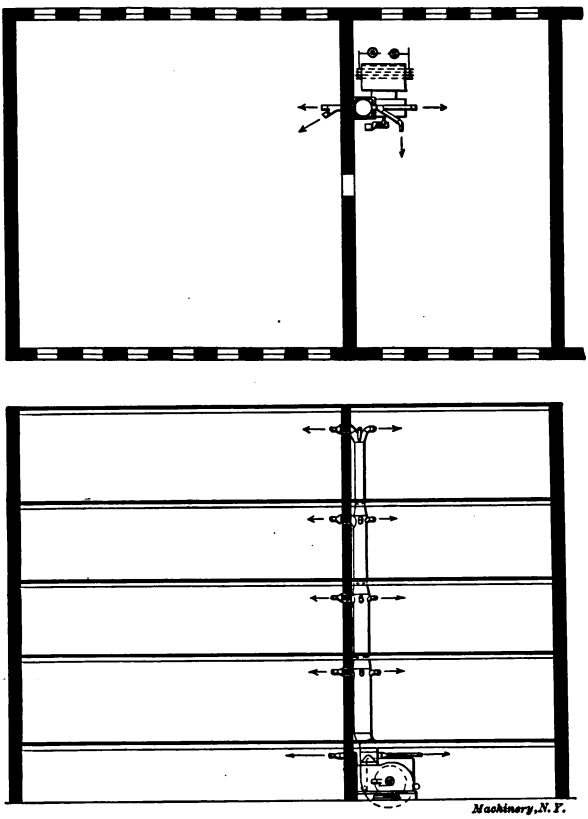


Fig. 31. Simple Arrangement of Heating Installation in a Small Building

which gives practically an equal resistance to the flow of air in each of the four main distributing ducts.

A plan and elevation of the fan and heater of one of the units is shown in Fig. 32. Two double inlet fans are used, with wheels 8½ feet in diameter attached to a common shaft and driven by a belted motor. The heater is made up with a double header, as in the previous layout. It is 20 pipes deep, 13½ feet wide and 10 feet high. The air is taken from the building, but is forced through the heater instead of being drawn through by suction, as in the other arrangements mentioned.

These five buildings illustrate the more common methods of arranging the distributing systems in the heating of machine shops of modern construction.

Material Used for Ducts and Flues

The airways are either constructed of brick or galvanized iron. In brick buildings where the heating system is planned before the building is constructed the flues may be most readily and economically built in the walls as the building is erected. When this is done, care should be taken to give them as smooth an interior as possible by removing all projecting motor from between the bricks.

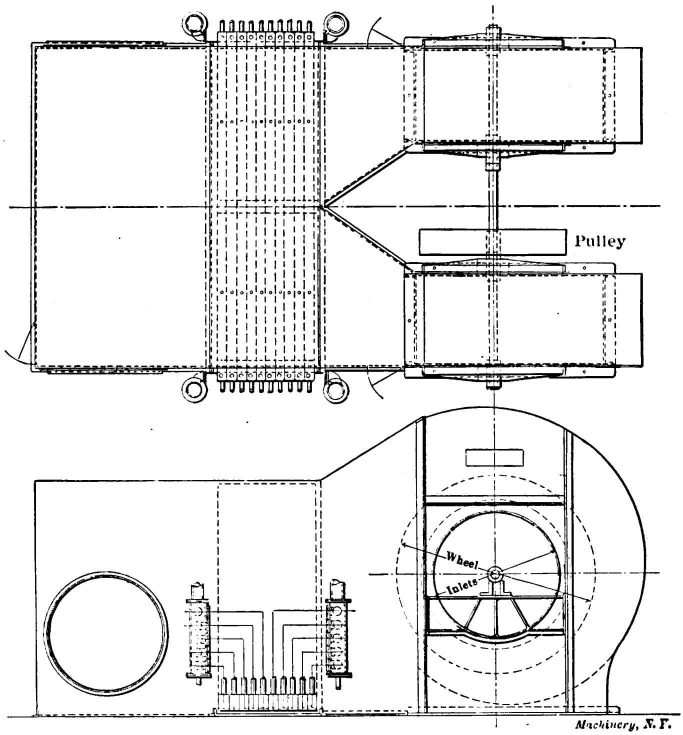


Fig. 32. Fan and Heater of Installation shown in Fig. 36

Underground ducts are built either of brick or concrete for the larger sizes, and generally of glazed tile for the branches. In buildings of wooden construction, and also those of brick when erected before the heating system is laid out, it is customary to use galvanized iron. This is easily worked into the required form, is light in weight, and takes up a minimum of space for a given area.

Construction of Ducts and Flues

Great care should be taken in the design and construction of a system of ducts and flues. When a change in the direction of flow is necessary, a gracious curve should be provided. For 90-degree turns, the elbow should be made with at least five pieces, and the radius of the inner side of the elbow should not be less than the diameter of the pipe. This relation between the radius of curvature and the size of pipe should hold in the case of rectangular ducts as well.

When a branch is taken off from a straight run of pipe, it should be given an angle of 45 degrees at the point of connection, and the remaining change in direction made by an easy turn. The main run of pipe is commonly reduced at each branch or take-off by an easy taper, about 28 inches in length, which can be made from a sheet of iron of standard width, which is 30 inches. Whenever the duct or pipe branches, the construction should be such as to divide the air volume into the required proportions, giving to each branch an easy

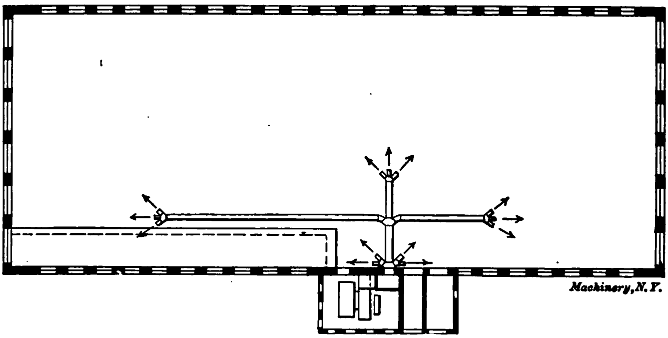


Fig. 33. Heating Installation in a Building of Larger Size than the one shown in Fig. 31

change in direction, when possible. While due regard should be given to the proper proportioning of the pipe areas, it is not possible to get a sufficiently accurate distribution of air without the use of dampers and deflectors. In the case of a large number of small outlets from a main duct, the best results are usually obtained by the use of adjustable dampers in each outlet. Where there are several branches of considerable size leading from the main, it is well to place adjustable deflectors at the junction of the ducts, so that the air volume can be deflected into the branches in such quantities as may be desired. In the case of brick or concrete underground ducts, the same points relating to curves, dampers, etc., should be observed as described above for galvanized iron.

Size of Ducts and Flues

The sectional area of ducts and flues is based upon the velocity of the air flow through them. It is a well-known fact that the frictional resistance to the flow of air through pipes increases as the square of the velocity, hence if the power required for driving the fan was an important factor, very low velocities would be required. As a matter of fact, this is generally neglected in practice, and velocities are based upon the most desirable speed of fan for this class of work, which is about 5,000 linear feet per minute tip velocity. This results in a velocity through the fan outlet of about 3,500 feet per minute. The size of the main duct is made such that the velocity commonly runs from 2,000 to 3,000 feet per minute, although some engineers make a practice of starting with a duct the same size as the fan outlet, but making a small increase in the size of the branches, so that their aggregate area shall be from 30 to 40 per cent greater than the area of fan outlet, thus bringing the velocity down to from 2,500 to 2,800 feet per minute.

It is frequently possible in shop practice to secure satisfactory circulation of the air with a limited extent of ducts by discharging it at a high velocity, as noted above, thus compelling it to continue its

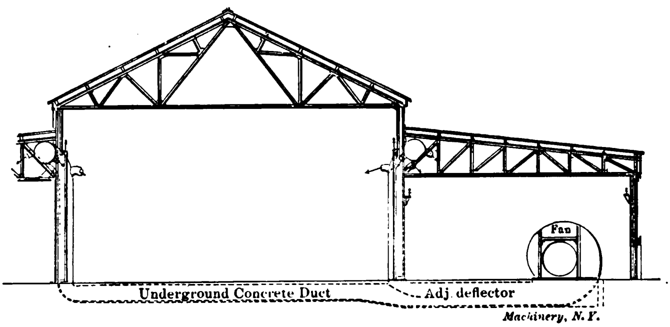


Fig. 34. End Elevation of Installation shown in Fig. 35

direction of movement for a considerable distance without the use of conducting pipes. It is not uncommon in such cases to force the air 100 feet or more from the outlets at a velocity of 2,000 to 3,000 feet per minute.

Weight of Iron Used

Table XVI gives the gage of iron commonly used for pipes of different diameter. All sizes above 60 inches are made of No. 16 gage. If the pipe is made much lighter, particularly in the larger sizes, it will not keep its shape when supported horizontally, which results in loosening the joints and also decreasing the area of the pipe. The common practice is to make rectangular pipes of the same gage as round pipes having the same sectional area, but under certain conditions, as in the case of a thin, flat pipe, bracing is necessary to prevent sagging, even with heavy gages. When braces are used, lighter iron may be used than given in the table.

Heaters

The subject of heaters for shop heating was quite thoroughly discussed in the previous chapter. A few of the results noted there will

TABLE XVI

Diameter of Pipe.	Gage of Iron
Less than 9 inch	28
9 inch to 14 inch	26
15 inch to 20 inch	25
21 inch to 26 inch	24
27 inch to 35 inch	22
36 inch to 46 inch	20
47 inch to 60 inch	18
61 inch and above	16

be given here together with some special reference to heating by air rotation. In shop practice, the amount of heating surface is generally expressed in linear feet of one-inch pipe for a given space to be heated. This, for average conditions, may be taken as follows. With all of the air taken from out of doors, there is generally allowed 100 cubic feet of space for one foot of pipe when exhaust or low-pressure steam is used, and 150 cubic feet with steam at 80 pounds pressure. When the building is heated by air rotation, the aboxe figures may be raised

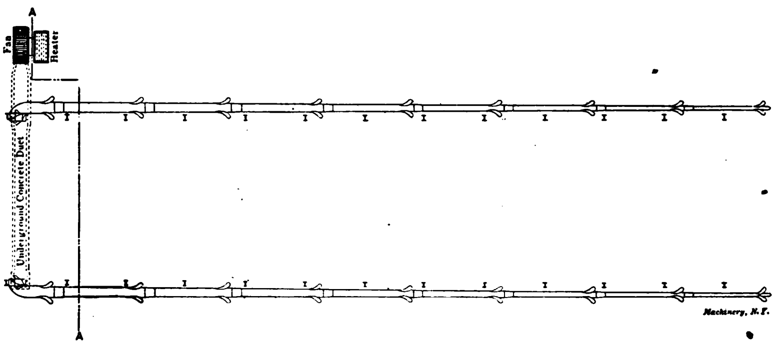


Fig. 35. Plan of Heating and Ventilating System of the Houston, Stanwood & Gamble Company, Covington, Kentucky

to about 140 and 200 for low-pressure and high-pressure steam, respectively. The heater is generally made about twenty pipes deep under ordinary conditions. Heaters of this type have an efficiency of about 1,300 heat units per square foot of surface for steam at 5 pounds pressure, and an efficiency of 1,600 for 60 pounds pressure.

Volume of Air Required

When the air is taken from out of doors for the purpose of ventilation, it may be based upon the number of occupants or upon a given number of air changes per hour. Usually the cubic contents is large per occupant and may vary considerably in different shops, so that under ordinary conditions it is best to use the former method. The air supply per occupant may be taken as about 25 or 30 cubic feet per minute, unless the building is very openly constructed, in which case the air volume may be reduced and leakage depended upon to a considerable extent. In many shops the heating is done entirely by air rotation, and leakage is depended upon entirely for ventilation. This is made possible because of the large enclosed space in proportion to the number of occupants and the thorough mixture of the inleaking air with that which is in rotation. When this method of heating is used, the air simply becomes the medium for transferring the heat to the different parts of the building, and the volume required will depend upon the amount of heat to be transferred and the temperature to which the air is raised.

Suppose the air is returned to the heater at a temperature of 60 degrees and delivered at a temperature of 140 degrees, the total rise being 80 degrees. In cooling one degree, one cubic foot of air gives out 1/55 of a heat unit, or in cooling 80 degrees will give out $1/55 \times 80 = 80/55$, or 1.4 heat unit. Therefore, if we divide the total amount of heat to be supplied in a given time, expressed in heat units, by 1.4, it will give the volume of air to be rotated in that time, assuming, of course, that it is cooled through 80 degrees during its passage through the room.

The heat loss from the building may be computed by any of the common methods in use, or the size of the heater may first be computed by the method already given, and the heat given off taken as the equivalent of the heat lost. Referring to Table XIV we find that a heater twenty pipes deeps with steam at 5 pounds pressure will raise the temperature of air from 0 to 140 degrees, and has an efficiency of 1,300 heat units. Steam at 5 pounds pressure has a temperature of 227 degrees. The average temperature of the air passing through the heater is $\frac{0 + 140}{2} = 70$ degrees, hence the difference in temperature between the steam and air is $227 - 70 = 157$ degrees. In case the air is rotated, its average temperature is $\frac{60 + 140}{2} = 100$ degrees, and the difference between the steam and air is $227 - 100 = 127$ degrees. The efficiency of a heater varies directly as the difference between the temperature of the steam and air; hence, in the second case, with the air rotated, the efficiency would be $157 : 127 = 1{,}300 : x$, and x, the efficiency in this case, would be approximately 1,100 heat units. Then the square feet of surface in the heater multiplied by 1,100 will be the heat given off per hour, and this divided by 1.4 will give the cubic feet of air to be moved per hour by the fan.

Size of Fan

The required size of fan for moving any given volume of air may be taken from Table XVII, which also gives the approximate speed and the horse-power required for driving the fan.

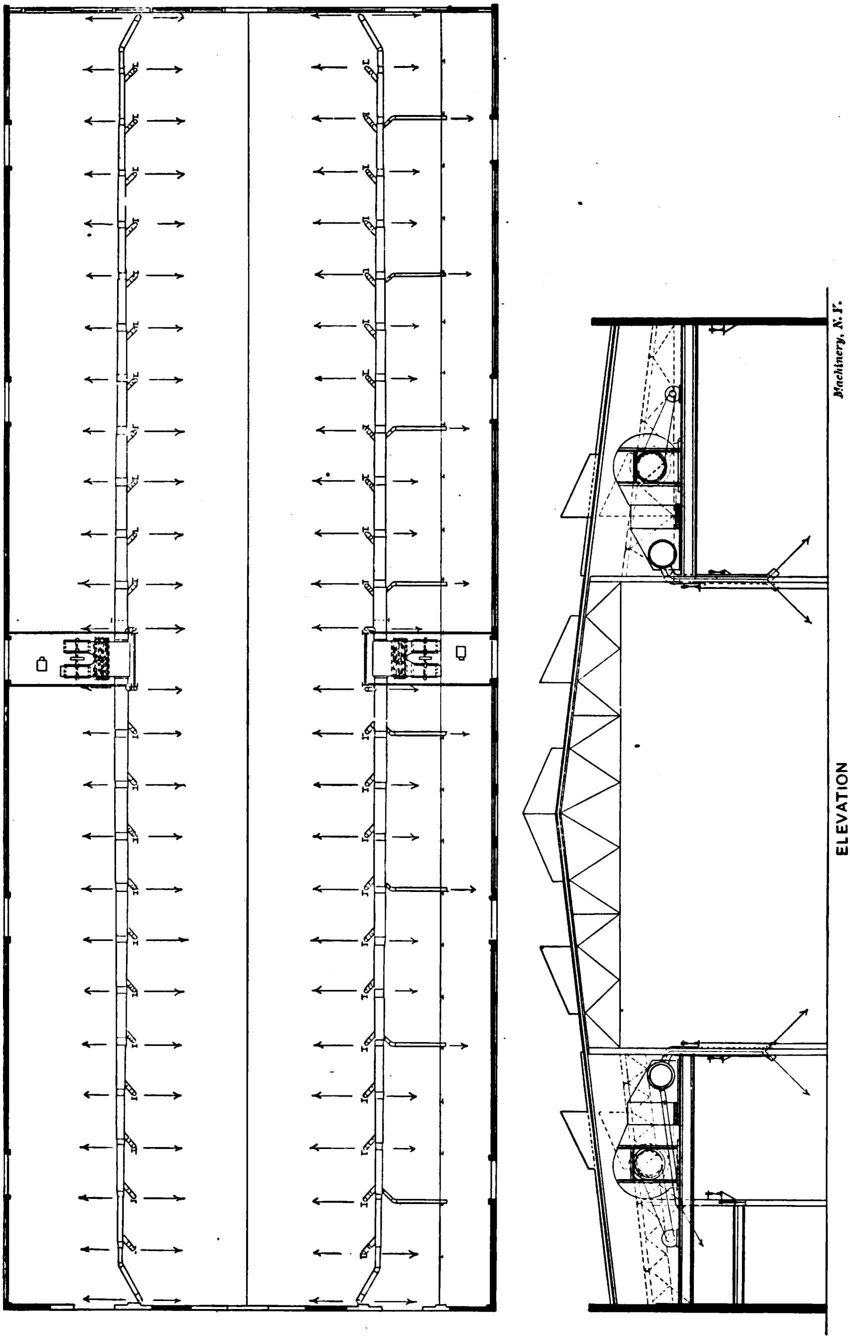


Fig. 36. Heating System of the Pennsylvania Railroad Company's Machine Shop at Trenton, N. J.

TABLE XVII

Normal Size of Fan, Height of Housing in Inches	Diameter of Fan Wheel in Inches	Width of Housing in Inches	Ordinary Speed Giving ½ Ounce Pressure	Cubic Feet of Air Delivered per Minute	Horse-power of Engine to Drive the Fan
30	18	9	870	1000	½
40	24	12	580	1600	1
50	30	15	465	2600	1
60	36	18	390	4500	2
70	42	21	333	6000	2½
80	48	24	293	8000	2½
90	54	28	260	11000	4
100	60	32	233	12500	4
120	72	43	195	21500	7
140	84	48	167	28600	9
160	96	48	147	31800	10
	108	54	130	40400	13
	120	60	117	51000	16

The speeds given in the table are for ½ ounce pressure; should it be desired to deliver the air under a higher pressure, in order to force it a long distance from the outlets, it would be necessary to increase the speed of the fan somewhat, depending upon local conditions.