THE DESIGN OF A LOW SPEED WIND TUNNEL
FOR MICROMETEOROLOGY RESEARCH

By

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PREFACE

Oklahoma State University has received a grant from the National Science Foundation for the design and construction of a low speed wind tunnel for micrometeorology research. The knowledge of the flow and characteristics of the air near the earth's surface is important to many of the sciences. The purpose of this paper is to present the design of a wind tunnel in which the natural air flow may be simulated.

The author is indebted to Professor L. J. Fila for his willing advice and counsel. Further indebtness is due Dr. Gordon Nelson for his aid and advice throughout the design. Also a word of commendation is due Jack Fryrear and Jim W. Hale for their excellent preparation of the detail drawings.

The author wishes to express his appreciation to Dr. J. H. Boggs for his helpful suggestions and criticisms in the preparation and writing of this paper.

The author would be remiss if he failed to mention the sacrifices made during the writing of this paper by his wife, Thelma, and his son, Neil, to whom this paper is dedicated.

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NOMENCLATURE

A	Area, $ft.^2$
B	Expansion ratio, dimensionless
C	Contraction ratio, dimensionless
D	Height, ft.
F	Temperature, degrees Farenheit
G	Roughness coefficient, $sec.^{-\frac{1}{5}}$ $ft.^{\frac{5}{2}}$
H	Height, in.
K_o	Pressure loss coefficient, dimensionless
L_c	Characteristic length, ft.
L	Length, ft.
N	Theoretical energy conversion in a diffuser, per cent
N_o	Overall diffuser efficiency, per cent
N_f	Fan Efficiency, per cent
q	Dynamic pressure, lb. per ft.
R_n	Reynolds number, dimensionless
V	Volume, ft. per minute
$\bar{V}$	Velocity, ft. per sec.
$\bar{v}$	Velocity variation from the mean velocity, ft. per sec.
X	Distance from the intake section entrance, ft.

<u>Greek Letters</u>

ρ	Rho, Density, lb. $sec.^2$ per $ft.^4$
Δ	Delta, Change of a value in general

μ	Mu, Dynamic viscosity, lb. sec. per ft.^{2}
δ	Delta, Thickness of the boundary layer, ft.
λ	Lambda, Skin friction coefficient, dimensionless
θ	Theta, Diffusion angle, degrees

Abbreviations

ft	Feet
sec	Seconds
psia	Pounds per square inch absolute
in.	Inch
CFM	Cubic feet per minute
log	Logarithm to the base 10
Ke	Kinetic energy, foot pounds
E.R.	Energy ratio
BHP	Brake horsepower
AHP	Air horsepower
fps	Feet per second
MPH	Miles per hour
DFPA	Douglas Fir Plywood Association
PT	Point of tangency

Subscripts

i	Intake section
D	Diffuser
f	Fan
S	Screen
1	Intake section inlet

2	Intake section outlet
o	Test section
t	Transition section

CHAPTER I

INTRODUCTION

The flow and characteristics of air near the earth's surface are quite unlike those found at a point only a few feet above the surface. The knowledge of the characteristics of this flow and properties of the air within this region are important to many of the sciences. If the air within this region is to be studied, a means of controlling the characterisitics of the air and the velocity gradient, in order to simulate the natural air flow is needed. To do this a special type of wind tunnel is required other than the kind found in general aerodynamic usage. An aerodynamic wind tunnel is unsuited because the velocity distribution throughout the test section is nearly uniform and the test section is generally short.

Sutton (1) states that the air flow over the earth's surface is always turbulent. In order to simulate this flow, a tunnel of sufficient length to permit the growth of the turbulent boundary layer is required. O'Neill (2) gives the following expression for the standard wind gradient:

$$\frac{\overline{V}_D}{\overline{V}_{33}} = 0.656 \log (D+16) - 0.109 \qquad (1\text{-}1)$$

where:

$\overline{V}_D$ = velocity at any height above the earth's surface, feet per second

$\overline{V}_{33}$ = velocity at a point 33 feet above the earth's surface, feet per second

D = height, feet

This gradient is shown in Fig. 1. The height scale is such that one inch as shown in Fig. 1 is equivalent to one-hundred inches above the earth's surface. The velocity at a height of 33 feet was assumed to be 50.5 (fps).

If this gradient can by obtained, then it can be expected that experimental tests in regard to wind effects, will closely approximate the conditions prevailing near the earth's surface. There are no tunnel facilities to meet this and other comparable requirements, insofar as known, at any educational institution in the southwest. In the absence of suitable facilities within this geographical area, a low speed wind tunnel to be used for micrometeorology research was designed and built.

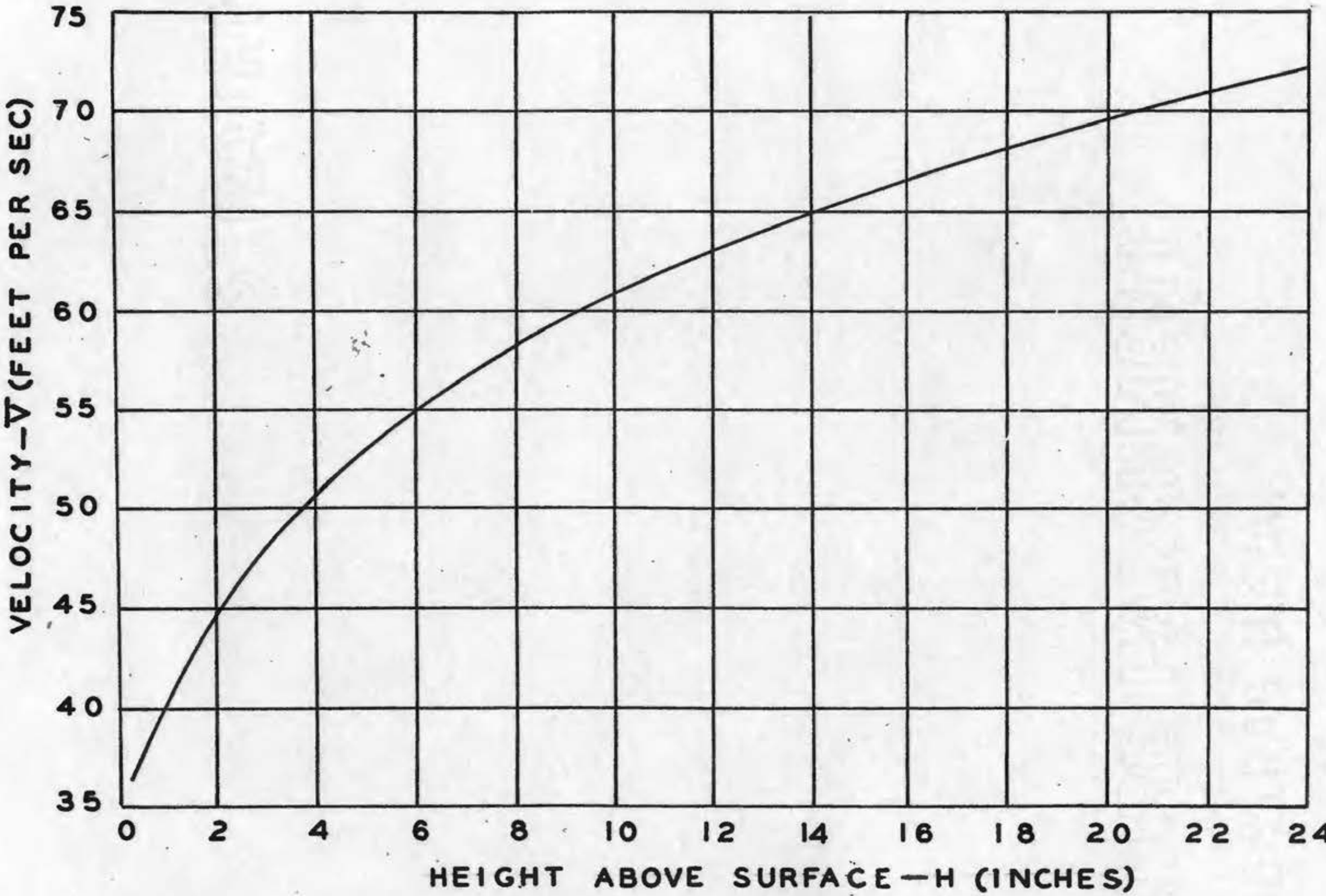

Figure 1. Standard Velocity Gradient Near the Earth's Surface

CHAPTER II

INITIAL DESIGN CONSIDERATIONS

In order that the tunnel might be suitable for as wide a range of research projects as possible, the functional requirements for micrometeorological research and related areas were determined. Conferences were held with researchers on the staff of Oklahoma State University who are doing research wherein wind near the ground is a pertinent factor. These areas of research include: agricultural engineering, poultry science, agronomy, horticulture, landscape design, entomology and animal husbandry. Other areas for research would include heat transfer studies and industrial applications.

Specific investigations in which a low-speed wind tunnel of special design will be used include:

1. Wind force effects on structures.
2. Surface cooling by wind of roof coverings exposed to solar radiation.
3. Wind cooling and ventilation of livestock and poultry production buildings.
4. Drying of harvested grain with wind-induced air circulation.
5. Control of evaporation from reservoir surfaces.
6. Control of evaporation from surfaces of irrigated lands.
7. Wind-conveyance of particles and pathological organisms.
8. Influence of wind on distribution of defoliants, insecticides,and herbicides, and on the design of application

equipment.

9. Convective air cooling of agricultural products.
10. Vibratory loads due to wind on surface coverings for roofs.
11. Effect of wind on surface temperatures and sub-surface temperature gradients of soils and impounded water.
12. Effect of wind on evapo-transpiration requirements of plants.
13. Heating and cooling requirements for residences as influenced by wind effects.

A. FUNCTIONAL DESIGN REQUIREMENTS

As a result of the functional design considerations, it was decided that the tunnel should:

1. Be large enough for, and have easy access for bulky objects.
2. Be long enough to permit the turbulent boundary layer to reach a depth of 18".
3. Have a drive unit capable of a wide range of speed.
4. Be able to operate for long periods of time.
5. Have test sections of variable height.
6. Have temperature and humidity controls.

B. ANALYSIS OF THE FUNCTIONAL REQUIREMENTS

1. Tunnel Length

The study of motion within the boundary layer of a flat surface may be regarded as the idealized problem of the natural wind flow over a small portion of the earth. For a flow parallel to a smooth surface a laminar boundary layer is first formed starting at the leading edge, but for sufficiently large distances downstream, or for sufficiently high velocities, the boundary layer becomes turbulent. Thus on proceeding downstream from the leading edge, a laminar boundary layer is first formed, then a transition region, and finally a fully developed turbulent boundary layer (see Fig. 2). The transition region is quickly reached and occurs, according to Binder (3), when the Reynolds number becomes equal to 2,000. The Reynolds number is defined as:

$$R_n = \frac{\rho V L_c}{\mu} \qquad (2\text{-}1)$$

where:

ρ = density, pound second squared feet^{-4}

L_c = characteristic length, feet

μ = dynamic viscosity, pound second per foot squared

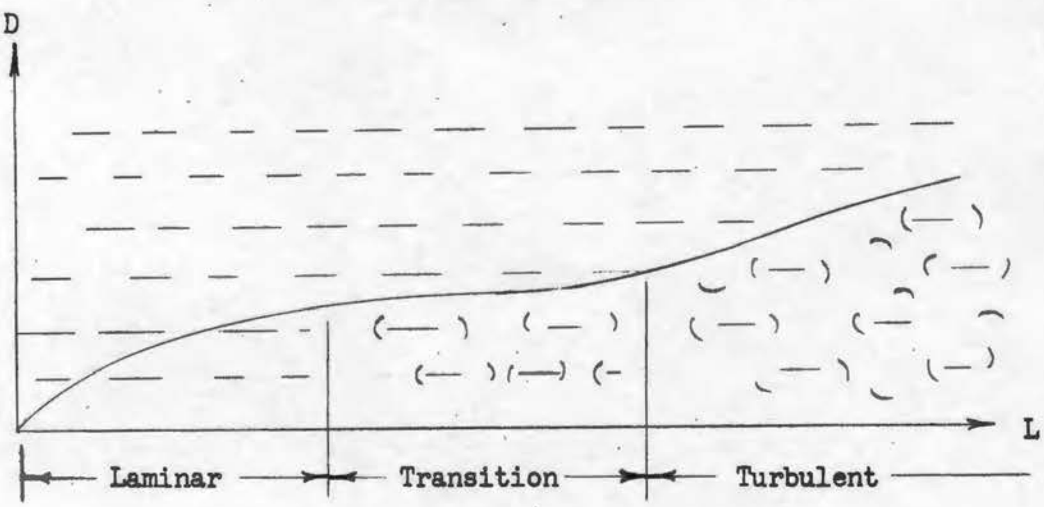

Fig. 2--The Turbulent Boundary Layer

In natural wind flow the leading edge is always an infinite distance upstream from the point of observation; thus it may be assumed that the trubulent boundary is always fully developed. (1).

Sutton (1) gives the following equation for the depth of the turbulent boundary layer for flow over a smooth surface as a function of the distance from the leading edge:

$$\delta = 0.366 \ L \left(\frac{\mu}{\rho V L} \right)^{\frac{1}{5}} \qquad (2\text{-}2)$$

where:

δ = thickness of the boundary layer, feet

In all subsequent calculations, standard sea level conditions of 14.7 psia and 69° F are assumed. For these conditions $\frac{\rho}{\mu} = 6380$ sec. ft.$^{-2}$ For an assumed mean velocity of 35 fps and a boundary depth requirement of 18 inches, the tunnel length by Eq. (2-2) is 153 feet. Because of space limitations the tunnel test sections were restricted to a length of 50 feet. This length gives, by Eq. (2-2), a turbulent boundary layer depth of 7.8 inches.

The boundary layer depth may be increased by artificially roughening the surface over which the flow occurs. Zingg and Chepil (4) performed experiments in which the boundary layer depth was determined over an artificially roughened floor. The floor was covered with gravel ranging in diameter from one-eight to one-quarter inches in diameter. When the floor length was 48 feet, and the mean velocity 38.2 fps, the turbulent boundary layer was found to have a depth of 10.5 inches.

In order to replace the constant 0.336 Eq. (2-2), which is for flow over a smooth surface, with one which accounts for a roughened surface, the results of Zingg and Chepil are substituted into Eq. (2-2) giving a value of 'G' equal to 0.475 seconds$^{-\frac{1}{5}}$ feet.$^{\frac{5}{2}}$ The constant 'G' is the roughness coefficient. The turbulent boundary layer depth for flow over the gravel roughened surface may then be expressed as:

$$\delta = 0.475 \ L \left(\frac{\mu}{\rho V L} \right)^{\frac{1}{5}} \qquad (2\text{-}3)$$

For a length of 50 feet and a mean velocity of 35 fps the boundary layer depth is found to be 11.0 inches.

Summarizing:

Boundary layer depth over a smooth floor $\simeq$ 7.8 inches.

Boundary layer depth over a gravelled floor $\simeq$ 11.0 inches.

2. Temperature and Humidity Control

Temperature and humidity control to the extent of increasing the humidity and decreasing the temperature may be provided for. This may be achieved by providing a short removable test element. Into this space another element may be placed whereby the air may be adiabatically saturated. Cooling of the air by refrigeration was not deemed feasible. The size of the air conditioning unit required would make the cost prohibitive. Maintaining an air temperature in the tunnel higher than atmospheric temperature can be accomplished by heating the laboratory space.

3. Energy Source and Speed Control

As an ample source of electrical power was available, the use of an electric motor with some variable speed type drive was indicated. The use of these units would result in good speed control, and the tunnel could be operated unattended.

4. Tunnel Size and Configuration

A basic test section of 4 X 4 feet in cross section was deemed sufficiently large. If a test were to be conducted which required a larger test section, some of the basic test sections could be removed. They would be replaced by a larger test section and additional diffuser and contraction elements. These additional elements were not designed.

A floor which was adjustable in height would permit the use of a false floor. This would be desirable when tests were to be conducted involving water surfaces and plants. The water or plant containers themselves would not affect the flow pattern.

An adjustable floor would make it possible to provide for a slight divergence of flow. This would permit the maintenance of a uniform maximum velocity throughout the test section length.

CHAPTER III

STATEMENT OF THE DESIGN OBJECTIVES

From preliminary design considerations, the following final objectives were to:

A. Have a test section length of 50 feet.

B. Have a basic test section of 4 X 4 feet in cross-section.

C. Provide a variable depth test section.

D. Assemble the test section from component elements.

E. Attain a velocity of 40 MPH.

F. Use an open type tunnel with provision made for the possible conversion to a closed type tunnel.

G. Use a contraction ratio of six for the intake section.

H. Provide instrumentation for the tunnel to insure accuracy of velocity measurements at low velocities.

I. Provide for the possible increase of the air humidity within the tunnel.

J. Dimension the contraction element to give constant acceleration at discharge.

K. Achieve low turbulence of the airstream at exit from the contraction element.

L. Provide a fan with at least 6 blades with the blades having an adjustable angle of attack.

M. Use a tunnel of the induced flow type.

N. Provide a diffuser for maximum energy recovery.

CHAPTER IV

COMPONENT DESIGN

The tunnel is located in the Agricultural Engineering Laboratory No. 1-A. A floor plan of the of the Laboratory showing the tunnel location is given in Fig. 3. Air flowing from the end of the tunnel is discharged into a dust settling room having a floor area of approximately 350 square feet and an average height of 13 feet. From this room the air may be either exhausted to the atmosphere or returned to the tunnel laboratory for recirculation. If the air is exhausted to the atmosphere, ample openings exist to provide a fresh supply of air.

An induced flow-type tunnel was selected as this type tunnel has several advantages over one in which air is blown in from the intake end. The entering air is undisturbed by the fan, making a honeycomb or other straightener devices unnecessary. One-third of the fan dynamic pressure may be lost in the straightener devices of a forced flow-type tunnel. (4). Bagnold (5) states that there is also the practical advantage that the negative pressure within the tunnel will aid in keeping the access doors tightly closed. This applies also to the sealing of the flange joints between test elements. If seals are found to be necessary, the seals would be located on the outside of the tunnel joints. With the seals located externally, less disturbance of the air flow would result.

Figure 4 is a schematic drawing showing all the component parts. The detail drawings and specifications of all component parts are on file

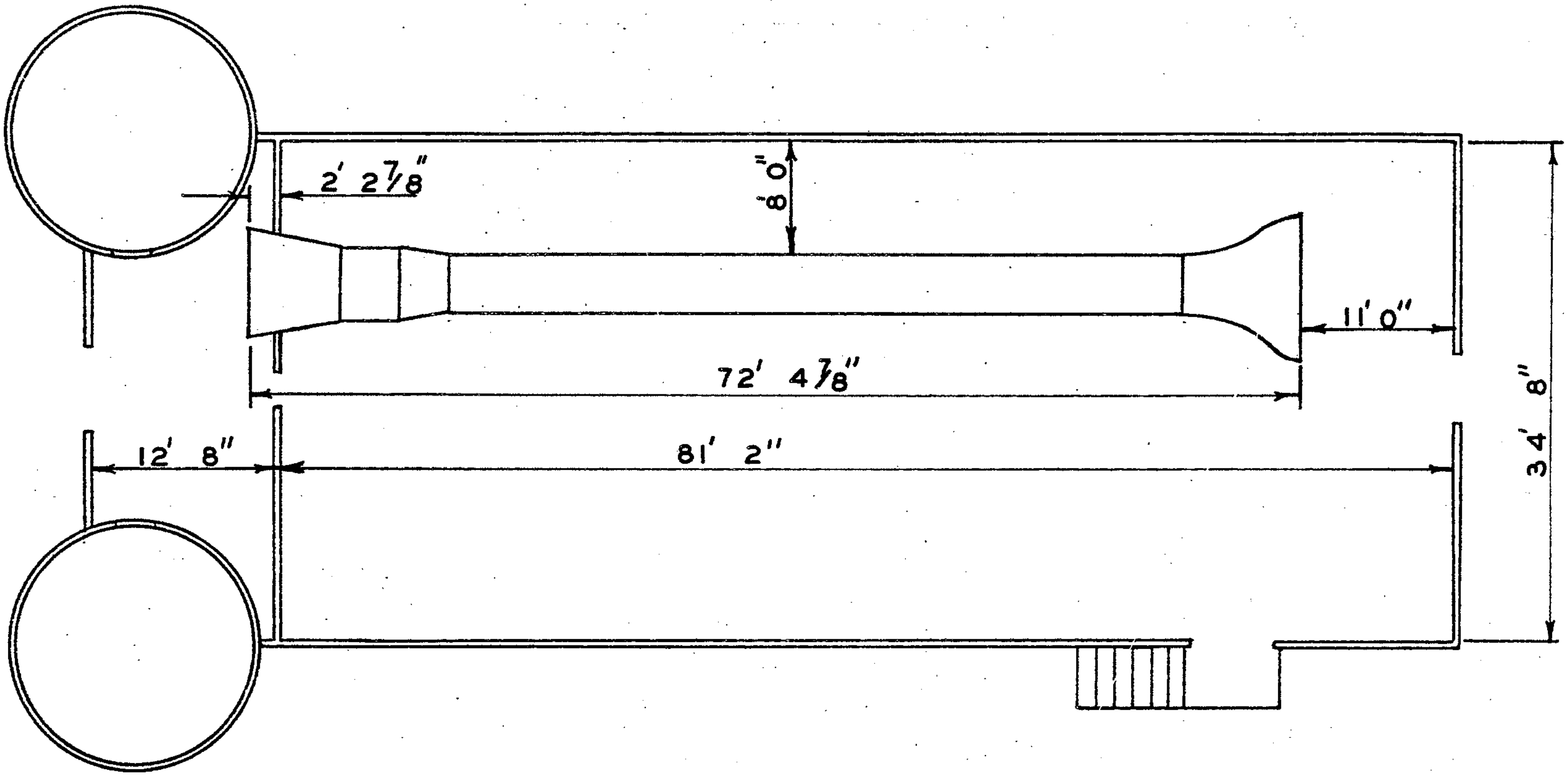

Figure 3. LABORATORY FLOOR PLAN

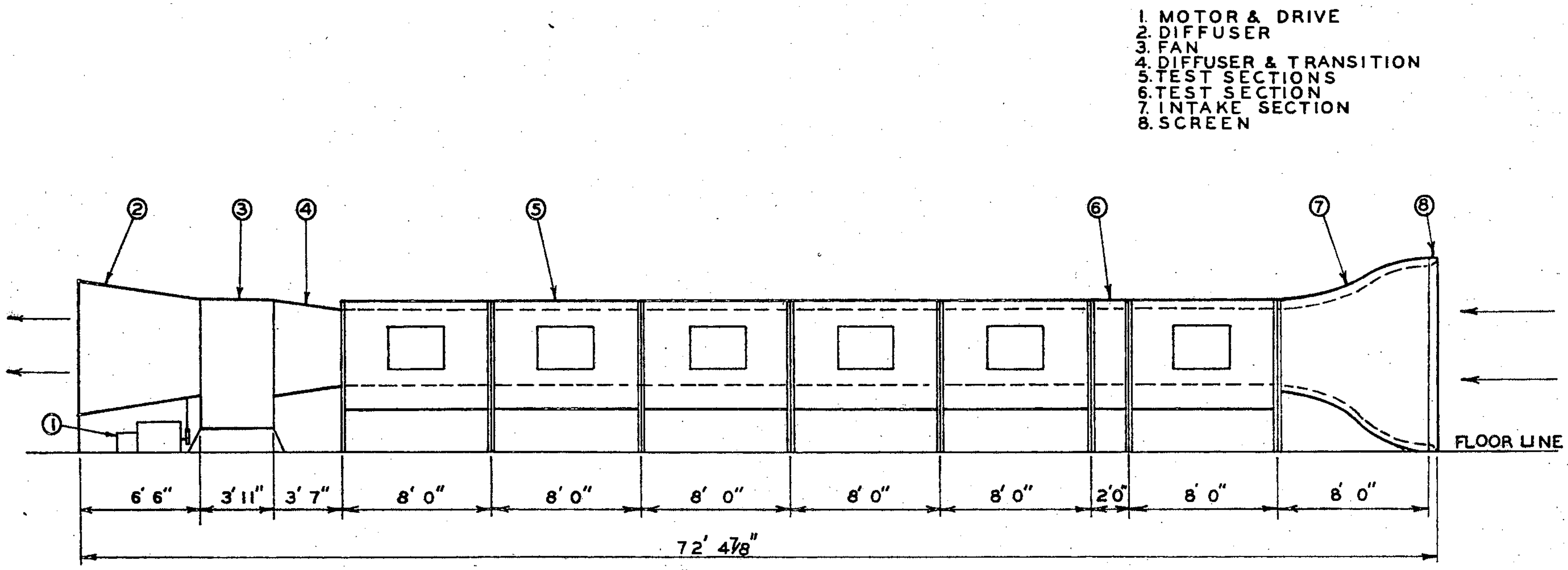

Figure 4. SCHEMATIC OF TUNNEL

in the office of the Argicultural Engineering Department.

It was not deemed necessary to discuss all of the mechanical problems of the design. Only those problems concerned with the air flow, the tunnel performance, or some unique feature were thought worthy of consideration.

A. Tunnel Test Sections

The tunnel test sections are seven in number, six of which are 8 feet in length and one which is 2 feet in length. Figure 5 shows the main features of the test sections. The framing members are of 2 X 6 inch kiln dried structural grade fir. In order to reduce skin friction and to protect against warping a plywood having a hard smooth overlaid surface was used. The bottom, top and side panels are of one-half inch high density plywood (Exterior DFPA), of natural color and overlaid on both sides.

The floor is adjustable in height at either end through a range of 9 inches. The entire floor panel may be removed and replaced with a specialized floor panel if so desired. If the floor is adjusted so as to give a slope, provision has been made to fill the space which will result between the floor sections, by a series of angular wedges.

A 2 X 3 foot access and observation door is provided on each side of the test section. The door panel is of three-eights inch thick plexiglass. Both the door and plexiglass panel are fitted flush on the inside. Larger size objects than can be accommodated by the access doors may be placed within the tunnel. The top is made as a separate panel and may be removed to provide access for large objects.

Figure 5. TEST SECTIONS

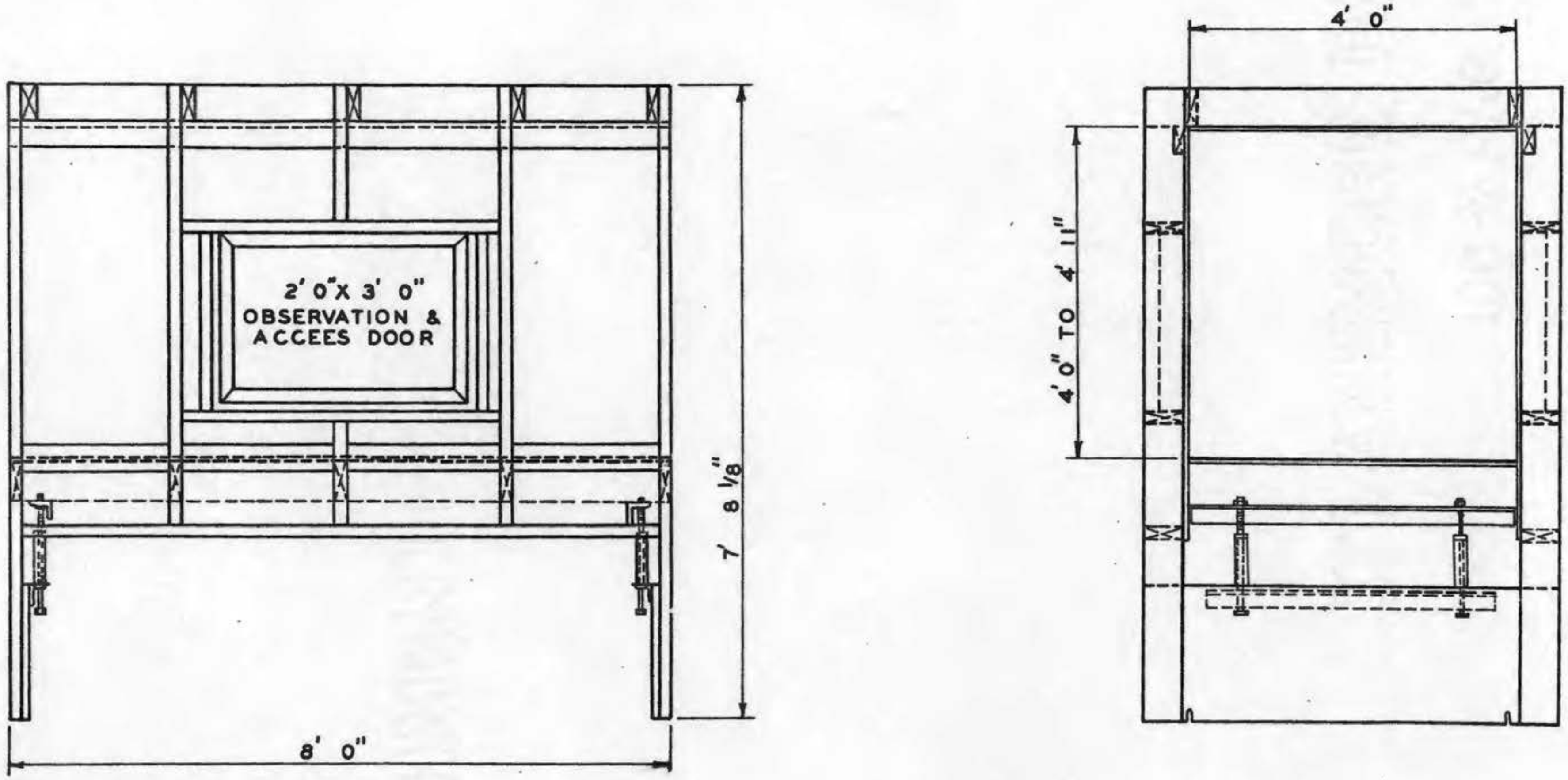

Figure 5.

The test sections may be drawn tightly together to provide proper sealing at the flange joints. Provision has been made for this by insetting the flanges slightly. A template was used in the manufacture of the test section elements to insure proper alignment of the panel ends.

Rigidity of the sections is provided for by a 1 inch thick by 20 inch deep plywood stiffener. A stiffener is placed at the bottom of the flange legs at each end of each section. Since the end flanges are cantilever beams, proper rigidity would be maintained in event the top panel were removed. The two foot section has no supports, being supported by the flanges of the end sections. This section may be removed readily to provide an area for the installation of any special air flow modification devices.

As the air proceeds along the tunnel sections the thickness of the boundary layer increases. To satisfy the law of continuity, the velocity in the center must increase. The velocity increase, in turn, results in a decrease in the static pressure. If the static pressure is to remain constant throughout the tunnel, the walls must diverge. Pope (6) states that the total angle of divergence between the walls should be approximately one degree. Adjusting the floor slope simultaneously with longitudinal static pressure measurements will result in obtaining the proper angle of divergence.

A safety screen is provided between the last test section and the transition section.

B. Intake Section

The purpose of the intake section is to increase the velocity of the air continously in such a manner so that at the exit of the intake section the velocity will be uniform and the turbulence decreased. To do this, the flow through the intake section must be such that no separation of the boundary layer occurs. The intake section should achieve its purpose within a reasonably short length.

Pope (6) states that the variation in velocity from the mean velocity varies inversely as the square of the contraction ratio. Thus:

$$\frac{\bar{v}_1}{\bar{V}_1} = \frac{1}{C^2}\,\frac{\bar{v}_2}{\bar{V}_2} \tag{4-1}$$

where:

C = the contraction ratio

$\bar{v}_2$ = velocity variation from the mean velocity at the intake section exit, feet per second

$\bar{V}_2$ = velocity at the intake section exit, feet per second

$\bar{v}_1$ = velocity variation from the mean velocity at the entrance to the intake section, feet per second

$\bar{V}_1$ = velocity at the intake section entrance, feet per second

A large contraction ratio would result in a low velocity variation, but would increase the cost. In event the tunnel were later converted to a return type tunnel, the return portion would be increased in size as well as the intake section. A contraction ratio of six was selected for use as best fulfilling both flow and cost considerations. Substitution into Eq. (4-1) yields a velocity variation decrease to 2.78 per cent of its initial value.

Several types of geometrical curves are given as suitable for the intake section profile. (6,7). Rouse and Hasson (8) state that a profile having an infinite radius of curvature at the point of tangency with the test section is preferred. They have shown by experiment that unless this condition exists, there will be, in general, a pronounced drop in pressure just before the juncture of the curved and uniform sections. The pressure thereafter will rise to that of the uniform flow. Cubical arcs (i.e., curves of the form $y = ax^3$) are preferable to either elliptical or circular arcs owing to the fact that their radius of curvature is infinite at their points of tangency with the uniform sections. (8).

Figure 6 is a sketch of the intake section profile. It consists of two cubical parabolas each of which are tangent to the horizontal and to each other.

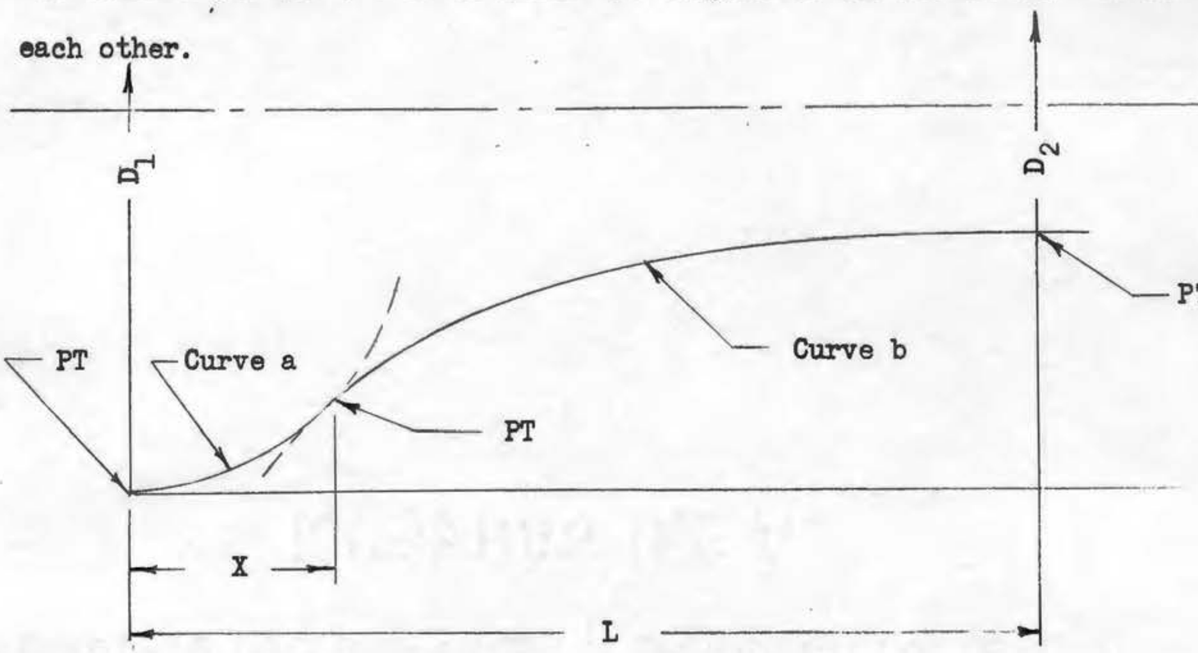

Figure 6. Intake Section Profile

The results of Rouse and Hasson's (8) experiments to obtain cavitation free profiles are shown in Fig. 7. The use of Fig. 7 permits the selection of the transition characteristics for a given contraction ratio and a permissible length ratio. The ratio X/L must be equal to or less

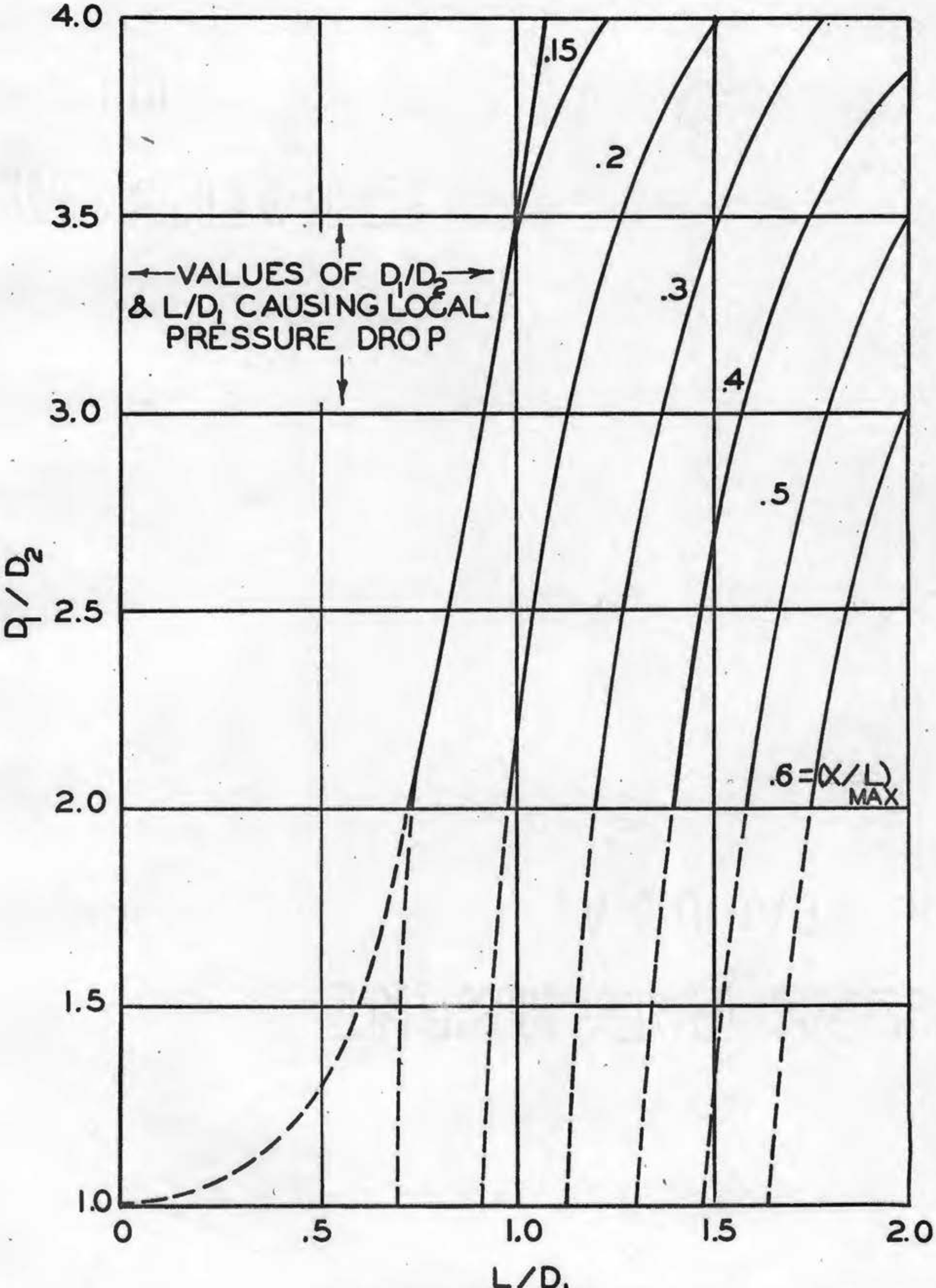

Figure 7. Contraction Ratios for Cavitation Free Flow

than the values indicated on the curves.

The height at exit from the intake section is the test section flow height; $D_2 = 4$ feet (see Fig. 6). Because of space limitations the maximum intake section length was 8 feet. The contraction ratio may be expressed as:

$$C = \frac{D_2}{D_1} \tag{4-2}$$

from which $D_1 = 9.81$ feet. Thus $L/D_1 = 0.816$ and $D_1/D_2 = 2.45$. From Fig. 7, a value of $X/L = 0.118$ is obtained from which the distance $X = 1.44$ feet may be found. Referring to Fig. 6, the value of 'y' at the point of tangency of curve 'b' with the test section is 2.905 feet.

The equations of the two parabolas are:

$$y_a = a_a x^3 \tag{4-3}$$

$$y_b = a_b x^3 \tag{4-4}$$

The boundary conditions are:

(1) at $x = 0$ $\qquad \dfrac{d\,y_a}{d\,x} = 0$

(2) at $x = 1.44$ $\qquad \dfrac{d\,y_b}{d\,x} = \dfrac{d\,y_a}{d\,x}$

(3) at $x = 8$ $\qquad y_b = 2.905$ and $\dfrac{d\,y_b}{d\,x} = 0$

From the boundary conditions the constants can be evaluated to give:

$$y_a = 0.175\,x^3 \qquad \text{for } x \leqq 1.44 \tag{4-5}$$

$$y_b = 2.905 - (8 - x)^3(0.008434) \quad \text{for } x \geqq 1.44 \qquad (4\text{-}6)$$

Equations (4-5) and (4-6) were used to obtain the ordinates of the intake section. The ordinates and intake section height for incremental values of length are given in Table I.

The profile and important dimensions are shown in Fig. 8. An isometric veiw of the intake section showing the important construction features is given in Fig. 9. The panels are laminated from two one-quarter inch plywood sheets. All ends are jointed over construction members. The ribs are made of 1 inch plywood and have a depth of 8 inches. At the intersection of all construction members, framing anchors are used.

C. Screen

A curved trim of elliptical shape is provided for the entrance to the intake section. The trim helps provide a smooth flow to the intake section in addition to providing a frame over which a turbulence reducing screen is stretched. The screen functions to reduce turbulence by breaking the large size disturbances into smaller ones which soon decay. (7). Pankhurst (7) states that if the screen is to be effective in reducing non-uniformity of flow, the pressure loss coefficient should be of the order of two. The pressure loss coefficient is defined as:

$$K = \frac{\Delta P}{\frac{1}{2}\rho \bar{V}_s^2} \qquad (4\text{-}7)$$

According to curves in <u>Wind-Tunnel Technique</u> by Pankhurst and Holder (7) the opening of the screen should be 51 per cent of the screen area

TABLE I

INTAKE SECTION ORDINATES

Distance From Entrance X-Feet	Curve Ordinates y-Feet	Height of Intake Section-D-Feet
0.0	0.0	9.81
0.50	0.022	9.766
1.00	0.175	9.460
1.25	0.342	9.126
1.44	0.523	8.764
1.50	0.586	8.638
1.75	0.843	8.118
2.00	1.083	7.644
2.50	1.502	6.806
3.00	1.851	6.108
3.50	2.137	5.536
4.00	2.365	5.080
4.50	2.544	4.722
5.00	2.677	4.456
5.50	2.773	4.264
6.00	2.838	4.134
6.50	2.877	4.056
7.00	2.897	4.016
7.50	2.904	4.002
8.00	2.905	4.000

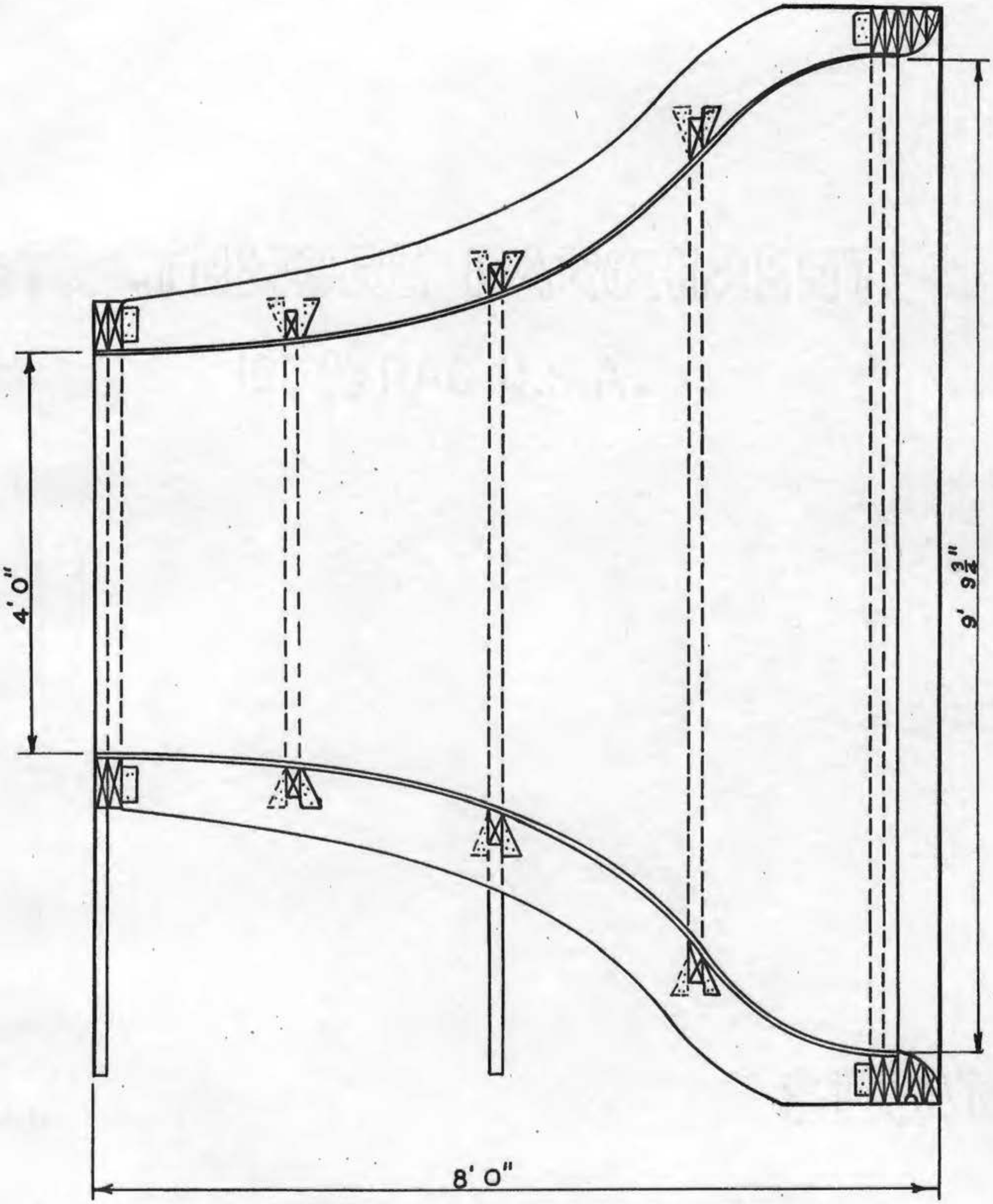

Figure 8. TUNNEL INTAKE — LONGITUDINAL SECTION

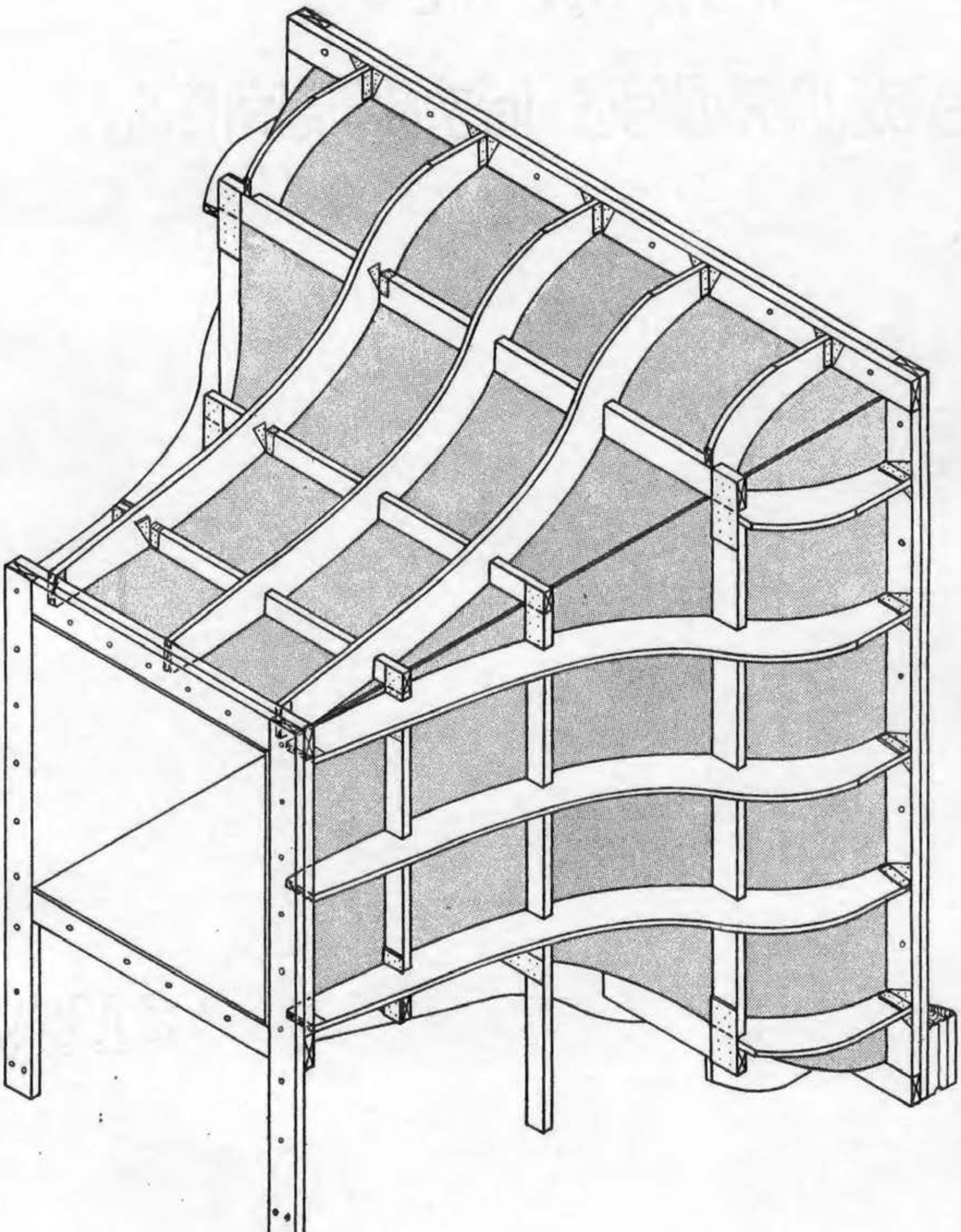

Figure 9. INTAKE SECTION

for a pressure loss coefficient of 2.5. If a test section velocity of 58.6 fps is assumed, the velocity at the screen is 9.78 fps. Substitution into Eq. (4-7) gives a value of P_s equal to 0.285 pounds per square foot.

Pankhurst (7) states that the R_n should be kept low. This requirement dictated the selection of a small diameter wire. The specifications for the wire screen are as follows:

Manufacturer	Buffalo Wire Works Co.
Mesh per Linear Inch	22 X 22
Diameter of Wire	0.0132 inches
Approximate Per Cent Open Area	50.5
Grade	Light
Material	304 Stainless

D. Diffuser

A diffuser is a channel which has for its purpose the conversion of kinetic energy in such a manner that the pressure increases. If the air flow were to take place directly from the fan to the atmosphere, the whole of the kinetic energy would be dissipated by friction into internal energy. An additional quantity of energy equal to that dissipated would then need to be supplied to the air stream. If a diffuser is used in an open-type tunnel, the pressure at exit from the fan will be lower than atmospheric pressure; thus the static pressure rise across the fan will be less than that required in the absence of a diffuser. This lower pressure rise will result in a lower power requirement by the fan. For small pressure rises Leonard (9) states that the power rating may be expressed as:

$$AHP = \frac{V \Delta P}{33{,}000} \qquad (4\text{-}9)$$

where:

V = volume, feet cubed per minute

The pressure gradient along the walls of a diffuser is such that it is difficult to avoid local separation or thickening of the boundary layer. Diffusion is, therefore, an inefficient process. At low speeds it is seldom possible to recover more than 90 percent of the kinetic energy. (3,7). The diffuser efficiency is defined as the ratio of the pressure rise in the diffuser to the dynamic pressure at the diffuser inlet. Thus:

$$N_D = \frac{P_D}{\frac{1}{2}\rho \bar{V}^2} \qquad (4\text{-}10)$$

For Fig. 10, the diffuser efficiency at low velocities is plotted as a function of the total included angle of the diffuser. (7). Each of the curves refer to a particular expansion ratio. The efficiency increases rapidly between 20 and 10 degrees, reaching a maximum at about 10 degrees. Both Pope (6) and Pankhurst (7) state that the optimum angle of divergence is about 5 degrees.

Because of space limitations the length of the diffuser was limited to 7.5 feet. If a high diffusion efficiency was desired, a small angle of diffusion would be required. For a short diffuser, a small expansion ratio would result in a high kinetic energy loss at the exit. The problem then was to find the angle of diffusion which would result in the highest overall efficiency. The overall efficiency may be expressed as:

$$N_o = N_D \times N \qquad (4\text{-}11)$$

where:

N_o = the overall diffuser efficiency, per cent

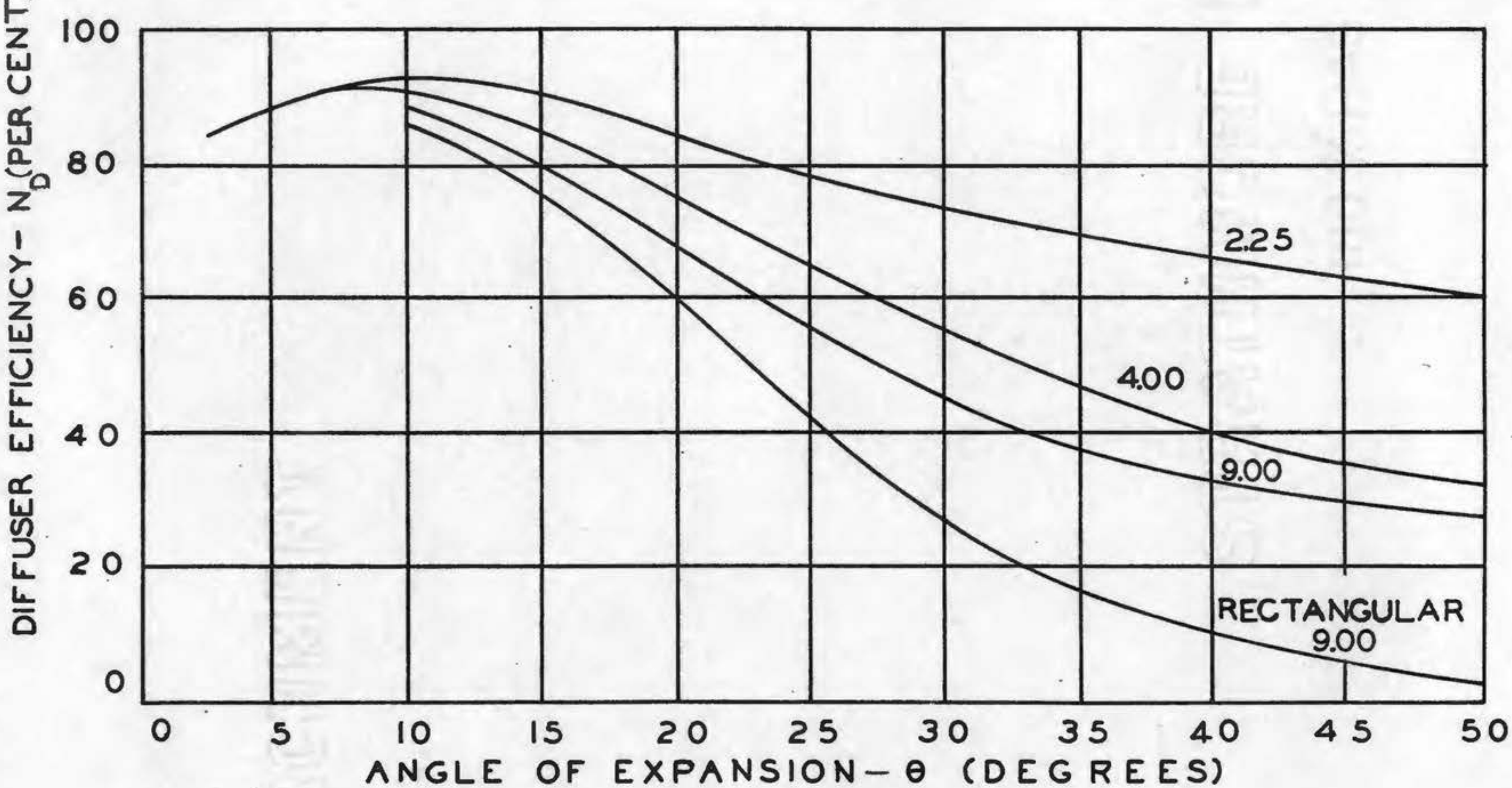

Figure 10. Diffuser Efficiency

N_D = the diffuser efficiency, per cent

N = the theoretical energy conversion within a diffuser, per cent

The theoretical energy conversion which is possible within a diffuser may be expressed as a function of the expansion ratio. In terms of the kinetic energy:

$$N = \frac{Ke_1 - Ke_2}{Ke_1} \tag{4-12}$$

Kinetic energy may be shown to be:

$$Ke = \frac{1}{2} \rho \bar{V}^2 \tag{4-13}$$

Thus:

$$N = 1 - \frac{\bar{V}_2^2}{\bar{V}_1^2} \tag{4-14}$$

If the flow is considered to be incompressible:

$$A_1 \bar{V}_1 = A_2 \bar{V}_2 \tag{4-15}$$

Thus Eq. (4-11) may be expressed in terms of the expansion ratio as:

$$N = 1 - \frac{1}{B^2} \tag{4-16}$$

where:

B = the expansion ratio, dimensionless

The graph of Eq. (4-16) is shown in Fig. 11.

From preliminary design considerations the space available for the

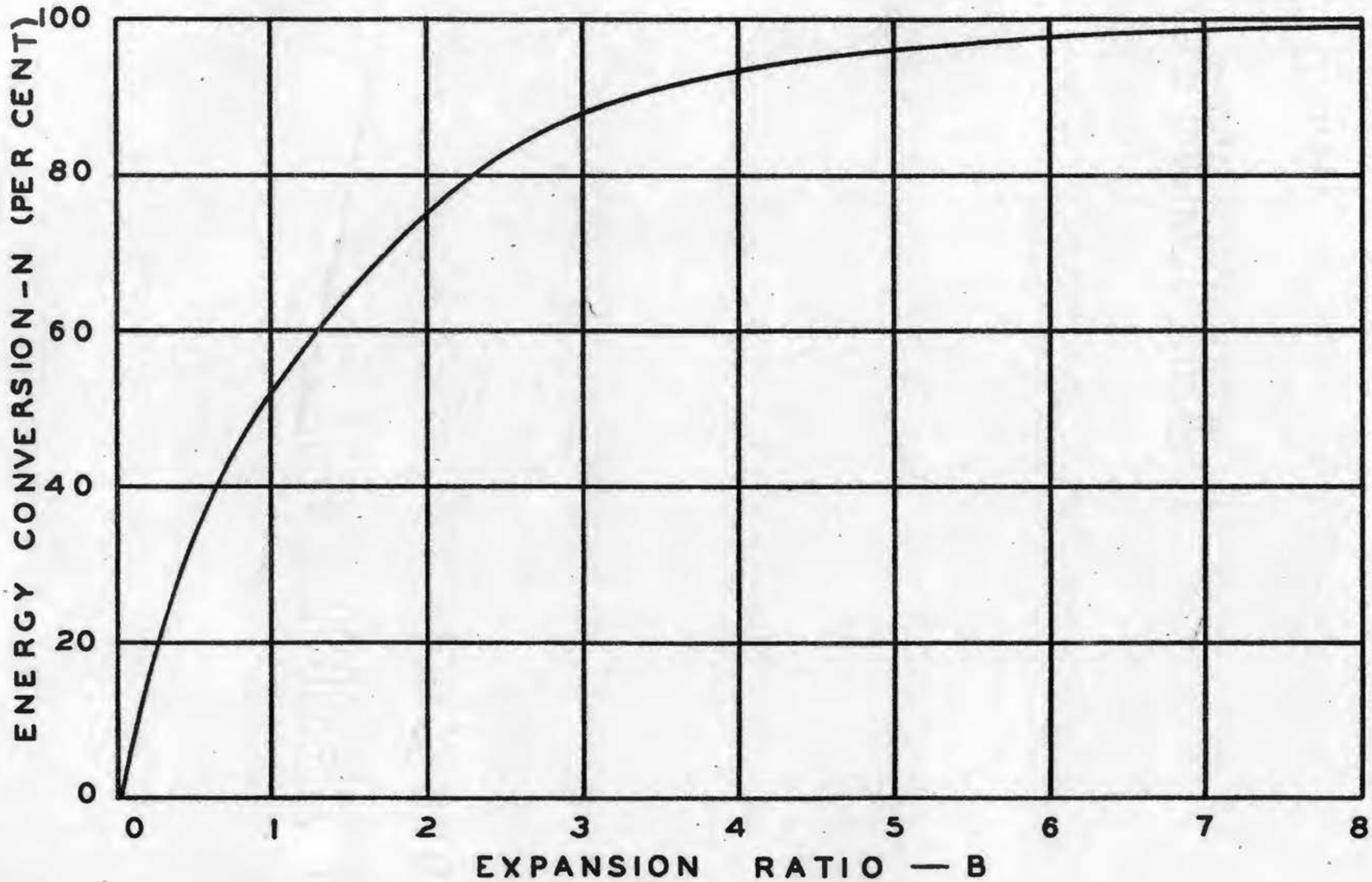

Figure 11. Theoretical Energy Conversion in a Diffuser

diffuser was within a range of 5 to 17 feet. In order to find the maximum overall efficiency for any length diffuser within this range, the maximum efficiency was found for several lengths. For each length, the angle of expansion for various expansion ratios was determined. Values of 'N_D' and 'N' were obtained from Figs. 7 and 8 respectively. These values were then introduced into Eq. (4-11) to give the results which are tabulated in Table II and shown graphically in Fig. 12. The maximum overall efficiency values were then obtained and used as the ordinate of Fig. 13. Figure 13 shows the maximum overall efficiency as a function of the length and angle of diffusion.

After the other components were designed, the length remaining for the diffuser was 6.5 feet. The optimum angle of diffusion is 19 degrees (see Fig 13). The efficiency was estimated to be 65 per cent. The diameter at exit for the diffuser angle of 19 degrees is 7.18 feet which corresponds to an expansion ratio of 2.06.

Figure 14 shows the principal features and dimensions of the diffuser. The diffuser is constructed of 20 gauge sheet metal. Stiffeners are placed around the outside. Angle flanges are provided at each end. Between the fan and intake flanges, a sponge rubber gasket is provided to reduce the transmission of vibrations to the diffuser. The diffuser is supported at the intake end by the fan flange connection. An access opening is provided in the bottom of the diffuser for a B-belt drive. If the noise level is found to be high, the diffuser may be covered with a sound deadening material.

E. Fan

Throughout the wind tunnel the successive pressure drops which occur

TABLE 11

DIFFUSER EFFICIENCIES FOR VARIOUS LENGTHS

Expansion ratio-B	Theoretical Eff.-%	Exit Size of Sq. Diffuser-Feet	Angle of Diffusion-Deg.	Diffuser Eff.-N_D	Overall Diffuser Eff.-N_o
A. L = 14 feet					
6.0	97.4	9.81	23.0	65	63.3
5.0	96.0	8.94	20.3	73	71.0
4.5	95.1	8.48	18.2	76	72.2
4.0	93.7	8.00	16.4	83	77.8
3.5	91.5	7.48	14.1	88	80.5
3.0	89.0	6.93	11.9	91	81.0
2.5	84.0	6.32	9.5	91	76.3
2.0	75.0	5.66	6.8	90.0	67.5
1.0	00	4.00	00	--	00
B. L = 17 feet					
6.0	97.4	9.81	19.4	75	73.1
5.0	96.0	8.94	16.5	82	78.7
4.5	95.1	8.48	15.0	84	79.9
4.0	93.7	8.00	13.4	87	81.5
3.5	91.5	7.48	11.7	90	82.3
3.0	89.0	6.93	9.9	91	81.0
2.0	75.0	5.66	5.6	89	66.7
1.0	00	4.00	00	--	00

TABLE 11 (Continued)

Expansion ratio-B	Theoretical Eff.-%	Exit Dia. of Conical Diffuser-Ft.	Angle of Diffusion-Deg.	Diffuser Eff.-N_D	Overall Diffuser Eff.-N_o
L = 9 feet					
6.0	97.4	12.25	44.0	33	32.2
5.0	96.0	11.15	37.8	41	39.4
4.5	95.1	10.60	34.6	45	42.8
4.0	93.7	10.00	31.0	52	48.7
3.5	91.5	9.34	27.2	66	60.5
3.0	89.0	8.67	23.0	75	66.7
2.5	84.0	7.90	18.3	84	70.5
2.0	75.0	7.06	13.0	92	69.0
1.5	64.0	6.12	7.1	90	57.5
L = 5.54 feet					
6.0	97.4	12.25	66.2	25	24.4
5.0	96.0	11.15	58.0	27	25.9
4.5	95.1	10.60	53.6	28	26.6
4.0	93.7	10.00	48.6	32	30.0
3.5	91.5	9.34	42.8	42	38.4
3.0	89.0	8.67	38.6	51	45.4
2.5	84.0	7.90	29.4	72	60.5
2.0	75.0	7.06	21.0	84	63.0
1.5	64.0	6.12	11.6	92	59.0

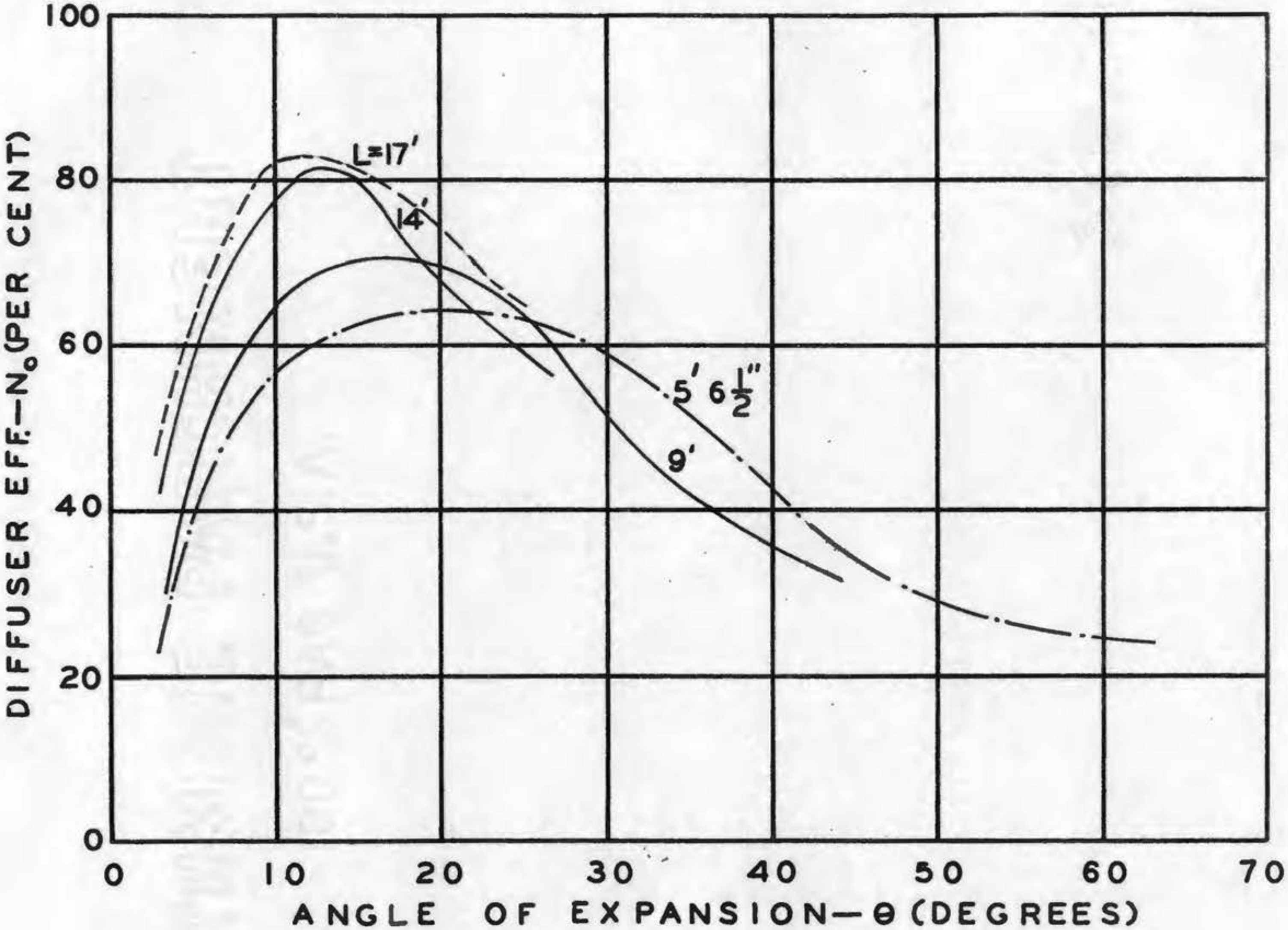

Figure 12. Overall Diffuser Efficiency

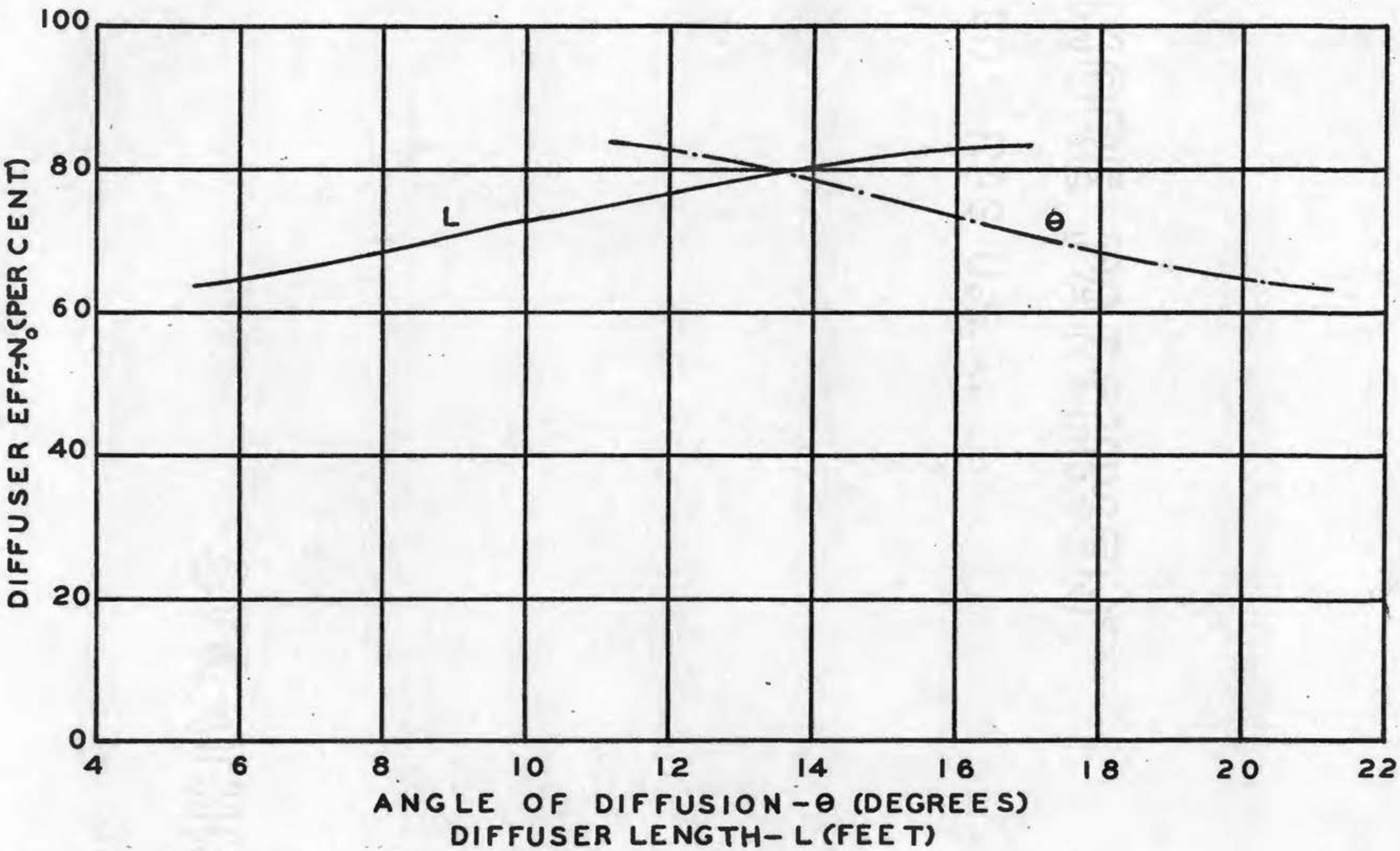

Figure 13. Maximum Overall Diffuser Efficiency

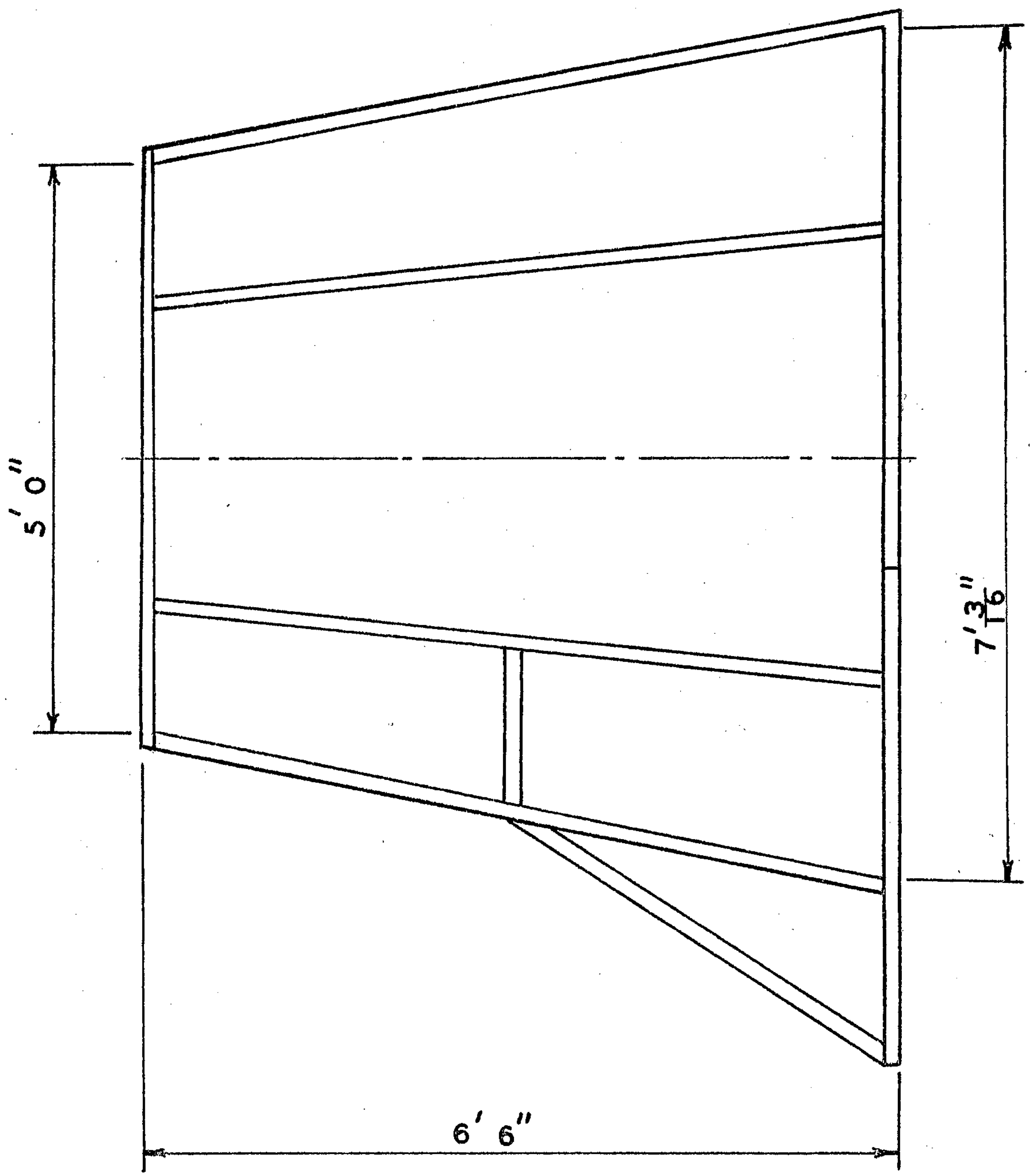

Figure 14. DIFFUSER

must be balanced by the fan. Before the fan could be selected it was necessary to determine the total pressure loss. Figure 15 is a sketch of the relative static pressures throughout the tunnel.

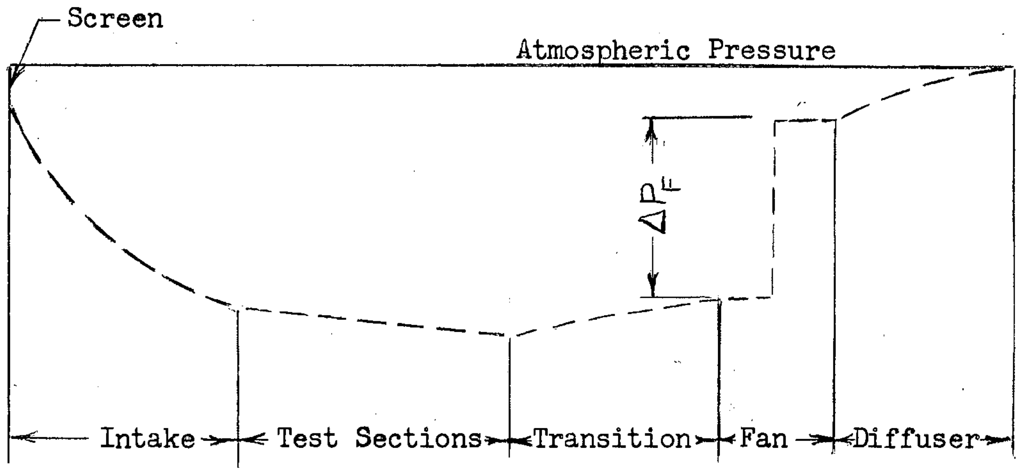

Figure 15. Static Pressure Variation Through Tunnel (Not to Scale)

Pope (6) has pointed the way towards a logical approach to the calculation of the losses in a wind tunnel. The procedure is to divide the total pressure losses into those occurring in the (1) intake section, (2) screen, (3) test sections, (4) transition section, (5) diffuser, and (6) those due to leakage.

In each section the pressure loss may be written in coefficient form as a ratio of the local pressure loss to the test section dynamic pressure. Thus:

$$K_o = \frac{P \quad q}{(q)(q_o)} = K \frac{q}{q_o} \tag{4-17}$$

where:

q = the dynamic pressure, pounds per square foot

K = pressure loss coefficient, dimensionless

The dynamic pressure is expressed as:

$$q = \frac{1}{2} \rho \bar{V}^2 \tag{4-18}$$

The static pressure rise across the fan becomes:

$$\Delta P_F = q_o \sum K_o \tag{4-19}$$

The following losses are found for a test section velocity of 58.6 (fps), corresponding to q_o equal to 4.08 pounds per square foot.

1. Intake Section

Pope (6) derives a relationship for K_{oi} to be:

$$K_{oi} = 0.32 \frac{L}{D_o} \tag{4-20}$$

By Eq. (2-1) the Reynolds number at exit from the intake section is 1.495 x 10^6. Figure 16 gives the skin friction coefficient as a function of the Reynolds number. (6). The friction factor is 0.013 (see Fig. 16). Substitution into Eq. (4-20) gives $K_{oi} = 0.083$.

2. Screen

The pressure drop across the screen is 0.285 (see page 25). Substitution into Eq. (4-17) gives $K_{os} = 0.07$

3. Test Sections

The Darcy-Weisbach equation for the evaluation of pressure losses in ducts is: (3)

$$\Delta P = \lambda \frac{L}{D} q \tag{4-21}$$

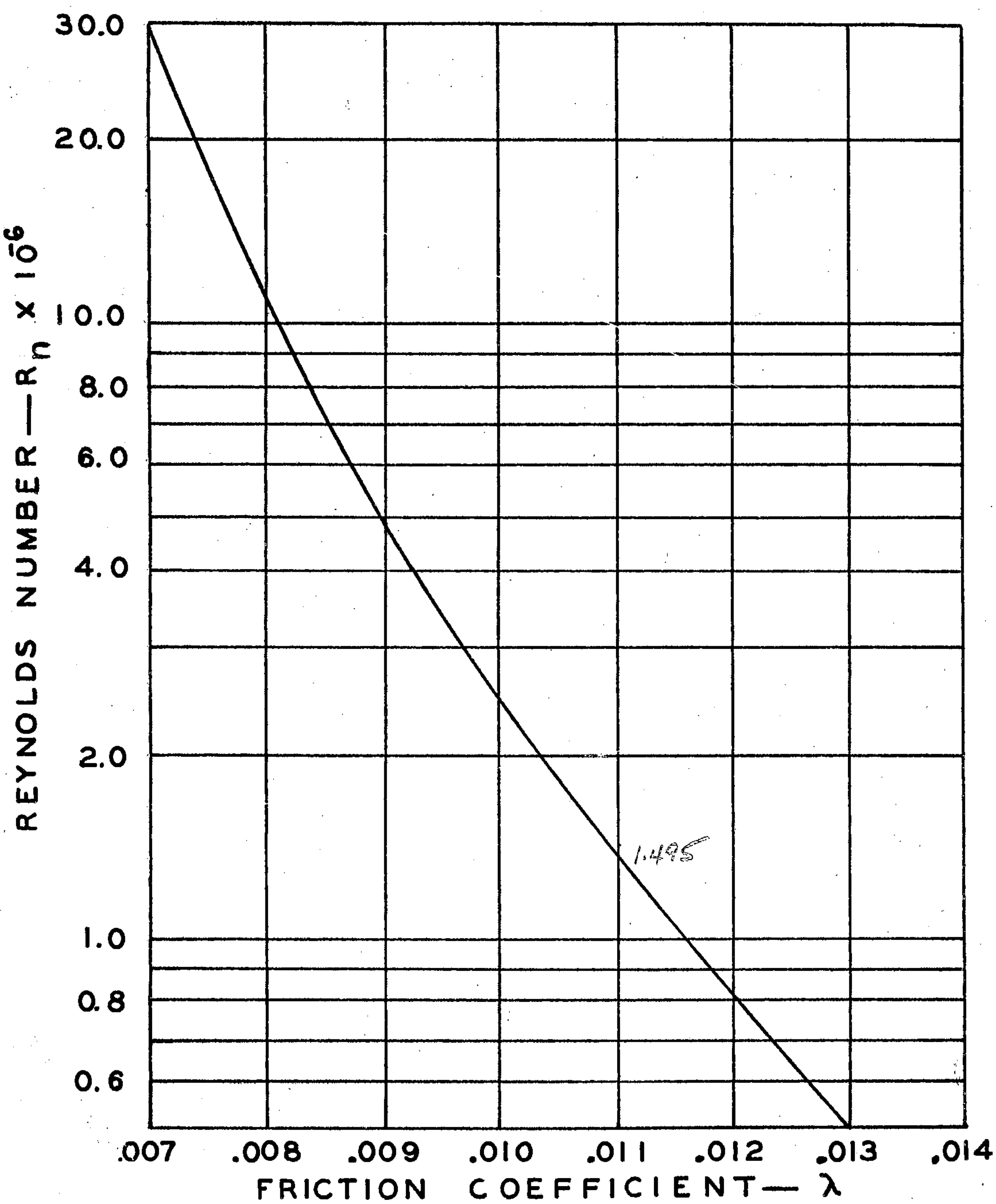

Figure 16. Friction Coefficient for Smooth Ducts

or in coefficient form as:

$$K_{oo} = \lambda \frac{L}{D} \qquad \frac{L}{D} = 12.45 \tag{4-22}$$

By Fig. ~~15~~ 16, $\lambda = 0.011$. Thus $K_{oo} = 0.137$.

4. Transition Section

The transition section changes the flow area from 4 ft. square to 5 ft. circular within a horizontal distance of 3 feet and 7 inches. This permits an increase in flow area which is equivalent to an angle of diffusion of 7.5 degrees. In a divergent section, both wall friction and expansion losses occur, which may be summed up by: (6)

$$K_{ot} = \left[\frac{\lambda}{8 \tan\left(\frac{\Theta}{2}\right)} + .6 \tan\left(\frac{\Theta}{2}\right) \right] \left(1 - \frac{D_1^4}{D_2^4} \right) \tag{4-23}$$

where:

D_1 = the smaller diameter, feet

D_2 = the larger diameter, feet

The effective smaller diameter = 4.5 feet. By Fig. 16, $\lambda = 0.011$. Substitution into Eq. (4-22) gives $K_{ot} = 0.020$

5. Diffuser

The overall diffuser efficiency has been estimated to be 65 per cent (see page 30). The overall efficiency term includes both diffuser losses as well as the residual kinetic energy loss. Combining Eq. (4-10) and Eq. (4-17):

$$K_{oD} = N_o \frac{q_d}{q_o} \tag{4-24}$$

At the diffuser entrance the velocity is 47.8 (fps), which corresponds to a dynamic pressure of 2.72 pounds per square foot. Substitution into Eq. (4-23) gives $K_{oD} = 0.433$.

6. Leakage Loss

The loss due to leakage and joints may be assumed to be approximately 10 per cent of the other losses. (6). A value of $K_{oL} = 0.070$ was assumed.

The loss coefficients are shown in the following tabulation:

Loss Coefficients

Intake Section	0.083
Screen	0.070
Test Sections	0.137
Transition	0.020
Diffuser	0.433
Leakage	0.070
	0.813

By Eq. (4-19) the pressure rise across the fan is 3.32 pounds per square foot or 0.64 inches of water. The flow rate is 56,200 CFM. Substitution into Eq. (4-9) gives a value of the air horsepower equal to 5.65.

Following are the specifications of the fan selected:

Manufacturer	Joy Manufacturing Co.
Model Number	60-26-860
Diameter	60 inches
Hub Diameter	26 inches
Number of Blades	16
Recommended Speed	860

The fan performance curves are given in Fig. 17. The fan has a wide performance range made possible by varying the blade setting. The fan has integral straightening vanes by means of which a portion of the rotational kinetic energy imparted to the flow may be recovered. In addition to the fan, the Joy Manufacturing Co. furnished the proper V-belt drive and transition section.

F. Drive Unit

The power requirement of the fan is 11.5 horsepower (see Fig. 17). If a drive efficiency of 95 per cent is assumed, the drive motor required is 12.1 horsepower. The motor and drive specifications are given in Appendix A.

The term energy ratio is commonly used to express the relative magnitude of the losses within a wind tunnel. The energy ratio is defined as the ratio of the kinetic energy of the air within the test section to the input energy. Basing the input energy as that imparted to the air by the fan, the energy ratio becomes:

$$E.R. = \frac{q_o}{q_o \sum K_o} = \frac{1}{\sum K_o} \tag{4-25}$$

If the energy ratio is defined to include the fan losses, then Eq. (4-25) maby be modified as:

$$E.R._F = \frac{N_F}{\sum K_o} \tag{4-26}$$

The fan efficiency is:

$$N_F = \frac{AHP}{BHP} \tag{4-27}$$

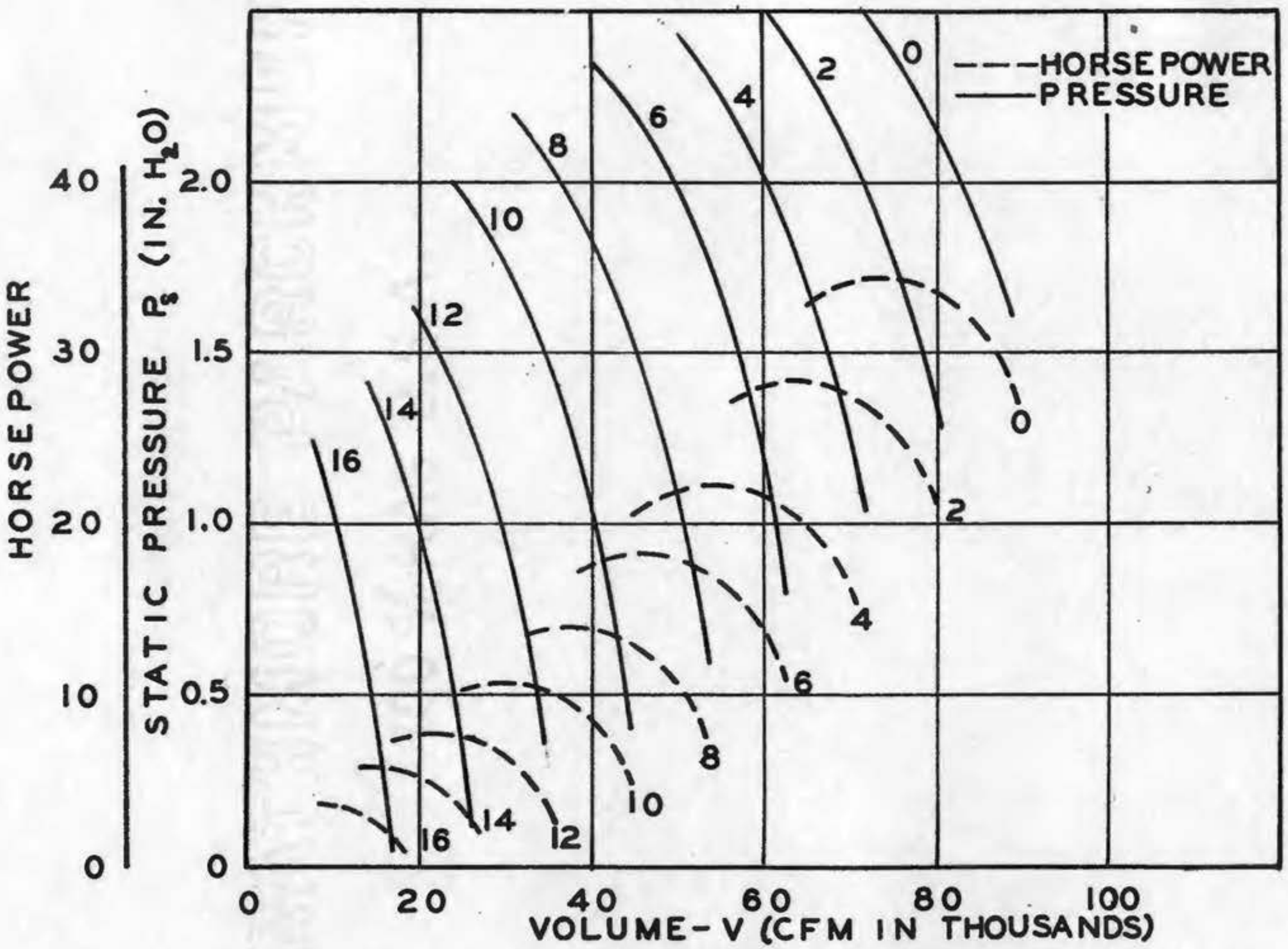

Figure 17. Typical Performance Curves for Joy Fan—Model 60-26-860

Using the values of horsepower previously determined, the fan efficiency is found to be 49 per cent. Substitution into Eq. (4-26) gives an energy ratio including fan losses of 0.602.

CHAPTER V

SUMMARY AND CONCLUSIONS

A low speed research wind tunnel has been designed. At the present time all components have been fabricated with the exception of the diffuser; therefore, the fulfillment of the stated objectives await verification. The prime objective of the tunnel is to provide a simulated natural air flow. It is believed that this objective will be satisfactorily attained.

If the boundary layer gradient requires modification, this may be accomplished. To do so, requires a change in the floor roughness. Thus, a combination of the proper velocity and floor surface roughness should make duplication of all natural air flows possible.

Higher velocities may be attained by the addition of a larger drive unit. The fan has capacity suitable for a tunnel speed of 62 MPH. Higher velocities may also be attained by the use of a smaller test section. Several of the standard test sections could be removed and in turn be replaced by a contraction element, smaller test section, and a diffuser.

The tunnel may be readily adapted to fit the requirements of many research projects. Because of this, in addition to other factors mentioned, the tunnel should prove to be a valuable tool for many research activities.

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APPENDIX A

DRIVE UNIT SPECIFICATIONS

Motor

Manufacturer	Robbins and Meyers
Serial Number	M5929SG
Horsepower	15
Amperes	39.5/19
Volts	220/440
Type	Alternating Current
Phase	3
RPM	1140

Variable Speed Drive

Manufacturer	Reeves Pulley Co.
Serial Number	MD-13046
Size	6281-C-12
Gear Ratio	1.54-1
Belt Number	M661430
Max. RPM	1500
Min. RPM	250
Horsepower of Motor	15

APPENDIX B

Following is a list including the specifications, of the instruments which have been selected for use:

HOOK GAGE MANOMETER

Manufacturer	F. W. Dwyer Mfg. Co.
Manufacturer's No.	1420
Range	0-2 inches of water
Accuracy	± 0.001 inches of water

INCLINED MANOMETER

Manufacturer	F. W. Dwyer Mfg. Co.
Manufacturers No.	100
Range	0-1 inch of water
Accuracy	± 0.01 inches of water

HOT WIRE ANEMOMETER

Manufacturer	Flow Corporation
Model No.	HWA3

A. Wire Current Circuit

Heating	Constant current supply (25 ma. to 300 ma.)
Battery	Built-in, capacity 40 hr. at 60 ma. continous

B. Amplifier

Input noise level	10 volts rms
Max. output noise level	10 volts rms
Max. output signal	50 volts

C. Hot Wire Anemometer Auxiliary Equipment

2 Probes	Model HWP-A
Power Supply	Lambda Model 28
Filter	Model F-7

PITOT TUBES

A.

Manufacturer	F. W. Dwyer Mfg. Co.
Manufacturer's No.	164
Length	48 inches
Type Mounting	Flange
Material	Stainless Steel

B.

Manufacturer	F. W. Dwyer Mfg. Co.
Manufacturer's No.	163
Length	36 inches
Type Mounting	Flange
Material	Stainless Steel

VITA

Stanley Oscar Brauser

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