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The study of ventilation by the centrifugal fan system

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The Study
of
Ventilation by the Centrifugal Fan System.

Historical Review of Ventilation.

Ventilation in a general sense,is the continuous and more or less systematic changing or renewal of the air in a room or in closed space. It involves a study so wide and important in the preservation of life and promotion of health that the writers deem it profitable to give a short history of the subject particularly in its relation to mining.

From the earliest times the problem of mine ventilation has been a serious one. Upon the proper observance of the laws of ventilation depend the health and often the life of the miners and to a large extent,the economy of the plant. The earliest method of forced ventilation of a mine shaft,of which we have record,was carried on by shaking a large cloth at the mine entrance. Following this we have descriptions of large feather fans revolved revolved by hand or wind power and incased in wooden boxes. The ancient miners also utilized the principle of the heated column of air which is still extensively employed in mines having no dangerous gases.

At the present time there are in general four different methods of ventilation:

(1) The Natural, (2) The hot air or motive column,(3) the forced draft method by means of Fans or Blowers, (4) The forced draft method by means of compressors.

In the first method the mine has two openings,one at a higher elevation than the other and the difference in temperature of the air in the mine and above ground causes the air to flow. A shaft for such purposes is called an "up cast for ascending currents and a "down cast" for descending currents.

The second method operates on the same principal, artificial heat being applied at the "up cast" shaft. Ventilation by compression is used almost entirely in metal mining, the compressed air doing double duty in driving the machine drills and ventilating the face of the workings.

The forced draft by fans and blowers is the most extensively used not only in mines but in buildings of all kinds and it will be the aim of the writers to ascertain by experimental facts which may be of practical benefit both in building and mine ventilation.

Description of the System.

The system under investigation was designed by the National Blower Works and consisted of

One Fan (see fan drawing) ten feet in diameter (guaranteed to deliver forty four thousand cubic feet of air at one hundred and thirty six revolutions per minute) housed in a three sixteenths inch steel plate casing which stands two hundred inches high from bottom of wheel pit to top of housing.

One General Electric Motor of twenty horse power capacity at two hundred and twenty volts.

A plenum chamber thirty four feet by twelve feet six inches, from which extend forty galvanized iron air ducts varying in size and length.

The heating system, which consists of steam coils and a patent system of temperature regulation, was not studied in connection with this work.

Description of Apparatus Used in Experiments.

The instrument used in measuring the velocity of the air currents was a Eugene Dietzgen Air Meter or Anemometer.

The meter is made on the principle of the pin wheel,each revolution of the wheel being recorded on the dial as the velocity of the air in feet. The instrument is thrown in and out of gear by moving a lug at the side. Readings were usually taken a minute apart. Along with the instrument the makers sent a correction in the form of a curve,the abscessas of which represented different velocities and the ordinates gave the number of feet to be added to or subtracted from the dial reading to give true reading.

A series of tests were made on the anemometer and the results indicate that the above mentioned curve does not at all represent the constant or coefficient of the instrument.

The tests were made as follows:

The meter was attached to one end of a wooden arm,the other end being fastened to a horizontal wheel which was belt connected to an electric motor. Thus the arm was caused to move in a circle of ten feet in diameter in a tightly closed room and observations were taken at speeds varying from two and five sixths revolutions to twenty six and one tenth revolutions per minute or from eighty eight and nine tenths feet to eight hundred and nineteen and five tenths feet per minute. These observations were plotted,the abscessas representing the number of feet registered by the meter and the ordinates the distance travelled by the meter. The result was a straight line in the form $y = ax + b$, or $y = .75 + 35$. In order then to get a true reading we must multiply the real reading by .75 and 35, or better still read directly from the plotted line. (See Plate No.111).

The instrument used for reading pressures was a simple oil gage graduated to millimeters (oil was used instead of water,it being more sensitive to slight pressures).

Some little time was taken up in the study and construction of a differential oil gage which proved unsatisfactory.

The accompanying sketch will show the main features:

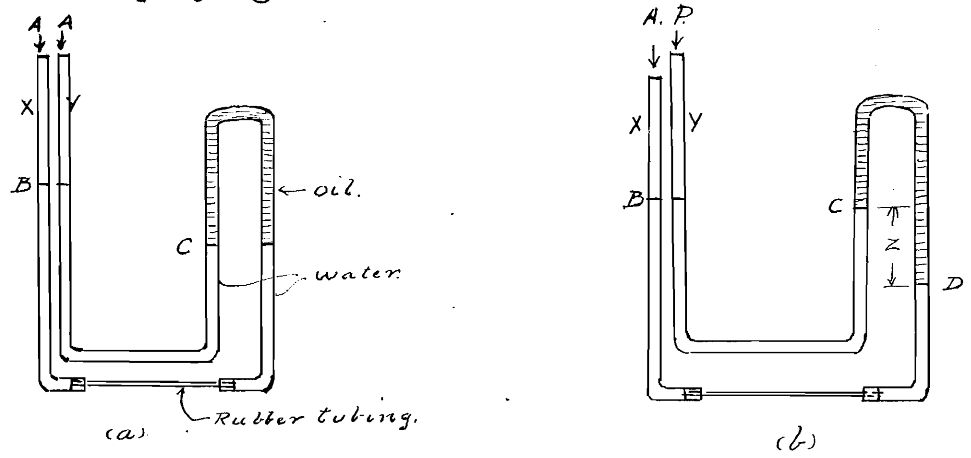


Figure (a) shows the gage balanced with atmospheric pressure, acting on both tubes. The water is level at B and oil at C. In figure (b) a positive pressure P is acting on tube (y) and atmospheric pressure on tube (x(. The first tendency is naturally to depress the wator level in tube (y) until the difference in head in the two tubes represents the excess of pressure and we have a simple water gage.

But lower tube (x) gradually until the water is again level at (B) and the pressure is measured by (Z)x(water-oil). Let 1 = Specific gravity of water and .8 = specific gravity of oil. Thus $P = (1-.8)Z = .2Z$ This shows that Z must be large and the gage correspondingly delicate. The reason assigned for the uncertain results obtained with this instrument is the difficulty in bringing the water in the tubes exactly level (very small errors in the water level would be increased five times in the other arm) Therefore the conclusion reached was that the gage was neither more accurate nor more delicate than the ordinary water gage.

Study of the Fan.

The fan was examined with the idea of obtaining reliable information concerning its capabilities at different speeds and if possible to establish more or less empirical formulae regarding its relations. The data was collected at different times and under varying conditions, the results being effected by the winds, temperature and barometric pressure. Thus all results obtained are more or less approximations but their value lies not in extreme accuracy but in that they were obtained under actual working conditions. No attention was paid to the relation between volumne and pressure as given by Boyle's Law since the pressures were so small as to be inconsiderable in that connection.

All discharges were measured at the intake windows but since the intake is practically equal to the discharge the error is very small.

Explanation of Table and Curves.

Table and Curve No.1 (showing the relation between speed of Fan and quantity of discharge) This curve plots in the form of a parabola and the logarithmic curve gives the equation $S = .945\ D^{1.3}$ (when S = speed of fan in revolutions per minute and D is the discharge in cubic feet per second) Thus it is only necessary to count the revolutions of the fly wheel in order to get the discharge at any given turn. Here it might be stated that the fan was specified to deliver forty four thousand cubic feet of air per minute while running at a speed of one hundred and thirty six revolutions per minute. In reality it delivers only thirty thousand seven hundred and twenty cubic feet at a speed of one hundred and forty one revolutions.

Table and Curve No.11 gives the relation between the pressure head at the intake and the pressure head in the fan drift. The head at the intake is naturally minus and is caused by the friction of the air passing through the tempering coils at the window. If the inlet were free there

there could be no pressure. The head is measured in feet of air. The curve shows that the ratio is a constant one naturally since the conditions are always the same before and behind the fan.

Curve and Table No.111 (Giving Relation between Pressure and Discharge with Constant Velocity and Variable Outlet Openings).

It is known in fluids flowing through passages under a total head H, that the pressure head P is directly proportional to the friction of the passage way. But the total head H = P + velocity head$\left(\frac{V^2}{2g}\right)$. Now H is proportional to the work done by the fan and the work depends on the speed of the fan. Therefore if the speed is constant and the work is practically constant between certain limits then the pressure head will be inversely porportional to the square of the velocity or since the area of the intake remains the same and $v = \frac{q}{a}$ we may write $P = \frac{c}{q^2}$ or $P = \frac{c}{D^2}$ since D and q are the same. The results plotted give the equation $P = \frac{1968}{D^{1.8}}$. It may be assumed that this approximation will hold good for any conditions under which the Fan would be run in practical working operations.

Curve and Table 1V (Giving Relation Between Pressure and Discharge at Constant Opening).

In this case operations are reversed and the pressure head is noted at different speeds of the fan and with a constant outlet.

From a theoretical standpoint this curve is very unsatisfactory since from well known laws the pressure varies as the square of the velocity.

The equation $P = .0023D^{2.8}$ gives some idea of the difficulty of obtaining reliable data in the experimentation of working systems. Possibly the discrepancy was the result of a dynamic pressure caused by cross currents at the point where the gage was read. This would tend to make the pressure increase more rapidly than the square of the velocity or discharge.

Table V. (Showing Relation Between Speed and Work Consumed in Friction by the Motor). was made more for the convenience of the investigators. It was necessary to know the energy consumed by the motor at different speeds in order to get the true efficiency of the fan when the work was measured in Watts delivered to the motor.

Plate 1 contains two logarithmic curves. No.Vl gives the equation between Power and discharge at different speeds and No.Vll and Vlll show the same relation at a constant speed and variable opening.

The equation for No.Vl, $P = .00012\ D^3$ is what would be expected from the fundamental equation. Work $= c\ v^3 = c\left(\frac{Q}{a}\right)^3$. In order then to find the horse power put into the fan,one could refer to equation 1 and get the discharge in cubic feet per second. Substituting the discharge in equation Vl one could calculate the work in foot pounds per second.

Curves Vll and Vlll are not satisfactory as one might be considered as accurate as the other and consequently neither could be absolutely depended upon.

Plate 11 contains only curves and establishes no equations. Curve No.1X gives the relations between Expended Work and the speed of the Fan. It plots in a straight line and shows the ratio to be directly as the first power. In order to get the expended work,the friction work of the motor and belt was subtracted from the entire work as read on the volt and ammeter. The friction work on the motor was obtained in the following manner: A spring balance was attached to the outer end of a fan blade and the fan was rotated by pulling on the balance. This was done a number of times with the belt off and again with the belt on connecting the motor. Let p = pounds pull; n = revolutions per minute of Fan, r = radius of fan in feet. Then work done on friction in feet/pounds/ second $= 2\pi\ r\ n\ p$ — (work on air). The last quantity is small enough to be neglected since the ~~fan was~~.

fan was rotated at a very slow speed (=.3' per second)

The work on both fan and motor = $2\pi r\ n\ p - 2\pi r\ n\ p'$ where p' is the pull when the two are connected.

Substituting values for n we get work in friction(see table V) The above equations depend on the assumption that the friction is constant for different speeds. Some writers claim that this is not true but that friction decreases with the speed up to a certain limit. However this is not thoroughly understood.

Curve X is the line of effective work.

Effective work was found by the formula W = Q P where Q=Discharge in cu ft/ sec. and P is the pressure at the discharge in lbs/square feet.

Cruve Xl represents the mechanical efficiency which is $\frac{\text{effective work}}{\text{expended work}}$ It is a parabola and varies from twenty two to twenty five and one-half percent.

Curve Xll represents the manometrical efficiency or

$$\frac{\text{effective head}}{\text{theoretical head}}$$

The theoretical head was computed from the formula

$$H = \frac{u^2 + uV\cos\beta}{g}$$

(see Theory of Centrifugal Pumps and Fans by Elmo G. Harris, Transactions American Society of Civil Engineers Vol.Ll page 166(1903)

H = total head

u = peripheral velocity of fan ft/sec.

V = velocity of air relative to fan blade ft/sec.

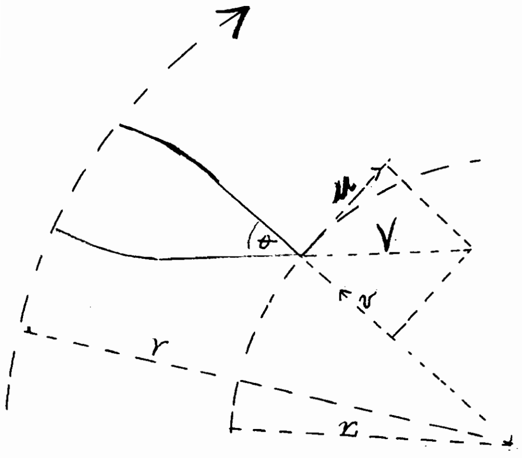
B = angle between direction of fan blade at tips and tangent to circumfrence at same point.

In this case B is 100°- 30' and cos B is a minus quantity but it is small enough to be neglected.

The effective head was obtained by measuring the combined pressure and velocity head in a volute by means of a Pitot tube. The curve is very irregular and the investigators know of no reason for its peculiar shape.

Some Natural Causes of Low Efficiency.

In the first place in order that the air may enter the fan without shock and friction the inner tips of the blades should point in the opposite direction to the relative motion of the incoming air. In the fan under consideration the vanes are radial inside and slope slightly backward at the outer tips. The ingoing air has an absolute velocity V in a radial direction (see Fig.1)



1.

Let u = the peripheral velocity at the inner limit and V = relative velocity. We can see immediately that the air will not glide on the vanes without shock but will cause eddies and energy will be lost. Let us find the angle which will fulfill these conditions at a speed of 100 revolutions per minute.

The intake is 426 cubic feet per second and the absolute velocity between the blades is $\frac{426}{2\pi r_1 b}$ where r_1 is radius of the intake=2-3/4' and b is breadth of fan blade = 3'.

Therefore $v = \frac{426}{52}$ = 8'/ sec. u = $\frac{2\pi r \times 100}{60}$ = 28.8/sec

tan $\theta = \frac{u}{v} = \frac{28.8}{8}$ =3.6. This would make $\theta = 74° - 30'$

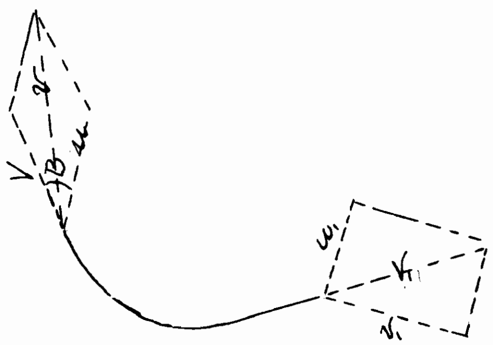
On the other hand the area of the port should equal the area of the throat.

The port has only a clear area of about 25 square feet while the area of the throat is twice as large (over 50 square feet), thus causing a sudden decrease in velocity as the air enters the blades. This together with the consequent drag of the fan blades as they move through the air without scooping it up is bound to lower the efficiency.

The conditions at the outer end of the propellers is somewhat similar. Take the case where the fan has a speed of 100 revolutions per minute. The blade tips move with a velocity of fifty two feet per second. The air in the volute moves with an average velocity of twenty nine feet per second (measurements taken by Pitot tube).

Under this condition there are two losses of energy,one caused by the friction of the air on the sides of the housing (due to excessive velocity) and the other caused by the drag of the propellers,which travels fifty two feet per second. A remedy for both losses might be found in so proportioning the volute that the air would enter it as from flaring nozzle thereby decreasing the speed uniformly without shock and preventing drag and irregular currents.

Again the effective head can be increased by curving the vanes forward instead of backward as in the case under consideration. As was stated before the effective head = $\frac{u^2 + w V \cos B}{g}$



Thus by making B smaller we increase the velocity of the outflowing air and can accordingly decrease the speed of the fan for the same output. This would be detrimental unless the size of the volute were increased accordingly since the friction increases as the square of the velocity. Good practice and economy of material would limit the above modification but one manufacturer has made B as small as twenty two and one half degrees,(Davidson & Co. of Belfast,Ireland) This company makes the Sirocco Fan and to prove that the above statements are not unwarranted the following data taken from the Eng. and Mining Journal April 14,1904 will be submitted:

Make of Fan	Dia. ft.	Speed R.P.M	vol.air moved cu.ft/min.	H.P Delivered
Gimbal No. 1	30	62	199.38	175
" " 2	36	52		
Sirocco	6.25	62	229.484	156

Which shows conclusively the advantage of a small high speed Fan built on the principle above stated.

Experimentation with the air ducts was carried on with a view of establishing coefficients of friction under different conditions. Six ducts were tapped near the plenum chamber and velocity and pressure heads were measured by means of a gage and Pitot tube.

Table No. X gives the results taken at different speeds of the fan. Taking the well established formula $P = \frac{Klov^2}{a}$ when p= pounds pressure per square foot, l = length of duct, O = perimeter, v = velocity in feet per second and K = coefficient, a = area of cross section in square feet, values were substituted for p, l, o, v^2 and a, thus obtaining values for K. The table gives $\frac{v^2}{2g}$ which was changed to v^2, and p as pressure head in feetn of air which was changed to pounds/square foot by multiplying by .076 (weight of one cubic foot of air at sixty degrees Farenheit and twenty nine (29") B.P.)

The coefficients were found to vary only slightly with the speed the tendency in most cases being to increase.

Ducts 23 and 24 are of the same length and size but open into different length flues.

The coefficient of Duct 23 (short flue) = .0000082; That of number 24 = .0000163, which shows that flues having a rougher rubbing surface increases the friction factor. The coefficients for the different ducts vary considerably but the average over all is =.00000973 which is slightly higher than that given by one author (.000008).

In order to ascertain if sufficient quantities of air were being circulated, one room was tested and the volumne calculated. The volumne of the room equalled 18835 cubic feet. If the air were changed three times per hour, there would be required 3 X 18835 cubic feet = 56505 cubic feet per hour. The discharge into the room is 1613.56 cubic feet per minute = 96813.6 cubic feet per hour. This quantity is delivered at ordinary running speed of eighty six revolutions per minute and is one hundred and eighty percent of the amount necessary from a sanitary standpoint, supposing the room was to be occupied by thirty persons.

The results of the foregoing efforts, the authors believe to indicate that a perfect mathematical analysis of a ventilation system is impracticable and unsatisfactory and that only by exhaustive experiments of different systems under varying conditions can authentic information be obtained from which laws may be deduced serving as a guide to designers.

Table No.1.

Speed and Discharge.

Speed of Fan R.P.M.	Quantity of Discharge Cu.ft.per Sec.	Log Speed	Log Quantity
86	325	1.9344	2.5118
103	378	2.0128	2.5774
109	407	2.0374	2.6095
117	427	2.0681	2.6324
124	447	2.0934	2.6503
130	456	2.1139	2.6589
134	397	2.1271	2.5987
141	512	2.1492	2.7092

Table No.11.

Relation between Pressure in Front of

and Behind the Fan in

Head of Air.

Head air in feet	- Head air in feet.
20.2	12.3
24.6	14.6
26.9	17.9
31.4	19.0
34.7	20.2
40.3	23.5
42.6	29.1

Table Number 111.

Pressure and Discharge.

(constant velocity)

Pressure head air	Discharge cu.ft/sec.	Log P.	Log Discharge
22.4	34.3	1.3502	2.5352
23.0	36.6	1.3617	2.5634
23.5	35.0	1.3716	2.5440
24.0	38.0	1.3802	2.5797
24.6	37.8	1.3909	2.5774
26.9	38.0	1.4297	2.5797
29.1	35.3	1.4639	2.5478
40.3	33.0	1.6053	2.5185
42.6	29.2	1.6294	2.4653.

Table Number 1V.

Relation Between Pressure and Discharge

Constant Opening.

Pressure ft. air	Discharge cu.ft/sec.	Log P.	Log.Discharge
26.3	373	1.4199	2.5717
28.0	325	1.4471	2.5118
28.0	366	1.4471	2.5634
29.6	348	1.4712	2.5843
31.0	343	1.4913	2.5352
32.4	384	1.5105	2.5843
32.5	365	1.5118	2.5622
32.5	378	1.5118	2.5774
34.3	394	1.5352	2.5995
36.0	463	1.5563	2.6655
39.2	407	1.5932	2.6095
39.4	410	1.5955	2.6127
44.8	429	1.6512	2.6324
50.4	447	1.7024	2.6503
54.9	456	1.7395	2.6589
54.9	397	1.7395	2.5987
63.8	512	1.8048	2.7092

Table Number V.

Relation Between Speed and Friction of Motor.

Speed R.P.M.	Motor with belt. Friction ft/lbs/sec.
87	455
91	474
95	497
98	512
100	523
101	528.2
107	559.6
114	596.3

Table Number Vl.

Relation Between Discharge and Power)constant opening)

Discharge	Power	Log Dis	Log P.
311	3596	2.4927	3.5558
337	3646	2.5276	3.5615
348	3958	2.5415	3.5974
365.5	4915	2.5628	3.6915
384	4350	2.5843	3.6384
394	5654	2.5954	3.7523
410	6690	2.6127	3.8254

Tables Number V11 and V111

Relation Between Discharge and Power (variable opening)

cu.ft/sec Discharge	ft/#/sec Power	Log Dis.	Log P.
292	4218	2.4654	3.6251
330	4406	2.5185	3.6440
343	4320	2.5353	3.6355
350	4314	2.5441	3.6349
353	4416	2.5478	3.6450
366	3454	2.5635	3.5383
378	4380	2.5775	3.6415
380	4334	2.5798	3.6369
380	4532	2.5798	3.6563

Table Number 1X. for Plate 11

Speed of fan R.P.M.	Expended work ft/lbs/sec	Effective work ft/lbs/sec	Mechanical Efficiency %	Monometrical Efficiency %
87	3194	717.8	22.3	34.7
87	3144	692	22	
91	3485	783	22.5	
95	3856	944.6	24.5	
101	4390	904	20.6	
103	3905	933.7	23.9	33.28
107	5098	1028	20.2	
109	4903.2	1212.9	24.7	33.
114	6092	1228.9	20.1	
117	5764	1458.6	25.3	32.51
124	6455	1681.37	24.5	32.24
130	7454	1901.5	25.5	30.87
93		746		39.3
100		907		34.1
108		1268		31.4
115		1443		
134		1655		38.4
141		2483.2		36.9
148				36.05

Table No.X.

Speed R.P.M.	Duct No.	Length in feet	Area sq.ft	Perimeter in feet	Velocity Head ft of air	Pressure Head ft of air	K.
86	8	36	.555	3	2.35	3.36	.00000872
98	"	"	"	"	3.584	2.24	.00000363
106	"	"	"	"	2.912	3.584	.00000753
114	"	"	"	"	2.688	4.48	.0000102
124	"	"	"	"	6.72	4.48	.00000403
130	"	"	"	"	5.6	6.72	.00000733
141	"	"	"	"	7.84	5.6	.00000436
						Average	.00000654
86	10	45.5	2.14	6'	3.808	4.48	.0000110
98	"	"	"	"	3.352	2.24	.00000889
106	"	"	"	"	4.928	6.048	.0000114
114	"	"	"	"	5.376	6.72	.0000116
124	"	"	"	"	6.272	4.48	.00000665
130	"	"	"	"	6.272	5.6	.00000836
134	"	"	"	"	7.392	6.72	.00000848
						Average	.00000945
86	23	50	.97	4	3.472	3.92	.00000651
98	"	"	"	"	1.792	4.48	.0000144
106	"	"	"	"	1.792	4.48	.0000144
114	"	"	"	"	4.704	4.704	.00000577
124	"	"	"	"	2.912	6.72	.00000173
130	"	"	"	"	6.72	4.704	.00000808
134	"				4.704	7.84	.00000963
141	"	"	"	"	2.24	8.96	.00001000
						Average	.00000882

Table No. X (continued)

Speed	Duct	Length in	Area	Perimeter	Velocity Head	Pressure Head	K
R.P.M.	No.	feet	sq.ft	feet	ft.air	ft air	
86	24	40	.97	4'	1.59	4.48	.0000202
98	"	"	"	"	2.016	4.48	.0000161
106	"	"	"	"	2.24	4.48	.0000144
114	"	"	"	"	2.46	4.48	.0000132
124	"	"	"	"	2.912	6.72	.0000166
130	"	"	"	"	4.704	7.168	.0000111
134	"	"	"	"	4.256	7.616	.0000129
141141	"	"	"	"	2.464	9.856	.0000257
						Average	.0000163
86	31	75	.555	3'	2.464	4.704	.00000293
98	"	"	"	"	2.24	3.808	.00000499
106	"	"	"	"	3.584	6.72	.00000548
114	"	"	"	"	4.704	6.72	.00000420
124	"	"	"	"	3.809	6.72	.00000580
134	"	"	"	"	4.032	7.168	.00000522
141	"	"	"	"	3.472	8.4	.00000707
						Average	.00000501
86	32	74	1.82	4'	2.464	4.704	.0000140
98	"	"	"	"	2.464	4.48	.0000133
106	"	"	"	"	3.136	4.48	.0000144
114	"	"	"	"	2.912	4.704	.0000118
124	"	"	"	"	3.808	2.272	.0000438
130	"	"	"	"	3.808	6.72	.0000128
134	"	"	"	"	3.808	8.96	.0000170
141	"	"	"	"	5.152	10.08	.0000143
						Average	.0000123

Table No. X (continued)

	K
	.0000654
	.0000945
	.0000882
	.0000163
	.00000501
	.0000123
	6.00005843
Average	.00000973