The Oscillating Vane-type Fan

Thesis by

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INTRODUCTION

The investigation herein described was undertaken to determine the performance characteristics of the oscillating vane-type fan. The fan was the invention of Mr. Van Dorn and Mr. Cornwell, primarily for use in circulating air in railroad refrigerator cars. In the present system of railroad refrigeration for the transportation of perishable foodstuffs, the refrigerator car is a heat insulated car, with bunkers at each end for the cakes of ice. When the car is loaded and the doors shut, cooling is obtained only due to the natural circulation of air in the car, caused by the difference in temperature between the ice in the bunker and the fruit. This circulation has not been found sufficient, in that there is a large temperature difference between the top and bottom fruit (see Appendix B) and the top half is too warm, often spoiling during transportation. Also, because of this, the car cannot be filled to its volumetric capacity. Therefore the requirements which led to the invention of this type of fan were as follows:

1. To provide a sufficient volume of air, against the resistance developed in passing through the fruit, to obtain the cooling characteristics required.

2. To provide a fan which would not require major

changes in the structural design of the car, or lessen the amount of fruit which could be carried.

3. To provide a fan, simple in construction and installation, and in its drive mechanism from the car axle.

The oscillating vane-type fan met conditions 2 and 3 in that it could be situated in the four and one half inch space between the floor racks and the floor of the car, through which the air naturally circulated, and in that it could be driven by a simple wheel and crank drive directly off the axle. This thesis presents the results of the investigation of the first condition.

The investigation was undertaken with the assistance of Dr. Von Karman, Mr. Van Dorn, and Mr. Knoblock.

DESCRIPTION OF FAN

The oscillating vane-type fan is a symmetrical airfoil section, pivoted about its leading edge, and oscillating on this pivot through small angles. The vane (or blade) is mounted midway between two walls of a rectangular channel, allowing sufficient clearance for the vane tip for the angular motion and for the vane deflection. The vane, sweeping through an angle, forces air through the channel due to the increased pressure on the leading side, and sucks air through the channel due to the decreased pressure on the trailing side. Because of this suction effect, the quantity of air delivered against low heads is greater than the volume swept through by the blade per stroke.

GENERAL PLAN OF INVESTIGATION

The volume of air delivered against fixed heads for different speeds (r.p.m.) was determined for blades of different chord lengths. The fan was operated as a blower in all tests. This fan was found to follow the standard fan laws and general equations for its performance were formulated. The efficiency of the blade operating as a fan was determined but not thoroughly investigated because it was low and because the efficiency was relatively unimportant in the use for which it was invented. The effect of structural additions was also investigated to determine the best method of installation in the refrigerator car.

EXPERIMENTAL SETUP
(Figures 1, 2, 3)

The test channel was rectangular in cross section, two feet wide, seven feet long, and with an adjustable height so that the proper tip clearance could be obtained to allow for changes in the angle of blade travel and in blade chord length. The blade was driven at both ends, by means of a crank system, from a quarter horsepower D.C. motor mounted on top of the channel. The drive shaft speed was controlled by the use of rheostats in the motor circuit and by changing the belt-pulley wheel ratios between the motor and the drive shaft. The eccentrics were flywheels, on the ends of the drive shaft, with a set of tapped holes in each to give the proper crank throws for blade travels of 20, 25, 30, 35 and 40 degrees. The blade was driven at both ends so that it could not twist.

The blades were made with a three-quarter inch tube forming the leading edge, extending beyond the blade proper to be held in the bearings. The blade turned about the tube center. Arms were braised to the tube, between the ends of the blade and the bearing points, to which the cranks from the eccentrics were attached. The blade length was the width of the channel, two feet, in all cases. The blades were made by forming ribs of thin metal sheet, braising them to the tube, covering with airplane fabric, and doping.

The bearings, holding the blade tube, were mounted on vertical angle iron sections on each side of the channel; the angle sections each had a row of holes so that the height of the bearings might be changed when blade angles or blade chord lengths were changed for the different test runs.

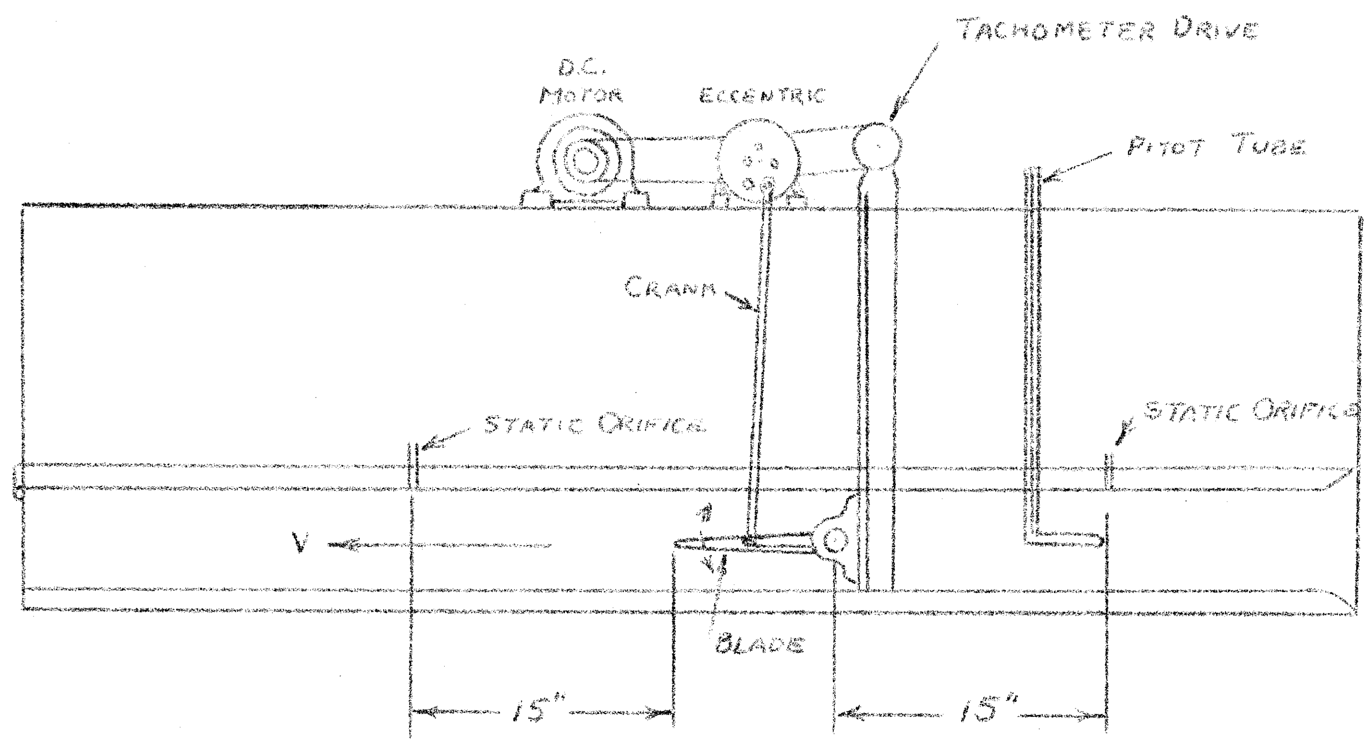
The blade r. p. m. was determined by the use of a chronometric airplane type tachometer. One r.p.m. was one complete up and down movement of the blade. The angle of blade travel is taken as the angle between the extreme up position and the extreme down position of the blade; the distance traveled above and below the horizontal was equal.

The static pressure orifices were located in the top of the channel, one fifteen inches ahead of the leading edge, and one fifteen inches behind the trailing edge of the blade, on the horizontal centerline. The pressure change across the blade is the difference in static pressure between the two static orifices. The static orifices were connected with rubber tubing to a manometer to determine the static pressure change.

The air velocity was determined by the use of a pitot tube mounted fifteen inches ahead of the leading edge of the blade in the center of the channel. (The velocity cross section was found to be uniform except for the expected falling off at the wall.) The pitot tube was connected to a manometer on which the velocity head was read.

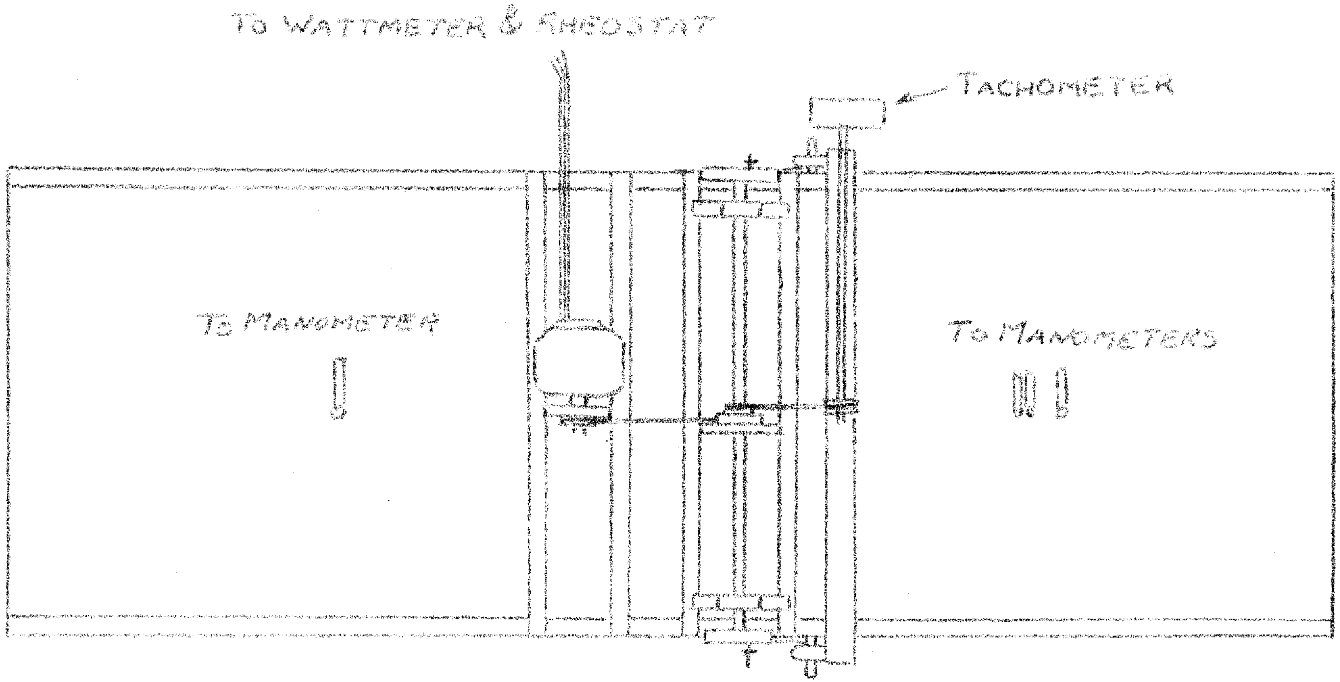
The power input was measured by a wattmeter connected in the motor circuit. A skeleton blade was used to determine the power input necessary to overcome the frictional and mechanical losses in the system.

The percent restriction of the channel exit was varied by fixing half inch metal strips across the channel exit. The blade tip clearance from the top and bottom of the channel was kept at one half an inch.



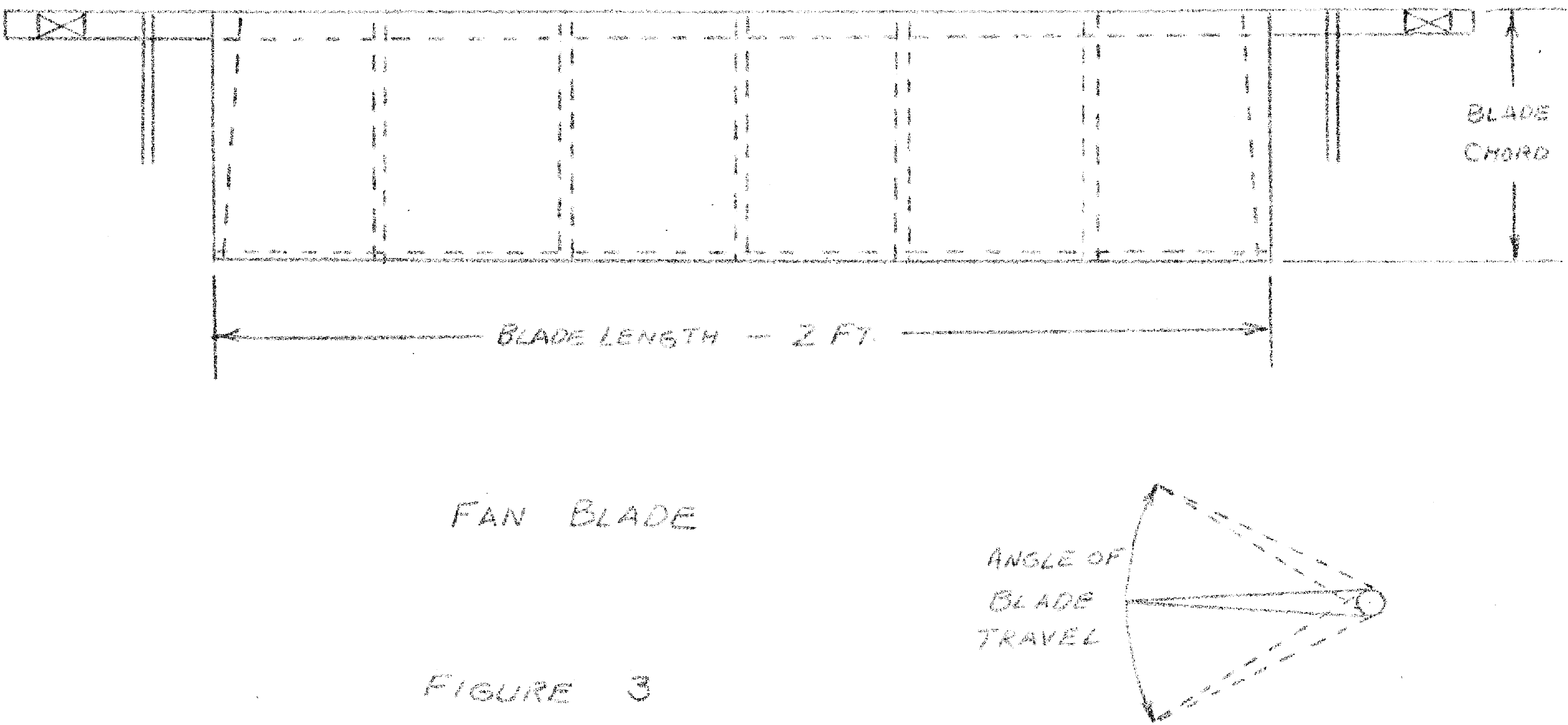
SIDE VIEW DIAGRAM

FIGURE 1



TOP VIEW

FIGURE 2



FAN BLADE

FIGURE 3

TEST PROCEDURE

Blades of three chord lengths were tested, (four and one half, nine, and twelve and three quarter inches.) The blades were run at five angles, 20, 25, 30, 35, and 40 degrees. In each case the air velocity and static pressure change across the fan was found for blade speeds from 600 to 2000 r.p.m., and of channel exit restrictions from zero to one hundred percent. Static pressure was plotted against air velocity for the different percent restrictions to give the characteristic fan curves.

The efficiency was determined only with the nine inch blade. The power required to run the blade at r. p. m. from 600 to 2000 was determined; similarly the power required to run the skeleton blade was found; the air horsepower was calculated from the air velocity and the pressure change across the fan. The blade horsepower was then taken as the difference between the power required to run the blade and the skeleton blade, and the blade efficiency calculated by dividing the air horsepower by the blade horsepower. The overall efficiency was not calculated because in this experimental setup the friction and vibration were much greater than they would be in a commercial application.

TEST RESULTS

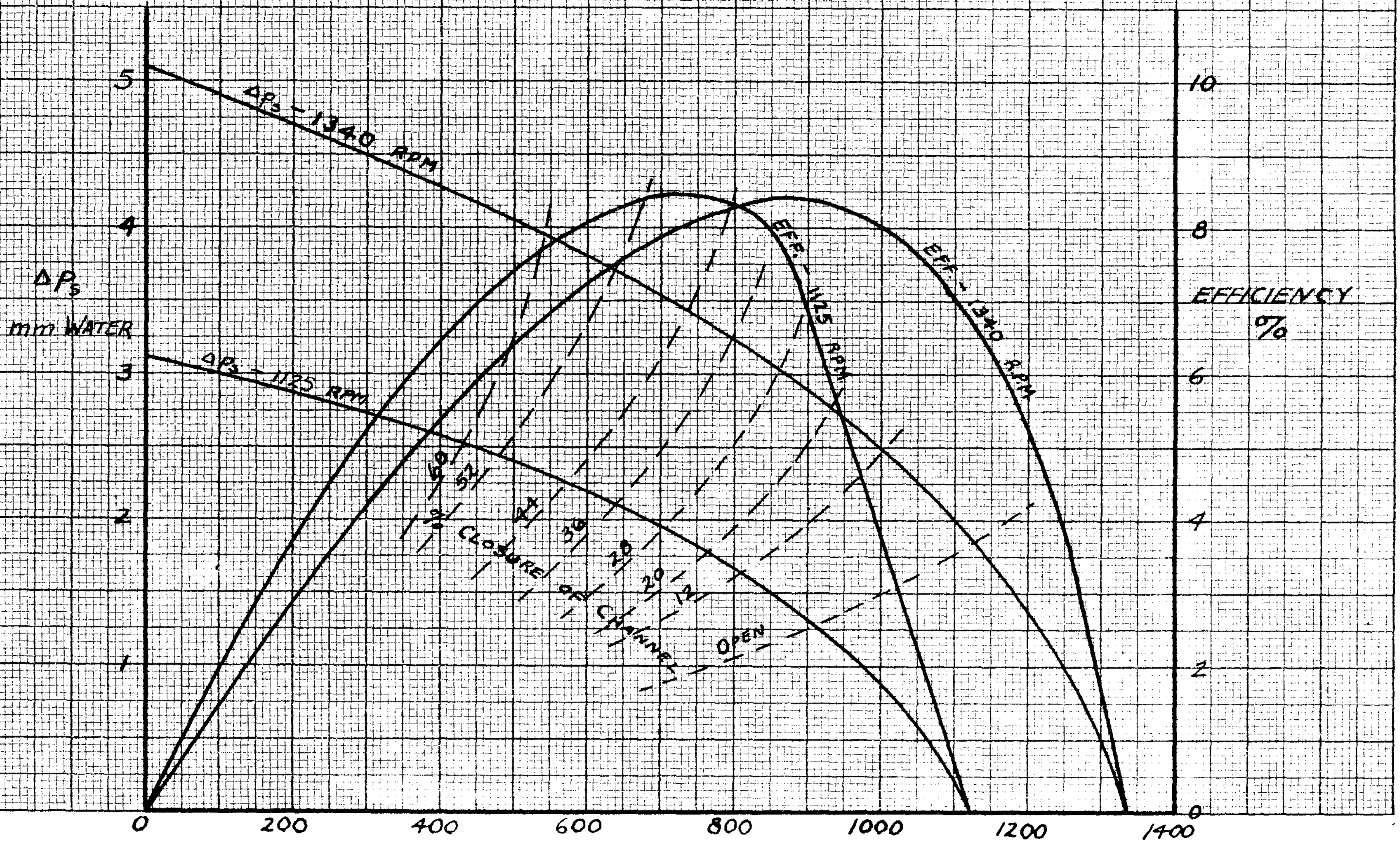
When a conventional fan is operated at various speeds through a given duct system, the volume of air moved varies directly as the fan speed, the static pressure developed varies as the square of the speed, and the power required varies as the cube of the speed. The oscillating vane-type fan was found to follow these laws and the performance equations deduced accordingly. The static pressure and the power also vary as the air density. The air velocity was found to vary as the chord length, and the static pressure as the square of the chord length. The variation with blade angle of travel is shown in figure 8.

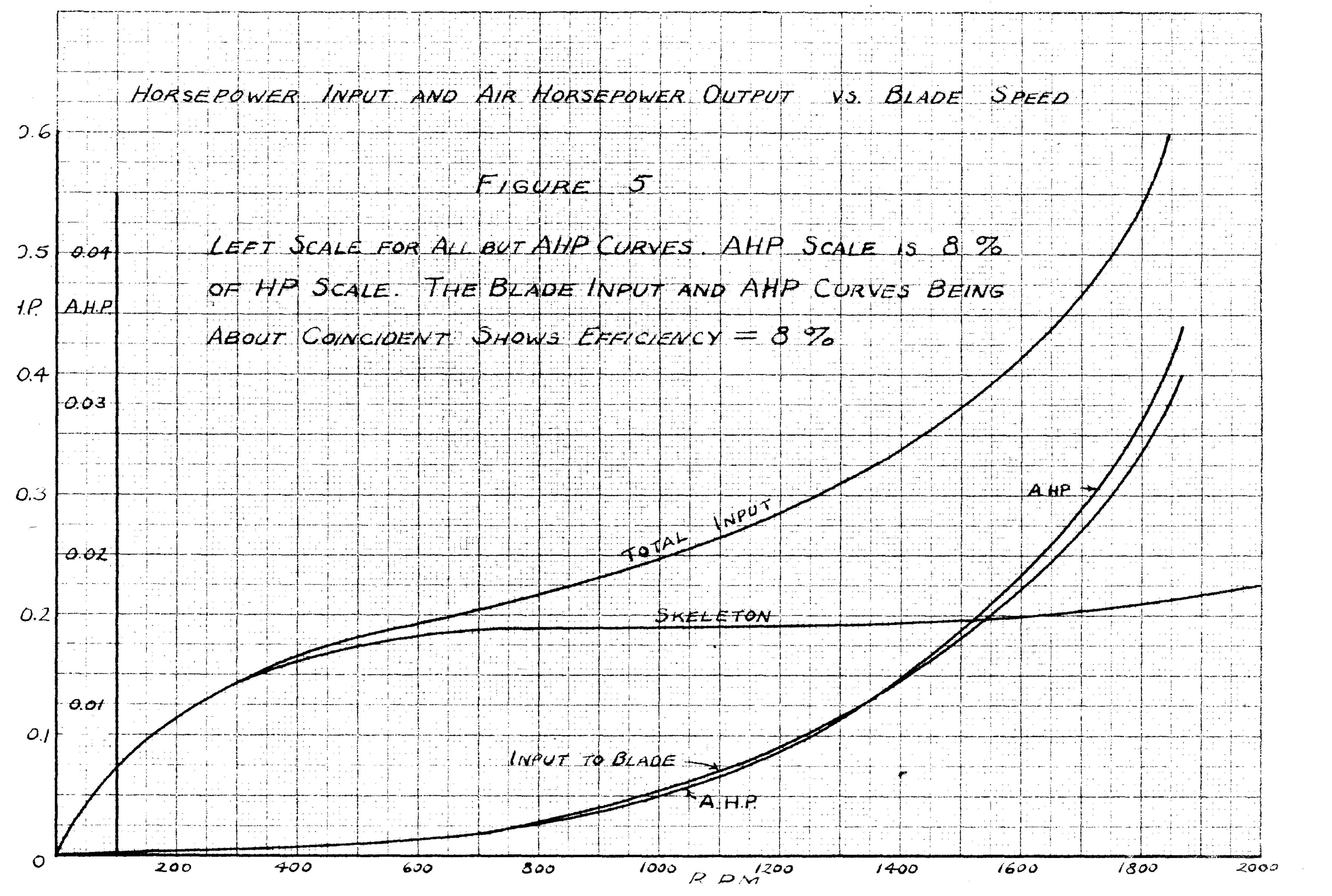
Figure 4 shows the typical static pressure versus air velocity curves. This fan has the steep or falling type characteristics of a low pressure fan. Figure 4 also shows the blade efficiency versus air velocity curves. The blade efficiency was found to be eight percent, occuring at a channel restriction of thirty percent of open channel, or sixty-five percent of the maximum air velocity. Since the blade efficiency was so low, it was only determined for one case, the nine inch blade with twenty degrees blade travel. Figure 5, horsepower versus r.p.m. curves, shows that the blade efficiency varies only slightly with blade speed. The output (air horsepower) was determined in two ways for a check: first by multiplying the quantity by the pressure change across the fan, and secondly by multiplying the quantity by the velocity head plus the static pressure (above atmospheric pressure) at the rear orifice.

PRESSURE CHANGE ACROSS FAN AND BLADE EFFICIENCY vs. AIR VELOCITY

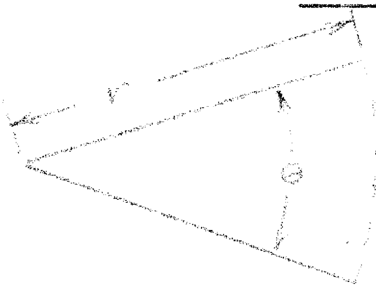
9 INCH BLADE – 20°

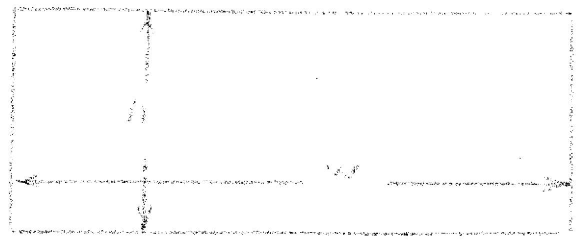
FIGURE 4




HORSEPOWER INPUT AND AIR HORSEPOWER OUTPUT VS. BLADE SPEED
FIGURE 5
LEFT SCALE FOR ALL BUT AHP CURVES. AHP SCALE IS 8 % OF HP SCALE. THE BLADE INPUT AND AHP CURVES BEING ABOUT COINCIDENT SHOWS EFFICIENCY = 8 %
H.P.
A.H.P.
0.6
0.5
0.4
0.3
0.2
0.1
0
0.04
0.03
0.02
0.01
TOTAL INPUT
SKELETON
AHP
INPUT TO BLADE
A.H.P.
200
400
600
800
1000
1200
1400
1600
1800
2000
R P M

DERIVATION OF EQUATIONS





r = blade chord $\quad$ w = channel width

θ = blade angle of travel $\quad$ N = blade speed (r.p.m.)

h = channel height

Swept Quantity $= Q_s = \dfrac{\pi r^2 \theta N w (2)}{360}$

Cross sectional Area $= A = w h$

Swept Velocity $= V_s = \dfrac{Q_s}{A}$

Actual Velocity $= V = \dfrac{\phi C Q_s}{A}$

The variation of V and P with θ is shown in Fig. 7. In order to make the equations for V and P hold for any angle θ, an angle coefficient ϕ is included in the equations, derived as explained on pg. 12.

ϕ = angle coefficient.

C = velocity coefficient (See pg. 10)

$$V = \frac{\phi C \pi r^2 \theta N w (2)}{360 h w}$$

making $h = \dfrac{\pi r \theta}{180}$

($h = 2r \sin \frac{\theta}{2}$ actually, but the angles considered being small and r large, the error is not large and is taken care of in the determination of the coefficient C.)

$$V = \frac{\phi\, C\, \pi\, r^2\, \theta\, N}{\pi\, r\, \theta} = \phi\, C\, r\, N \qquad \text{(Equation 1)}$$

Static Pressure change across the fan:

$$P = f\, V^2 \sigma$$

f = pressure coefficient

σ = relative air density

From equation 1 we get:

$$P = \sigma\, f\, \phi^2\, r^2\, N^2 \qquad \text{(Equation 2)}$$

(C is left out of this equation for simplification, and its variation is taken into account in the determination of the pressure coefficient f.)

Quantity

$$h = 2\, r \sin \frac{\theta}{2} + \frac{1}{12} \qquad \text{(Equation 3)}$$

where h and r are in feet. The added factor of one-twelfth is the sum of the clearance top and bottom in feet.)

$$Q = A\, V = w\, (2r \sin \frac{\theta}{2} + \frac{1}{12})\, V$$

$$= w\, (2\, r \sin \frac{\theta}{2} + \frac{1}{12})\, \phi\, C\, r\, N \qquad \text{(Equation 4)}$$

If the velocity V is compared to another velocity V, for a different set of conditions, the velocity ratio is:

$$\frac{V}{V_,} = \frac{\phi\, C\, r\, N}{\phi_,\, C_,\, r_,\, N_,} \qquad \text{(Equation 5)}$$

$$(N)^2 = \frac{(V\,\phi_,C_,r_,N_,)^2}{(V_,\phi\,C\,r\quad)}$$

Similarly comparing two pressures, using equation 2, we have:

$$(N)^2 = \frac{P_,\sigma\,f\,\phi^2\,r^2\,N^2}{P\,\sigma_,f_,\phi_,^2\,r_,^2}$$

Substituting, we get:

$$\frac{\sigma\,P_,\,f\,V^2\,C_,^2}{\sigma_,\,P\,f_,\,V_,^2\,C^2} = 1$$

If $f_, = f$ and $C_, = C$: $\frac{P}{\sigma V^2} = \frac{P_,}{\sigma_, V_,^2}$ (Equation 6)

Therefore $\frac{P}{\sigma V^2}$ is a useful parameter and is used as the base for C and f.

$$V = \phi\,C\,r\,N$$

$$P = \phi^2\,\sigma\,f\,r^2\,N^2$$

$$N = \frac{V}{\phi\,r\,C} = \left(\frac{P}{\phi^2\sigma\,r^2 f}\right)^{\frac{1}{2}}$$

$$\frac{P}{\sigma V^2} = \frac{f}{C^2} \qquad \text{(Equation 7)}$$

DETERMINATION OF COEFFICIENTS

The coefficients C and f were determined to describe the curves for r = 9 inches, θ = 20 degrees. The experimental curves (figure 7) were reduced to N = 1000 r.p.m. by the relationships $\frac{V}{V_,} = \frac{N}{N_,}$ and $\frac{P}{P_,} = \frac{N^2}{N_,^2}$ (figure 6)

The experimental error caused these curves not to be exactly coincident so the average of the reduced curves was drawn (figure 7), and the coefficients C and f calculated from this averaged curve.

$$C = \frac{\text{Actual } V}{r\ N} = \frac{\text{Actual } V}{9/12\ (1000)}$$

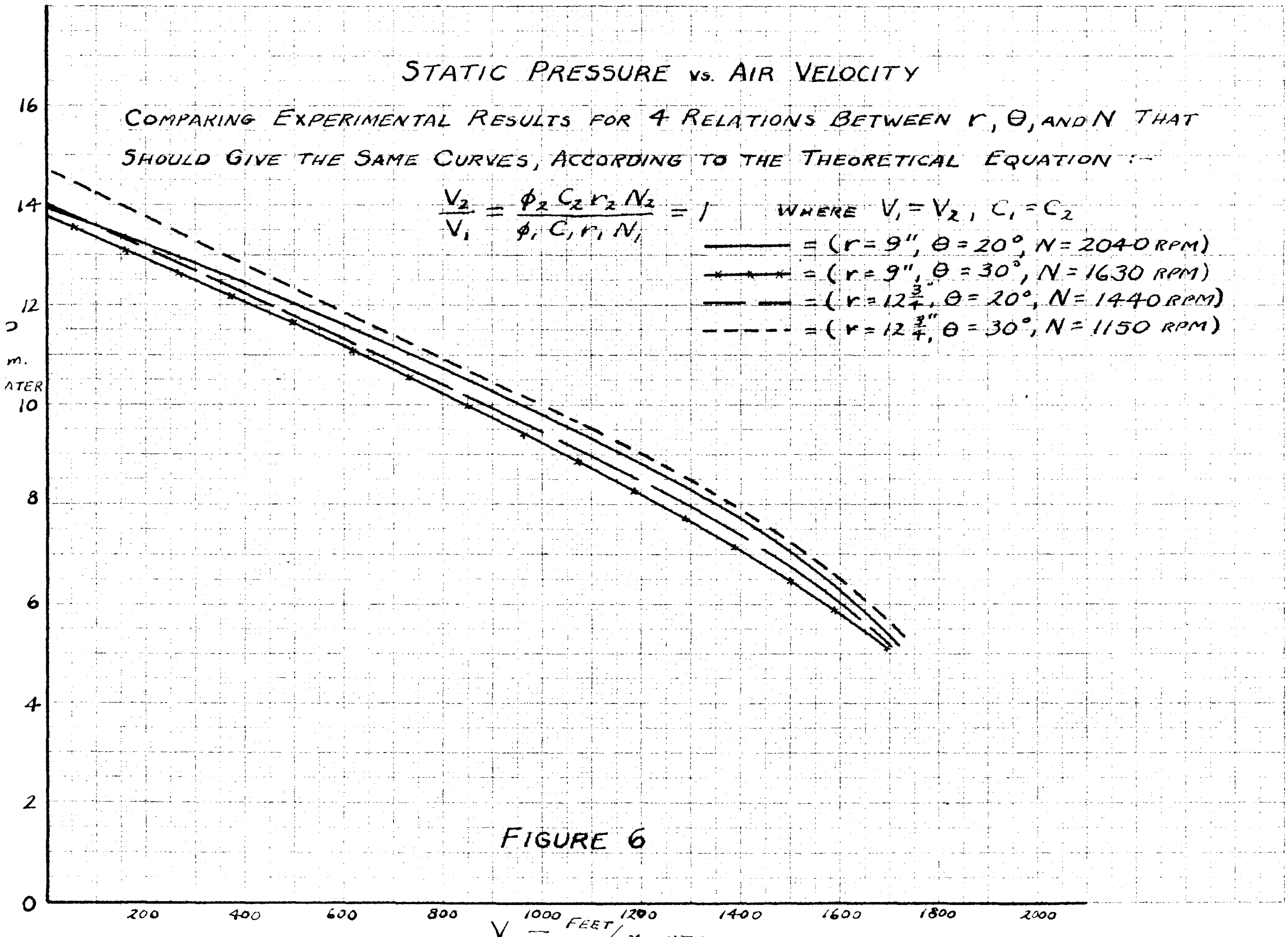
$$f = \frac{\text{Actual } P}{r^2\ \ N^2}$$

As shown by equation 6, $P/\rho V^2 = P_1/\rho_1 V_1^2$ is a constant when $C = C_1$ and $f = f_1$; therefore C and f were determined for the different values of $P/\rho V^2$.

Calculation of coefficients f and C:

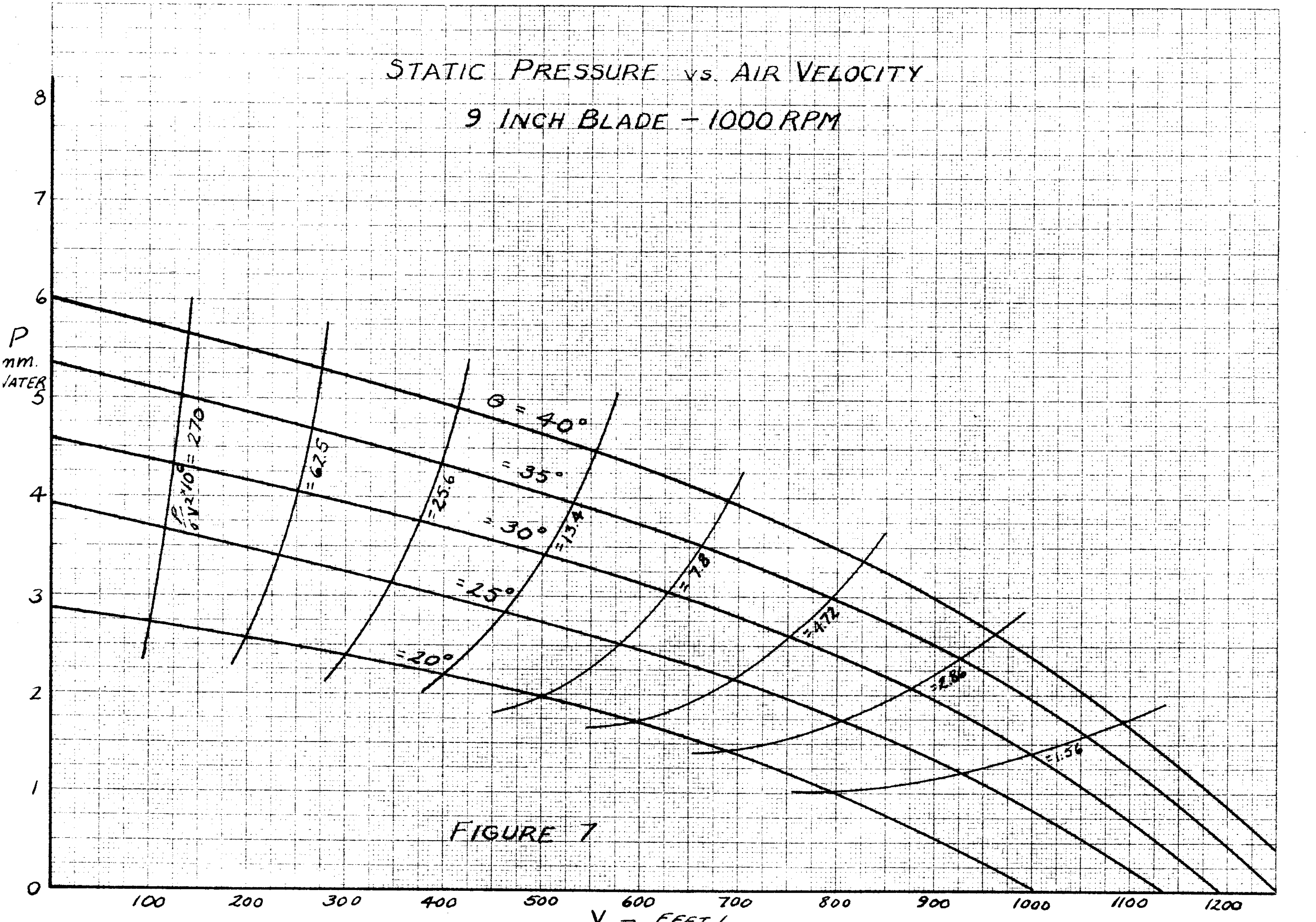
V is in feet per minute, P in milimeters water pressure: ρ taken equal to 1.

$10^6\ P/V^2$	V	P	C	10^6 f
0	1000	0	1.330	0
0.1	995	0.07	1.328	0.124
0.74	900	0.60	1.200	1.070
1.56	800	1.00	1.070	1.780
2.86	700	1.40	0.934	2.490
4.72	600	1.70	0.800	3.020
7.80	500	1.95	0.666	3.460
13.40	400	2.15	0.534	3.820
25.60	300	2.30	0.400	4.090
62.50	200	2.50	0.266	4.450
270.00	100	2.70	0.133	4.800
Inf.	0	2.85	0	5.060



STATIC PRESSURE vs. AIR VELOCITY
COMPARING EXPERIMENTAL RESULTS FOR 4 RELATIONS BETWEEN r, θ, AND N THAT
SHOULD GIVE THE SAME CURVES, ACCORDING TO THE THEORETICAL EQUATION :–
$\frac{V_2}{V_1} = \frac{\phi_2 C_2 r_2 N_2}{\phi_1 C_1 r_1 N_1} = 1$ WHERE $V_1 = V_2$, $C_1 = C_2$
$= (r = 9''$, $\theta = 20°$, $N = 2040$ RPM$)$
$= (r = 9''$, $\theta = 30°$, $N = 1630$ RPM$)$
$= (r = 12\frac{3}{4}''$, $\theta = 20°$, $N = 1440$ RPM$)$
$= (r = 12\frac{3}{4}''$, $\theta = 30°$, $N = 1150$ RPM$)$
0
2
4
6
8
10
12
14
16
m.
ATER
200
400
600
800
1000
1200
1400
1600
1800
2000
V — FEET/MINUTE

FIGURE 6



STATIC PRESSURE vs. AIR VELOCITY
9 INCH BLADE - 1000 RPM
P mm. WATER
V - FEET/MINUTE
θ = 40°
= 35°
= 30°
= 25°
= 20°
P/ρV² x10⁶ = 270
= 62.5
= 25.6
= 13.4
= 7.8
= 4.72
= 2.86
= 1.56

FIGURE 7

These results were plotted on log paper (figure 9) to make the curves more accurately readable; the accuracy of the coefficients is important in these calculations. The range of P/V^2 values plotted in figure 9 covers 87 percent of the total range and is all that is necessary.

DETERMINATION OF THE ANGLE COEFFICIENT ϕ

The curves (figure 7) plotted for N = 1000 r.p.m., θ = 20, 25, 30, 35, and 40 degrees from the reduced and averaged experimental results are of similar nature. It was found that the θ = 20 degrees curve, enlarged by the ratios $V/V_{20} = N/N_{20}$ and $P/P_{20} = N^2/N_{20}^2$, would very closely coincide with the θ = 25, 30, 35, and 40 degree curves. Considering this approximation sufficiently accurate, the angle coefficient was determined by the relationship: ϕ = N necessary to enlarge 20 degree curve to be coincident with the θ degree curve divided by the N of the 20 degree curve (= 1000 r.p.m.)

θ	20	25	30	35	40	degrees
N_θ/N_{20}	1.00	1.16	1.25	1.32	1.38	(Figure 8)

ACCURACY OF WORK

The experimental error was estimated to be 5 percent. Inaccuracies were due to the instruments used, plus the mechanical difficulties of blade deflection and vibration. The curves and formulas are to be used to give a close approximation of the blade sizes and blade speeds necessary to meet given requirements of pressure and velocity.

DISCUSSION

It is interesting to note that the output quantity, open channel, is greater than the quantity of air swept through by the blade per stroke. This shows that the blade is acting as an airfoil, deriving much of its 'push' from the suction on the trailing side of the blade. This is largely responsible for the large output per r.p.m. of this type fan.

The low efficiency must be a result of the fact that the passage for the air by the blade is constantly being half closed and opened, and the air must change its direction of flow from over to under the blade once every stroke. Much of the energy must be absorbed in friction, air turbulence, and in the rapid change in velocity head at the blade passage. The blade does not travel in any uniform path relative to the air as does the blade of a screw or cycloidal type propeller fan, nor does it have the pure force reaction of a centrifugal fan. The force is pulsating and therefore the efficiency is low.

The effect of a fixed leading edge ahead of the blade was investigated. The mechanical advantage of such an arrangement was realized for the hinging of a long blade as was planned to be used in the refrigerator car installation. A four and one half inch leading edge was fixed ahead of the nine inch blade, shaped as the leading edge of an airfoil. Pieces of rubber, one on top and one on the bottom, were glued to the fixed part and to the leading edge of the moving section. This reduced the vibration of the system, but did not change the output or the pressure developed by the fan.

Greater increases in length of the fixed leading edge decreased the performance of the fan, due to the fact that they increased the frictional resistance to the air in the channel. There was one exception to this. A ten inch thin metal sheet fixed ahead of the four and one half inch fixed leading edge already installed, changed the characteristics of the fan. This addition increased the maximum air velocity fifteen percent and decreased the maximum pressure developed seven percent. This system might be advantageous when some of the air should be drawn from a point further back than the rest; i.e. the center of the refrigerator car.

An investigation was started on a fan having cycloidal motion. Only one test was made. This was with a four and one half inch blade traveling on a two and one half inch circle with an angle of blade travel twenty degrees above and twenty degrees below the horizontal. The performance of this fan was similar to that of the nine inch oscillating vane-type at twenty degrees, using the same channel height, but with a blade efficiency of fifteen percent. Further investigation on this type of fan is recommended. By determining the proper blade size and angle of blade travel, for a given circle, and by the use of more than one blade operating on the circle, much greater efficiencies should be obtained, and better performance than that of the oscillating fan should be possible.

CONCLUSION

The oscillating vane-type fan is a low pressure fan. Its advantages are: 1. high air velocity per r.p.m., 2. structural simplicity, 3. adaptability to narrow, rectangular channels. Its disadvantages are: 1. low efficiency, 2. vibration difficulties.

GENERAL FORMULAS AND CHARTS FOR THE CALCULATION

OF THE OSCILLATING VANE-TYPE FAN PERFORMANCE

$$V = \phi\ C\ r\ N$$

$$P = \phi^2\ f\ r^2\ N^2\ \sigma$$

V = Air velocity in feet per minute

P = Pressure change across the fan in mm. water pressure

r = Blade chord in feet

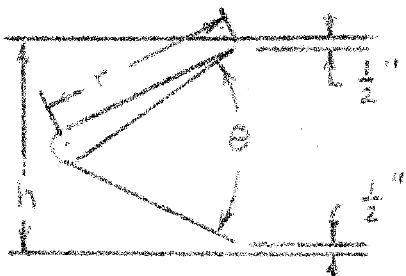
N = Drive r.p.m.

ϕ = Angle coefficient (figure 8)

C = Velocity coefficient (figure 9)

f = Pressure coefficient

σ = Relative air density

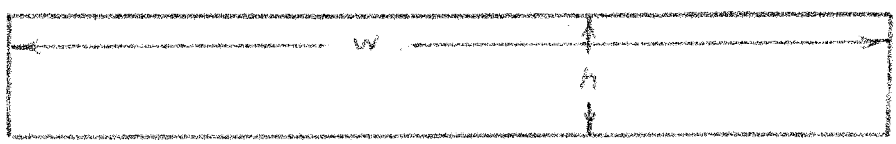


$$h = 2\ r\ \sin\ \theta/2 + 1/12$$

h = Channel height in feet (1/12 is for top clearance of 1/2 inch top and bottom)

$$Q = w\ h\ V$$

w = Channel width in feet



θ = Angle of blade travel, equidistant above and below the horizontal center line.

METHOD OF USE OF FORMULAS AND CHARTS

1. Determine the desired value of P for a certain V and σ for the proposed duct system.

2. Determine h by measurement of installation space.

3. $h = 2\, r \sin \Theta / 2 + 1/12$

select any Θ, preferably 20 degrees so that figure 10 may be used. Solve for r (figure 11)

4. Using Θ, find ϕ from figure 8.

5. Calculate $P/\sigma V^2$ and find C from figure 9. (Remember P is in milimeters.)

6. $V = \phi\, C\, r\, N$

Substitute and solve for N

7. $$\frac{V}{V,} = \frac{\phi\, C\, V\, N}{\phi, C, V, N,}$$

Since a given V and P are required,

$$V/V, = 1 = C/C,$$

$$\phi\, r\, N = \phi,\, r,\, N, = \text{constant}$$

but since h is fixed in 2, and from 3:

$$r, = \frac{h - 1/12}{2 \sin \Theta/2}$$

$$\phi\, r\, N = \phi,\, N,\, \frac{(h - 1/12)}{2 \sin \Theta/2} = \text{constant}$$

8. Substituting in the last equation the values determined in the first six steps for ϕ, r, N, and h, different arrangements of Θ, N, and r may be determined. The best combination must be chosen from mechanical considerations. Figure 10 may be used for this step.

EXAMPLE PROBLEM

On actual test it was found that the pressure developed by a circular fan in a refrigerator car, filled with crates of oranges and sealed, was one half inch water pressure at an air velocity of 1500 feet per minute. The allowable width of the channel was fifteen inches. (The velocity is the average velocity of the air in a rectangle fifteen inches across and the width of the car.) To find the values of Θ, N, and r to duplicate this performance.

Two parallel acting blades will be used in this case.

$h = 15/2(12) = 0.625$ feet

$r = (0.625 - 0.0833)/2 \sin \Theta/2 = 1.555$ feet

taking $\Theta = 20$ degrees, $\phi = 1$

$P/V^2 = 12.7/ (1500)^2 = 5.65 (10^{-6})$

$C = 0.75$

$N = 1500/1 (.75)(1.555) = 1288$ r.p.m.

$\phi\ r\ N = 2000$

From figure 10 we have a choice of the following combinations:

Θ	20	25	30	35	40	degrees
r	1.555	1.263	1.050	0.916	0.795	feet
N	1288	1370	1525	1680	1825	r.p.m.

TEMPERATURE-RELATIVE AIR DENSITY TABLE

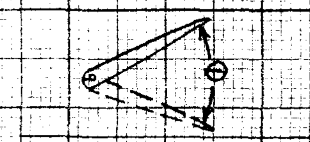
Atmospheric pressure = 29.921 inches water pressure

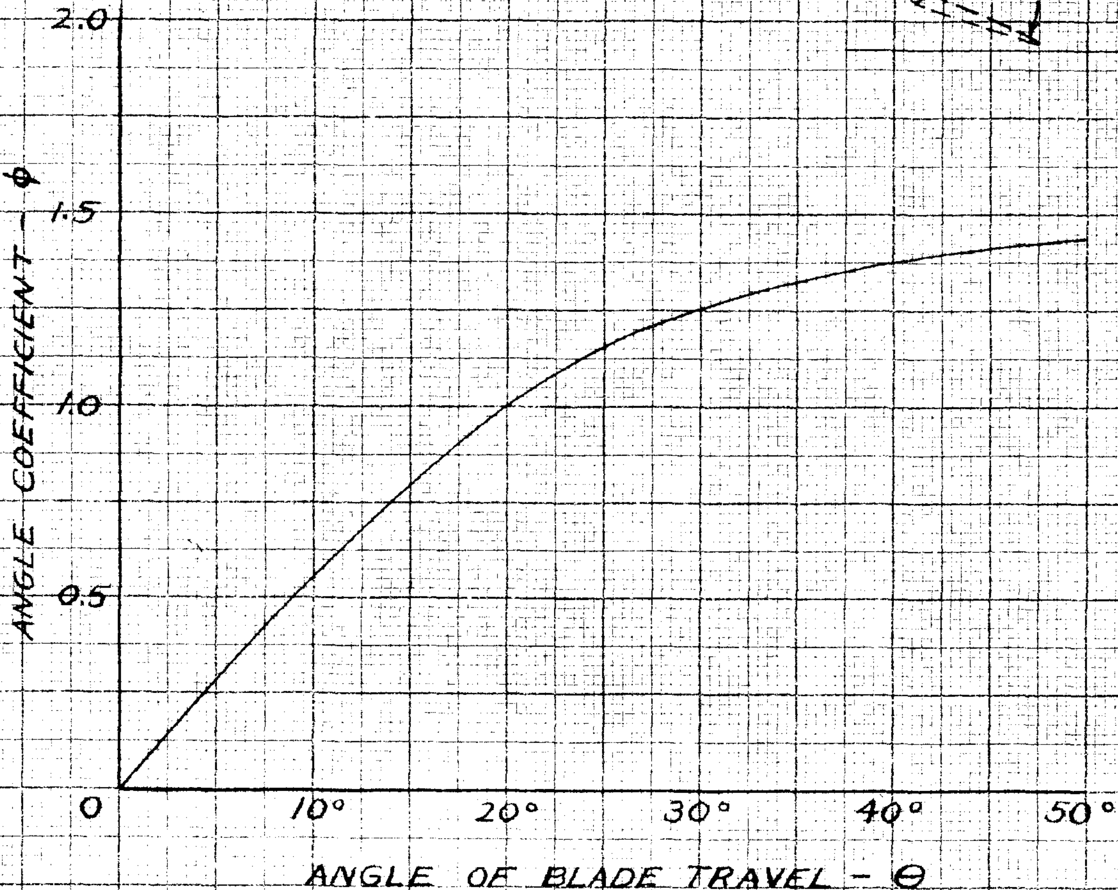
Temperature = 70 degrees F, $\sigma = 1$.

Temperature Degrees F	Relative air density σ
10	1.128
20	1.104
30	1.082
40	1.060
50	1.039
60	1.019
70	1.000
80	0.982
90	0.964
100	0.946

BLADE ANGLE COEFFICIENT vs. ANGLE OF BLADE TRAVEL

$$V = \phi C r N$$
$$P = \phi^2 f r^2 N^2 \sigma$$





Θ	ϕ
20°	1.00
25°	1.16
30°	1.25
35°	1.32
40°	1.38

FIGURE 8

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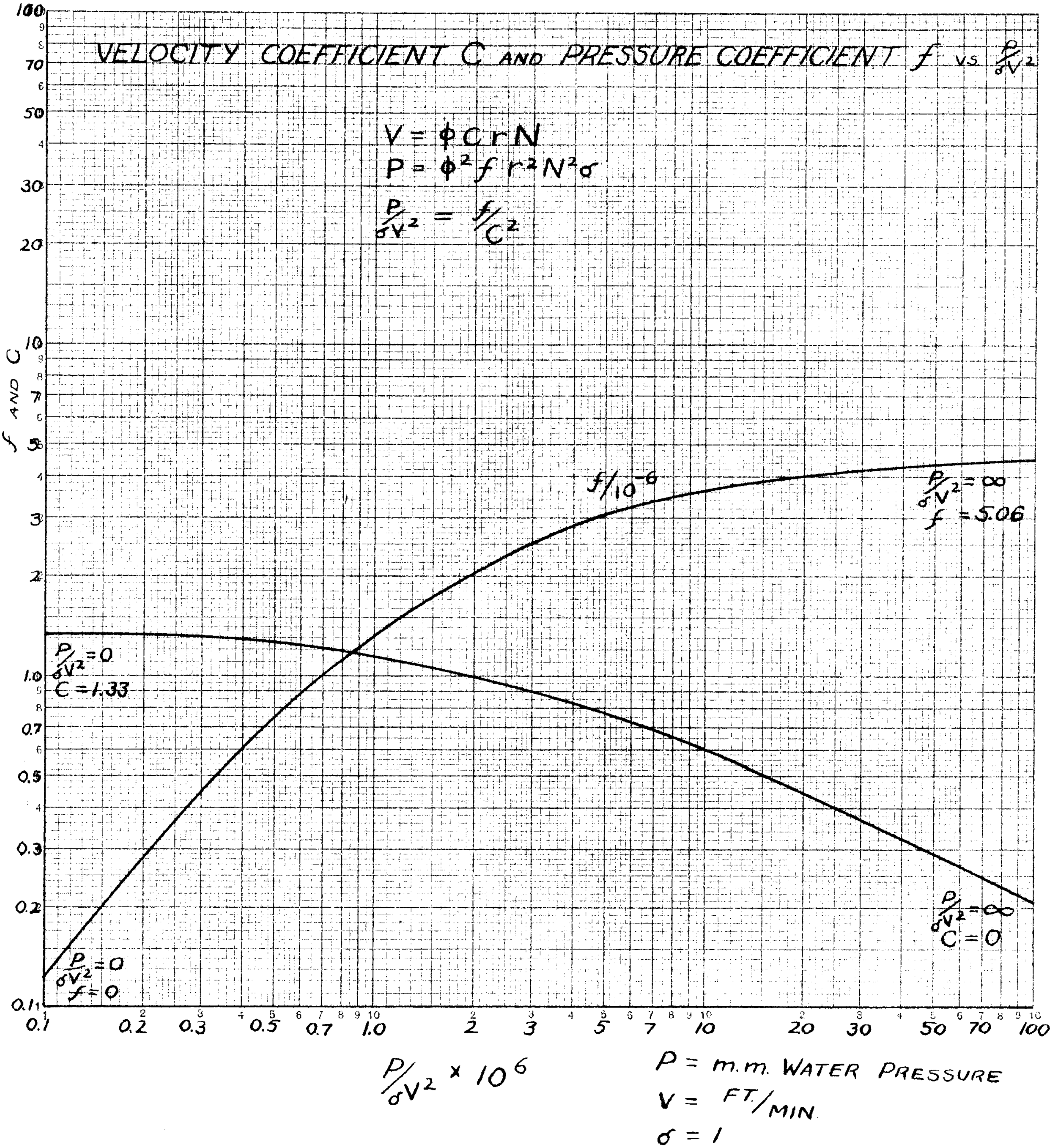

FIGURE 9

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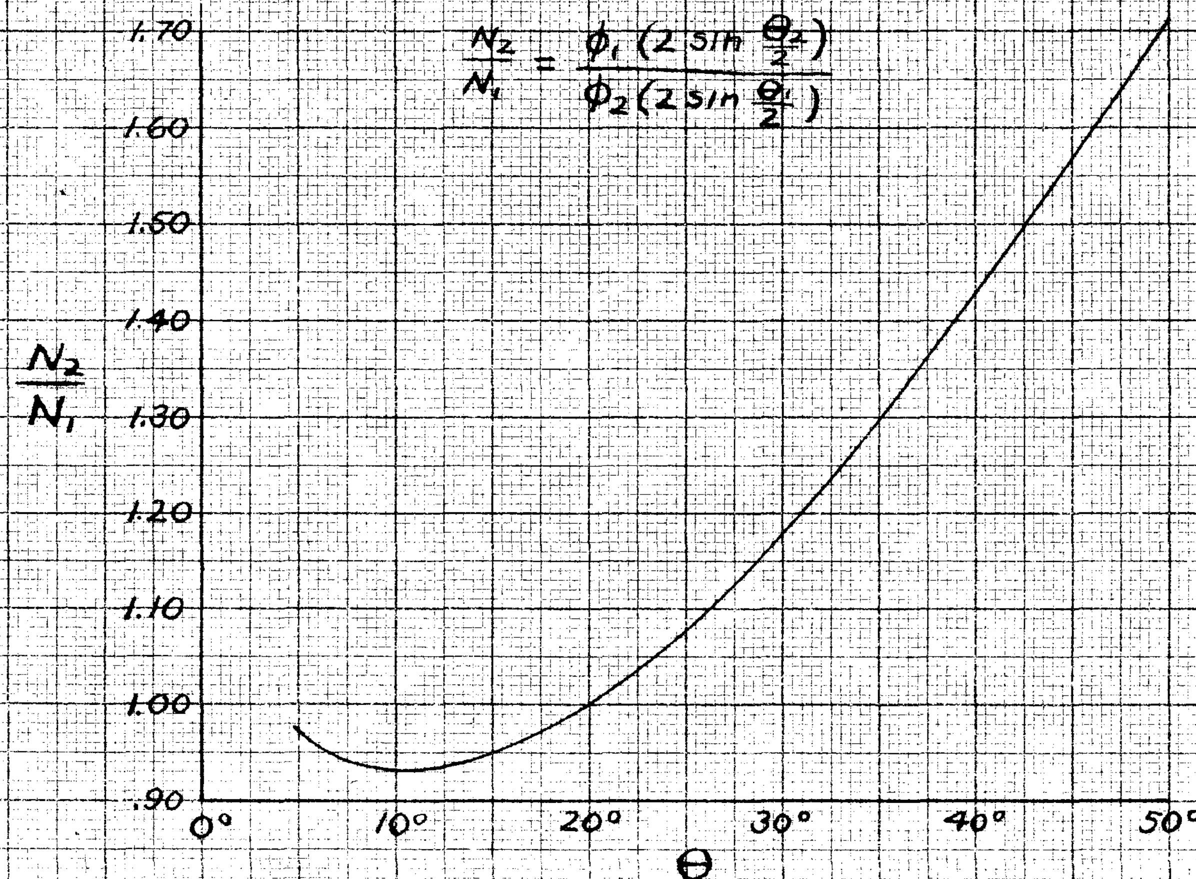
5/20/35

RATIO OF R.P.M. AT ANY θ TO R.P.M. AT $\theta = 20°$ vs. θ

THIS CURVE CAN BE USED ONLY FOR COMPARING DIFFERENT BLADE ANGLES IN A CHANNEL WITH A FIXED HEIGHT 'h', AND TO GIVE THE SAME VELOCITY, QUANTITY, AND PRESSURE AS THE $\theta = 20°$ CASE. (SEE FIGURE 11 FOR CHANGE OF 'r' WITH θ.)

N_1 = R.P.M. FOR $\theta = 20°$ CASE

$$\frac{N_2}{N_1} = \frac{\phi_1 \left(2 \sin \frac{\theta_2}{2}\right)}{\phi_2 \left(2 \sin \frac{\theta_1}{2}\right)}$$



P, V, h = CONSTANT

$$N_2 = \frac{N_2}{N_1} N_1 \sim \theta$$

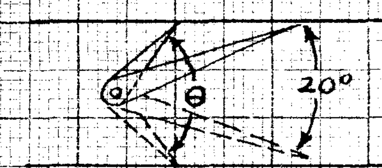


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FIGURE 10

BLADE CHORD vs. CHANNEL HEIGHT FOR DIFFERENT ANGLES θ

BLADE CLEARANCE $\frac{1}{2}$ INCH TOP AND BOTTOM

$$h = 2r \sin \frac{\theta}{2} + 1$$

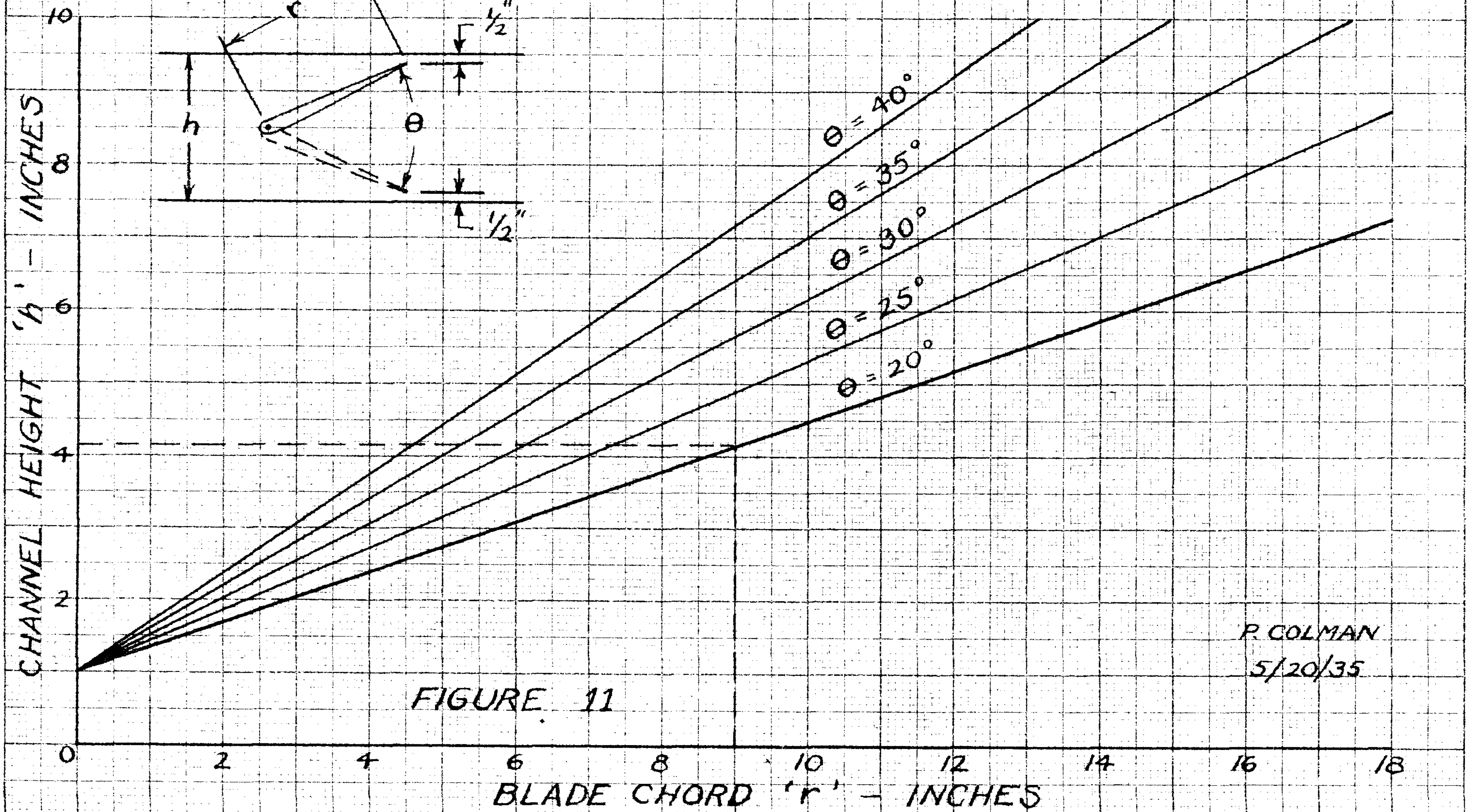

FIGURE 11

APPENDIX A

When the investigation was completed, a report giving the estimated performance of an oscillating vane type fan in a refrigerator car was prepared. The following comprises the performance charts and explanations submitted to the Santa Fe Railroad Co., May 29, 1935.

The following charts apply only to two specialized cases: Case 1 - For a fully loaded refrigerator car; Case 2 - for an empty refrigerator car.. The two cases covered are the two conditions of maximum and minimum restriction to air flow in the car. Case 1 is based on tests made in a fully loaded refrigerator car. Case 2 is similar to a fan operating in a room, and is based on such tests.

The arrangement considered best for the refrigerator car installation, as shown by the structural blueprints, is a blade of 9 inch chord length, 4 and 3/16 inch channel height, channel width of 8 feet, and blade travel of 20 degrees. This is indicated by the dotted line in figure 11, and the heavy r lines in figures 12 and 13.

Figures 12, 13, and 14 apply for the condition, air temperature equals 70 degrees F, atmospheric pressure of 29.921 inches of water. For the condition of air temperature for Case 1 equals 32 degrees, the pressure is increased 7 percent.

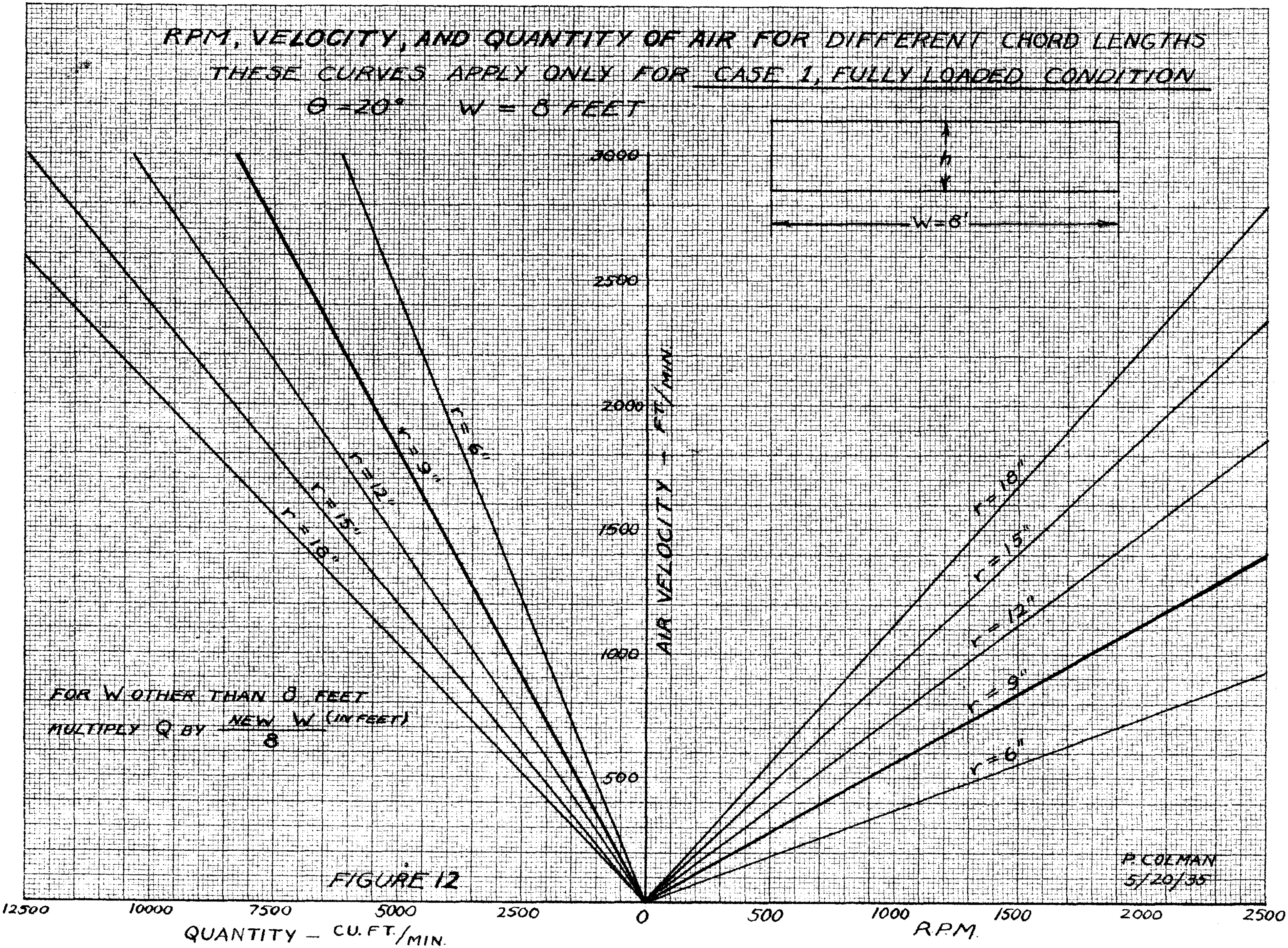
RPM, VELOCITY, AND QUANTITY OF AIR FOR DIFFERENT CHORD LENGTHS
THESE CURVES APPLY ONLY FOR CASE 1, FULLY LOADED CONDITION
θ = 20°   W = 8 FEET
h
W = 8'
AIR VELOCITY – FT./MIN.
3000
2500
2000
1500
1000
500
r = 6"
r = 9"
r = 12"
r = 15"
r = 18"
FOR W OTHER THAN 8 FEET
MULTIPLY Q BY NEW W (IN FEET) / 8
12500
10000
7500
5000
2500
0
500
1000
1500
2000
2500
QUANTITY – CU.FT./MIN.
R.P.M.
P. COLMAN
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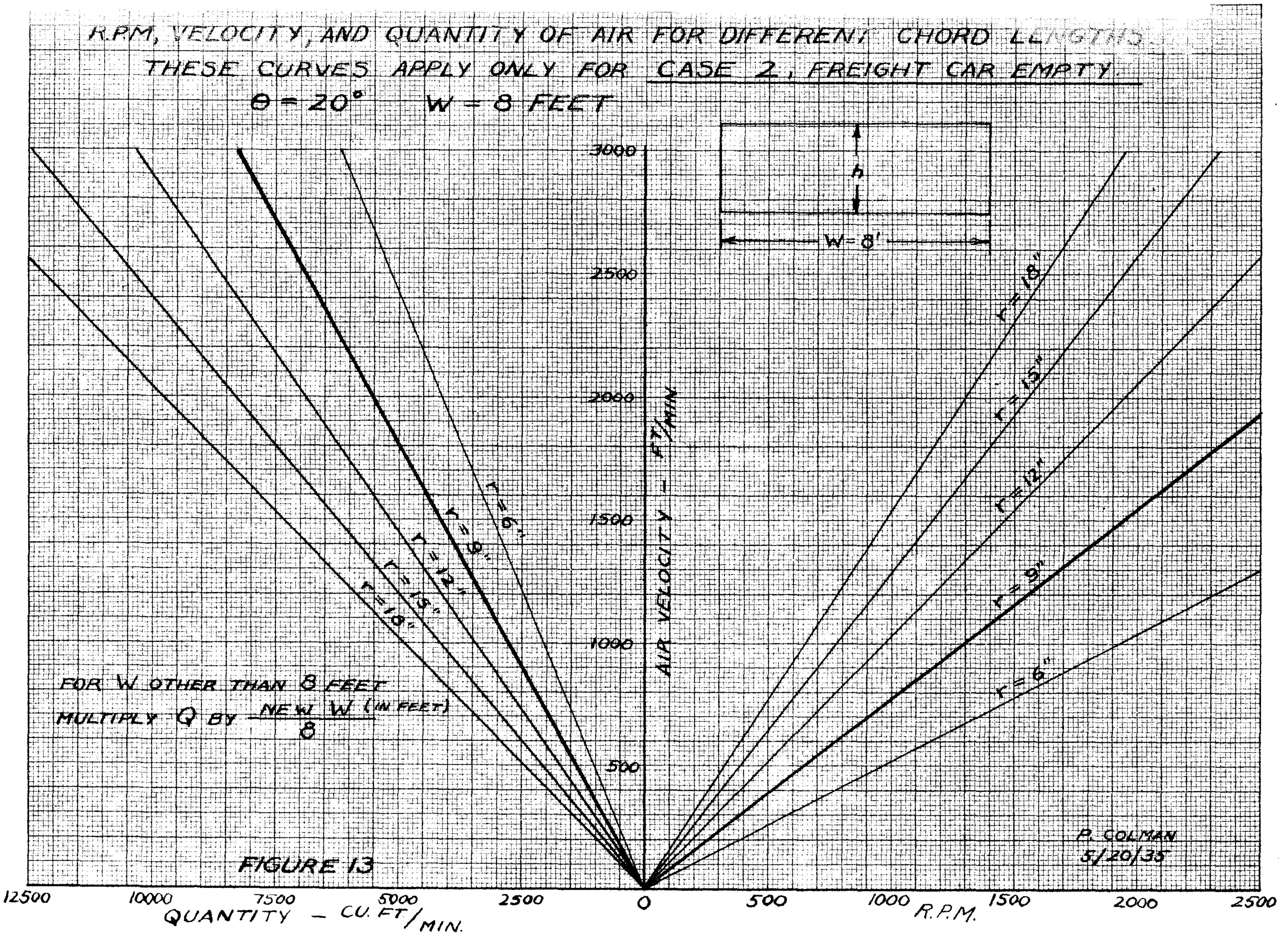
FIGURE 12



R.P.M., VELOCITY, AND QUANTITY OF AIR FOR DIFFERENT CHORD LENGTHS
THESE CURVES APPLY ONLY FOR CASE 2, FREIGHT CAR EMPTY.
θ = 20°   W = 8 FEET
h
W = 8'
r = 6"
r = 9"
r = 12"
r = 15"
r = 18"
AIR VELOCITY – FT/MIN
3000
2500
2000
1500
1000
500
FOR W OTHER THAN 8 FEET
MULTIPLY Q BY NEW W (IN FEET) / 8
FIGURE 13
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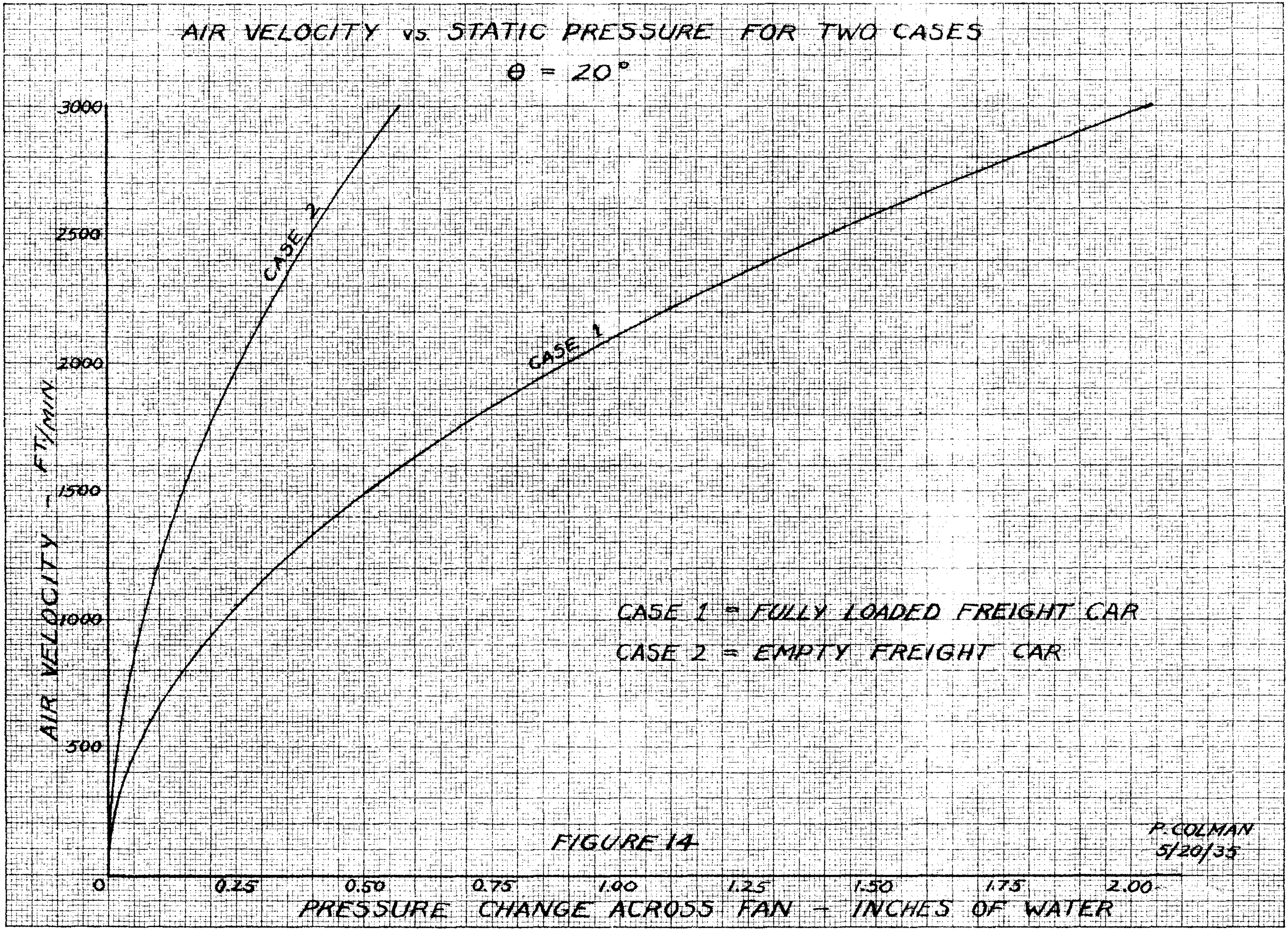
12500
10000
7500
5000
2500
0
500
1000
1500
2000
2500
QUANTITY – CU. FT/MIN.
R.P.M.



AIR VELOCITY vs. STATIC PRESSURE FOR TWO CASES
θ = 20°
AIR VELOCITY - FT/MIN
3000
2500
2000
1500
1000
500
0
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
PRESSURE CHANGE ACROSS FAN - INCHES OF WATER
CASE 1
CASE 2
CASE 1 = FULLY LOADED FREIGHT CAR
CASE 2 = EMPTY FREIGHT CAR
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FIGURE 14

APPENDIX B

The Santa Fe Railroad Co. installed two fans of this type, one in each end of a refrigerator car. They made a series of tests, with the fan operating so as to reverse, and with the fan operating so as to increase, the circulation in the car over the natural circulation. A short summary of the results are given here.

The measurement of the air velocity in the car closely checked the predicted results given in Appendix A.

The only disadvantage of this fan was found to be the structural difficulties.

RESULTS OF REFRIGERATOR CAR TESTS

General Discussion:

Test No. 1. Reversing the natural circulation of the car shows the advantage of keeping the temperature throughout the fruit uniform. Comparing the temperatures observed to the average temperatures obtained in Test No. 2, with reversed airflow direction, shows that the averaged temperatures in Test No. 2 are slightly lower. Therefore the overall efficiency of the arrangement used in Test No. 2 seems somewhat better. It is not possible to say how the two arrangements compare at higher speeds.

Test No. 2. The fruit temperatures and total ice meltage are considered in the following pages. The ice meltage per hour was also considered especially to check how far the ice meltage follows the variable car speed corresponding to the schedule. No effect due to outside temperature changes is noticeable.

General Results of the 48-Hour Test

	Fan Car	Std. Car
Ice melted in 48 hours	5750 lbs.	5100 lbs.
Average temperature at start	60.1°	59.5°
Average temperature after 48 hours	48.45°	50.6°
Temperature drop in 48 hours	11.65°	8.9°

These figures show: (1) The increase in cooling due to the fan $= \frac{11.65}{8.9} - 1 = \underline{31\%}$; (2) the increase in ice meltage with the fan $= \frac{5750}{5100} - 1 = \underline{13\%}$.

Since the heat taken from the fruit = constant x temperature change x pounds of fruit, we get a 31% decrease in temperature, or could have a 31% increase in load with the same average temperature as that of the present load in the standard car. 31% of 36,000 pounds equals 11,100 extra pounds of fruit could be carried.

Comparison of Cooling Characteristics

An important difference between the cooling characteristics of the two cars is revealed by comparing the cooling effects for 12 hour periods.

Average Temperatures

	Temperature		Temperature Reduction		
Hour	Fan Car	Std. Car	Fan Car	Std. Car	Fan Car / Std. Car
0	60.10	59.50			
			6.20	5.15	120%
12	53.90	54.35			
			2.45	2.03	120%
24	51.45	52.30			
			1.40	1.10	127%
36	50.05	51.20			
			1.60	0.60	267%
48	48.45	50.60			

The temperature of the bottom fruit is satisfactorily low also in the case of the standard car, and it is important to consider the differences as far as the top fruit is concerned.

Top Fruit Temperatures

Hour	Temperatures: Fan Car	Temperatures: Std. Car	Temperature Reduction: Fan Car	Temperature Reduction: Std. Car	Fan Car / Std. Car
0	64.0°	64.0°			
			4°	2.1°	190%
12	60.0°	61.9°			
			2.3°	1.9°	121%
24	57.7°	60.0°			
			2.3°	1.2°	192%
36	55.4°	58.8°			
			2.7°	0.8°	338%
48	52.7°	58.0°			

From this it can be seen that at the end of 48 hours the standard car is already reaching its equilibrium condition while the top fruit of the fan car is still being cooled at the rate of 2.4 degrees in 12 hours (approximately), or of 0.2 degrees per hour. There is no indication of a decrease in the cooling rate. There is an 8.5 degree difference in temperature between the top and bottom fruit after the 48 hours. At the rate of 0.2 degrees reduction per hour, the top fruit temperature will reach that of the bottom fruit in 42.5 more hours, while the standard car will always have a large temperature difference.

If the average speed of the fan would be increased, the time elapsed before the top fruit would reach the bottom fruit temperature would be proportionally reduced.

Insulation Losses

From Test No. 4 the heat loss through the walls may be determined. The temperatures in the two empty cars of this test were approximately the same.

	Fan Car	Std. Car
Ice meltage in 48 hours with load	5750 lbs.	5100 lbs.
Ice meltage in 48 hours, car empty	2100 lbs.	1870 lbs.
Percentage of ice melted to cool the sides of the car	36.4%	36.6%

Therefore, it can be said that the insulation losses are unaffected by the use of the fan. The actual pounds of ice melted to cool the load will then be 64% of the total pounds melted.

For 48 hours	Fan Car	Std. Car
Ice melted to cool load	3680 lbs.	3260 lbs.
Ice melted per degree drop in average temperature	316%	366%

The discrepancy in pounds of ice per degree drop is in the order of ± 10%, and must be due to other factors entering in the cooling rate. But this shows that such factors favor the fan equipped car.

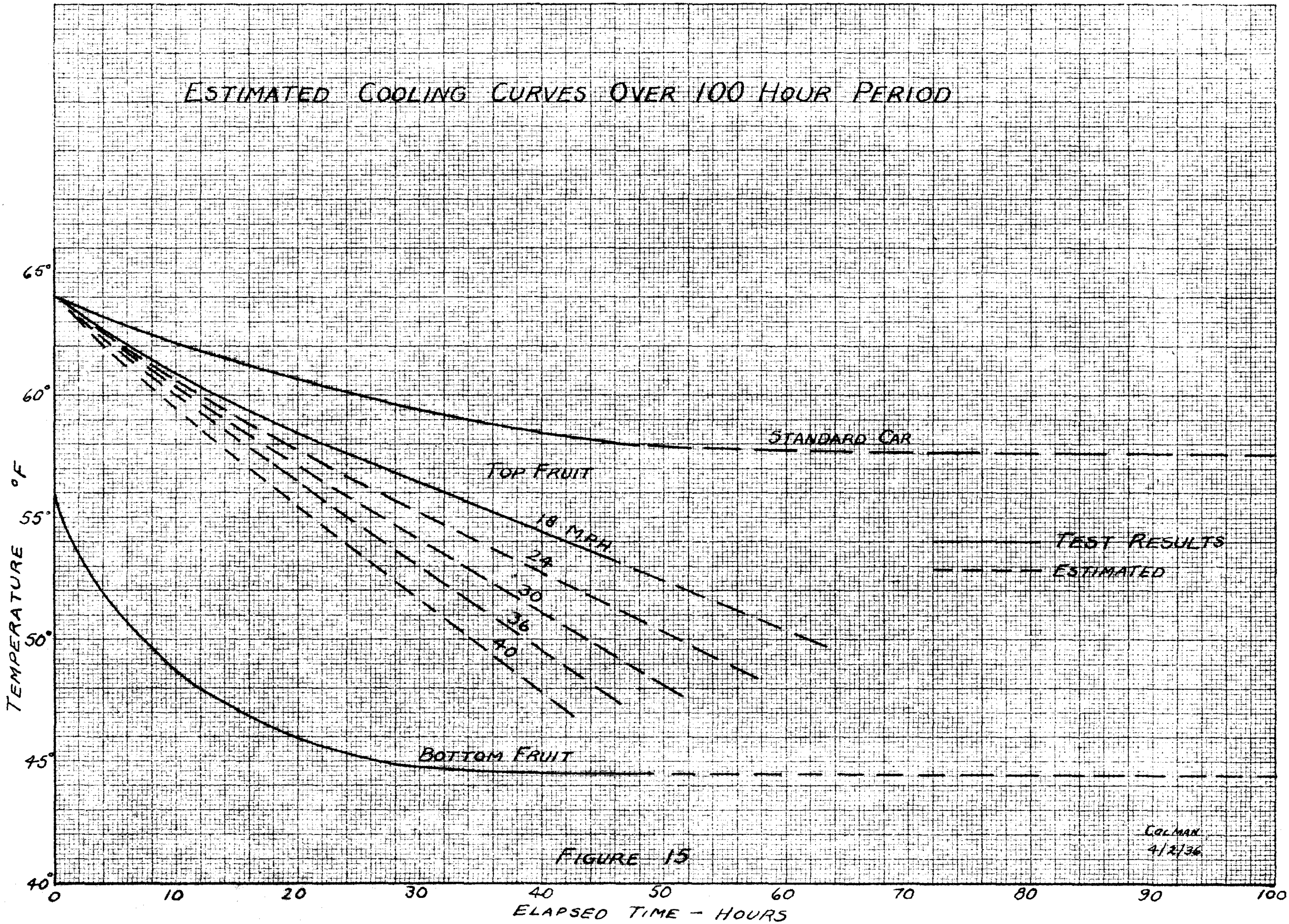
Discussion of Extrapolation to Higher Average Fan Speeds

The method used in determining the top fruit cooling curve for an average fan speed of 24 m.p.h. was as follows: The slope of the present top fruit temperature curve between the 34th and 40th hour, where the average car speed was 24 m.p.h., was used as the basis. The slope is known to be greater the first twelve hours and taper off as the elapsed time increases. This change in slope with time was estimated from the present curve and the estimated 24 m.p.h. curve drawn. As checks we have:

1. Between the hours 2 and 4.5, the average speed was 37 m.p.h., and the slope of the present curve is greater than that of the estimated curve.
2. Between the hours 5.7 and 8.5, the average speed is 27 m.p.h., and the slope of the present curve is steeper than that of the estimated curve.
3. Between the hours 27 and 32, the average speed is 24 m.p.h., and the slope of the present curve is the same as that of the estimated curve.

As a further check on this extrapolation, the top fruit temperature after 48 hours was increased by the percent increase of fan speed and the results fall on the estimated curve.

To bring the top fruit temperature to that of the bottom fruit in 48 hours, figuring by this same method, the fan must be run at an average speed corresponding to a car speed of 42 m.p.h. in the present car, or of 585 oscillations per minute using a fan identical to that used in these tests.



ESTIMATED COOLING CURVES OVER 100 HOUR PERIOD
STANDARD CAR
TOP FRUIT
18 MPH
24
30
36
40
BOTTOM FRUIT
TEST RESULTS
ESTIMATED
TEMPERATURE °F
65°
60°
55°
50°
45°
40°
0
10
20
30
40
50
60
70
80
90
100
ELAPSED TIME - HOURS
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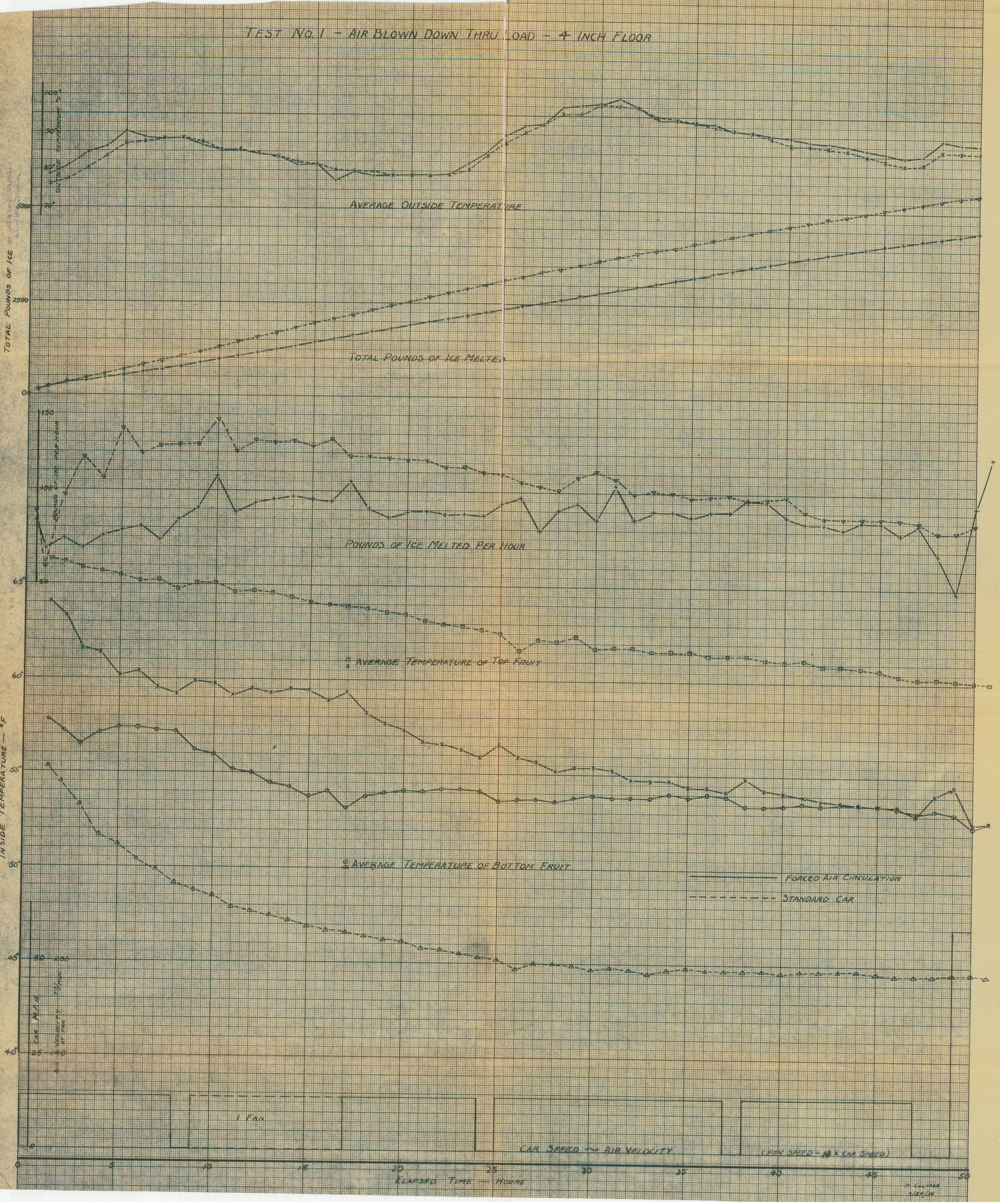
4/2/36

FIGURE 15



TEST NO. 1 - AIR BLOWN DOWN THRU LOAD - 4 INCH FLOOR
OUTSIDE TEMPERATURE °F
AVERAGE OUTSIDE TEMPERATURE
TOTAL POUNDS OF ICE
TOTAL POUNDS OF ICE MELTED
POUNDS OF ICE PER HOUR
POUNDS OF ICE MELTED PER HOUR
AVERAGE TEMPERATURE OF TOP FRUIT
INSIDE TEMPERATURE — °F
AVERAGE TEMPERATURE OF BOTTOM FRUIT
FORCED AIR CIRCULATION
STANDARD CAR
CAR M.P.H.
AIR VELOCITY AT FAN FT/MIN.
1 FAN
CAR SPEED ~ AIR VELOCITY
(FAN SPEED = 14 × CAR SPEED)
ELAPSED TIME — HOURS
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3/24/36

TEST NO 2 — AIR BLOWN UP THRU LOAD — 4 INCH FLOOR