# MODEL EXPERIMENTS FOR THE DESIGN
# OF A
# SIXTY INCH WATER TUNNEL

## PART IV

## DIFFUSER STUDIES

by

St. Anthony Falls Hydraulic Laboratory
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Project Report No. 13

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## PREFACE TO REPORT SERIES

Contract NObs-34208 between the University of Minnesota and the Bureau of Ships, Department of the Navy, provides for making hydrodynamic studies in compliance with specific task orders issued by the David Taylor Model Basin calling for services to be rendered by the St. Anthony Falls Hydraulic Laboratory. Certain of these task orders related to model experiments for the design of a 60-in. water tunnel.

The end result of the studies related to the 60-in. water tunnel proposed eventually to be constructed at the David Taylor Model Basin was crystallized as a series of six project reports, each issued under separate cover as follows:

- Part I DESCRIPTION OF APPARATUS AND TEST PROCEDURES (Project Report No. 10)
- Part II CONTRACTION STUDIES (Project Report No. 11)
- Part III TEST SECTION AND CAVITATION INDEX STUDIES (Project Report No. 12)
- Part IV DIFFUSER STUDIES (Project Report No. 13)
- Part V VANED ELBOW STUDIES (Project Report No. 14)
- Part VI PUMP STUDIES (Project Report No. 15)

The investigational program was under the general direction of Dr. Lorenz G. Straub, Director of the St. Anthony Falls Hydraulic Laboratory, and the work was supervised by John F. Ripken, Associate Professor of Hydraulics.

# PREFACE TO PART IV

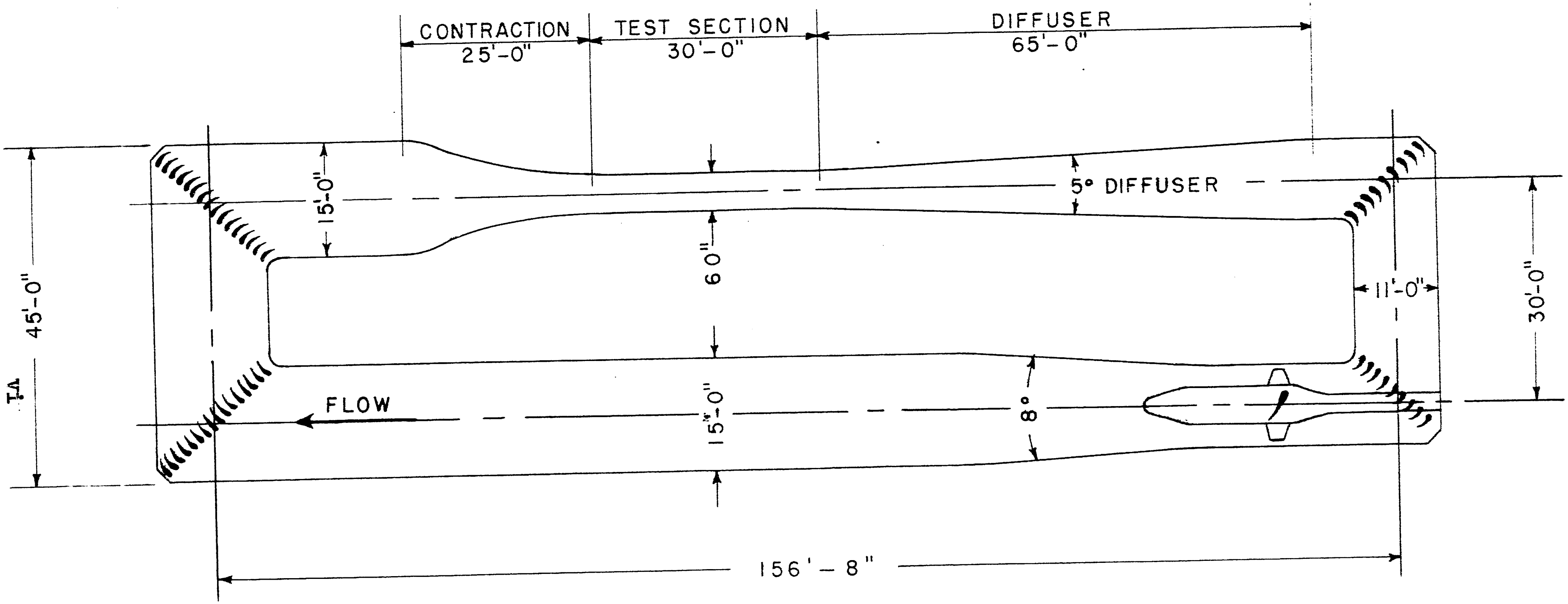
Part IV of the report series was prepared in accord with Task Order 2 of Contract NObs-34208.

This report was written by James S. Holdhusen, Research Fellow. Mr. Holdhusen was project leader on the experimental work and was assisted by Clyde O. Johnson and Harry D. Purdy. Ethel Swan and Elaine Hulbert assisted in preparation of the manuscript.

# SYNOPSIS

A study of the main diffuser (the diffuser following the test section) for a water tunnel was made at the St. Anthony Falls Hydraulic Laboratory as part of the research program for the design of the proposed 60-in. water tunnel of the David Taylor Model Basin, Bureau of Ships, Department of the Navy. The diffuser was designed on the basis of previous investigations of other experimenters, and the testing program was conducted to determine the action of the diffuser as it will be affected by the conditions of inflow and outflow in the recirculating system of the prototype water tunnel. It was determined that the diffuser will operate satisfactorily as to energy loss but that there will be slight maldistribution of velocity at the downstream end. It was decided, however, that this maldistribution will not be sufficiently detrimental to operation of the water tunnel to warrant making the diffuser longer and of a smaller angle of divergence.

## C O N T E N T S




PROPOSED 60" WATER TUNNEL

# D I F F U S E R  S T U D I E S

## I. INTRODUCTION

This paper discusses the experimental design studies of the simple conical diffuser proposed for the 60-in. water tunnel. The purpose of the water tunnel diffuser is to reduce efficiently the high discharge velocity of the test section before the stream encounters the other components in the return circuit. The advantages of the reduction in velocity are threefold:

1. The head loss in the return circuit is a function of the square of the velocity. Since an extremely high velocity is obtained in the test section, a lower velocity in the return circuit will markedly reduce the energy losses and required driving power.

2. Acceleration of the stream just before it enters the test section affords great advantages in control of uniformity of the test section flow and turbulence level. Since the acceleration is accomplished by reduction of area, the cross-sectional area of the stream must be increased in the return circuit or diffuser in order that the stream may be accelerated again.

3. A minimum of background noise during test operations is one of the fundamental requirements for the tunnel. Operations in existing tunnels indicate that a large source of such noise is the cavitation produced in the elbows and pump of the return circuit. Since cavitation is decreased by a decrease in velocity and an increase in pressure, an increase in duct area will contribute dual benefits with respect to noise abatement.

## II. DESIGN CONSIDERATIONS

### A. Main Diffuser Shape

The design of a diffuser centers on prevention of backflow, or separation, at the boundary. This separation is caused by the normal, adverse pressure gradient encountered in diffusers tending to force the fluid particles having low kinetic energy upstream, that is, those particles in the boundary layer. As soon as a particle loses so much energy that it can not overcome the pressure gradient, it reverses its flow and a zone of separation is set up which destroys the desired geometry of flow and prevents regain of pressure. In the violently eddying separation, high turbulent shears exist with consequent large energy losses. The zone of separation would also cause a distorted velocity front, which is detrimental to the operation of the tunnel components following the diffuser.

Separation of the boundary layer can be reduced by any of several methods. The simplest and most common method is to have the flow expansion gradual enough so that the adverse pressure gradient is not the dominant part of the dynamic forces which produce the bulk movement. Many experiments [1, 6, 7]* have been carried out to determine the best form of the flow expansion. The consensus of opinion of the investigators seems to be that a simple conical expansion is at least as good as the more complex forms investigated.

If space is limited and a rapid expansion of area is desired, separation can be prevented by suitable control of the boundary layer. Experiments by Prandtl and others have indicated that sucking away of the boundary layer is very effective, since the fluid having low kinetic energy is removed before it can reverse direction. Also, if adequate energy is supplied to the boundary layer, the pressure gradient can be overcome without separation. The additional energy can be supplied by introducing more turbulence into the stream, thus increasing the amount of energy carried from the interior regions of the flow to the boundary layers by momentum transfer. A second method of supplying energy to the boundary layer is one of pumping a jet of high kinetic energy into the boundary layer, or diversion of some of the flow having high kinetic energy to the boundary by deflectors or vanes. Still another method of boundary layer control is rotation of the flow about the diffuser axis. This rotational motion induces a radial pressure gradient and a radial flow, which add energy to the boundary layer.

All of the methods above of the boundary layer control were considered in the preliminary design of the water tunnel and were rejected as being inherently injurious to the flow quality, as increasing the possibility of cavitation in the tunnel, or as being too expensive. Also, models of various shapes and sizes will be mounted in the test section, and these models will greatly alter the velocity distribution and thus the efficiency of the

*Numbers in brackets refer to the corresponding numbers in the list of references, p. 17.

diffuser. Consequently, a particular method of boundary layer control might operate very well for a certain test installation and yet be of little or no value for another installation.

The main diffuser was therefore designed as a simple cone having a total included angle of divergence of $5^{o}$, since previous investigations pointed to this angle of divergence as giving a minimum head loss coupled with preferred flow qualities. Although it was realized that this small angle of divergence would necessitate making the diffuser quite long, this was considered necessary in order to obtain freedom from separation and good velocity distribution at the downstream end of the diffuser.

## B. The Transition Curve

A parabolic transition having a length of one-half test section diameter was provided to avoid a sharp break in the boundary at the intersection of the test section cylinder and the diffuser cone. Because the literature on the subject failed to establish a proven method of design, the shape and length of the curve were arbitrarily selected and for these studies are defined as shown in Fig. 1.

Easement of the boundary curvature is essential at this point because it is the seat of lowest tunnel pressure in combination with the highest velocity, and as such serves as the focal point of cavitation for the entire tunnel circuit.

## C. The Pump Diffuser

The short diffuser following the pump is not analyzed in this section. Since its function is dependent on the characteristics of the pump impeller, hub fairing and straightening vanes, the design of this diffuser will be considered the responsibility of the pump manufacturer. The mean velocity of flow will be quite small so that it may be assumed that the head loss will also be quite small, probably $0.12\ \bar{V}^2/2g$. ($\bar{V}$ is the mean velocity at entry.) Any undesirable velocity distribution resulting from the action of the diffuser will be smoothed out by the large diameter stilling section in the lower horizontal leg so that the other tunnel components will not be greatly affected.

## III. EXPERIMENTAL STUDIES

### A. Apparatus

The main diffuser in the model water tunnel was made in three sections. The upstream section was an aluminum casting bored to provide a transition between the cylindrical test section and the conical diffuser. The two downstream sections of the diffuser were fabricated from 3/16-in. thick sheet steel rolled to the proper shape and welded to steel flanges. The flanged joints were sealed with "O" rings permitting smooth metal-to-metal joints. The interior of the aluminum section was polished with emery cloth and the steel weldments were galvanized.

Velocity traverses were made at four different planes (designated as Stations 2, 3, 4, and 5 in Fig. 1) using a 3/8-in. diameter "cantilevered" pitot cylinder at Stations 3, 4, and 5 and a 1/4-in. "long" pitot cylinder at Station 2. Measurements were made at the piezometer taps located at the traverse stations. The details of this instrumentation are given separately in PART I, DESCRIPTION OF APPARATUS AND TEST PROCEDURE.

### B. Velocity Distribution

The velocity distribution in the diffuser is of importance because it affects the energy loss not only in the diffuser but also in the tunnel components following the diffuser. If separation occurs, energy is lost in turbulent shear and the velocity distribution is badly distorted; this distortion causes the kinetic energy of the stream to be high, which partly defeats the purpose of the diffuser. The mean velocity head of the stream is $\alpha \frac{\bar{V}^2}{2g}$ where $\alpha$ is defined as

$$\alpha = \frac{\Sigma \Delta A \; V^3}{A \; \bar{V}^3}$$

and the more non-uniform the velocity distribution is, the higher the value of $\alpha$ . In addition, a distorted velocity distribution at the end of the diffuser might persist in the stream up to the impeller, causing lowered efficiency in the pumping unit.

Velocity traverses were taken at Station 2 immediately preceding the diffuser, at Stations 3 and 4 in the diffuser, and at Station 5 immediately

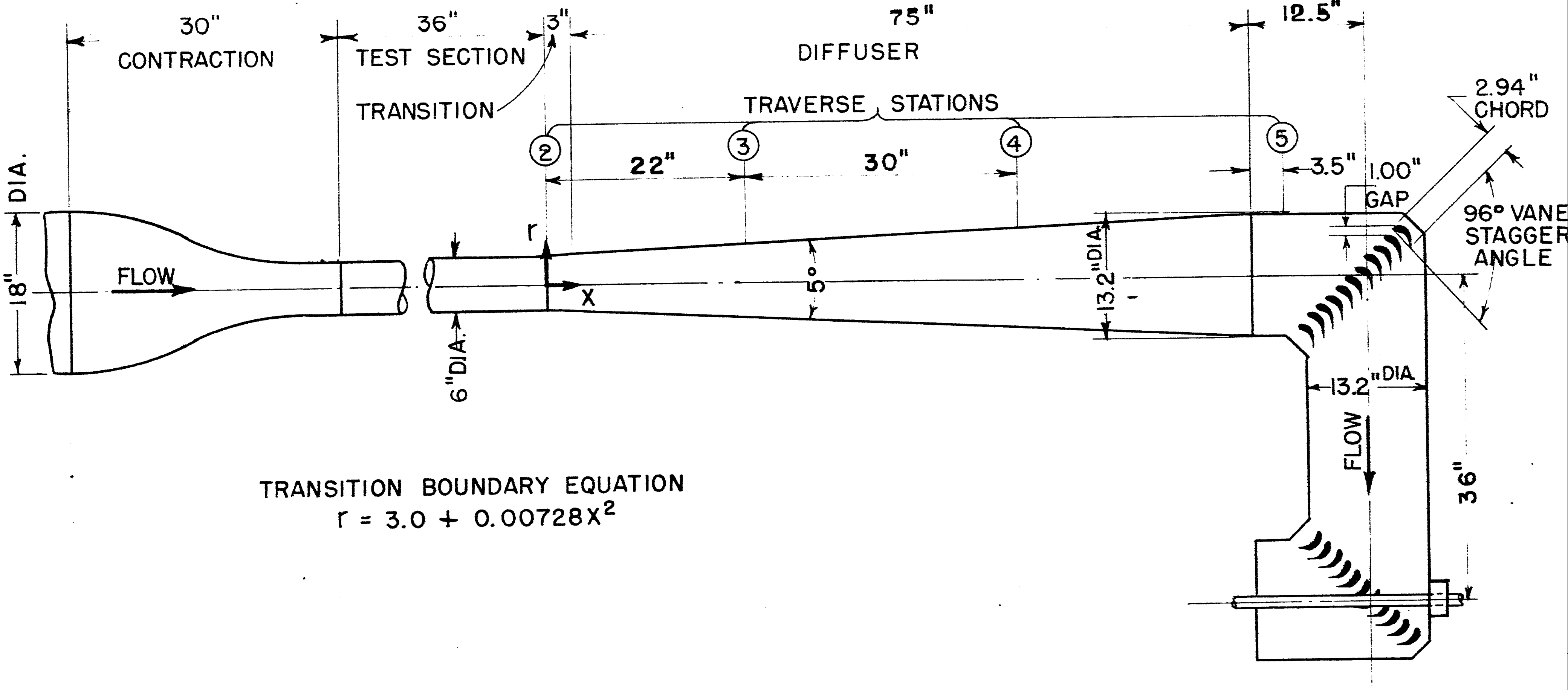



FIG. 1

SECTIONAL ELEVATION OF DIFFUSER FLOW CIRCUIT downstream of the diffuser. The location of these stations and dimensions of the diffuser are shown in Fig. 1. Traverses were taken on horizontal and vertical diameters at all four stations for a test section velocity of approximately 50 ft per sec, and on horizontal and vertical diameters of Station 2 at a test section velocity of 30 and 18 ft per sec. In addition, traverses were made on the vertical diameters of Stations 3 and 4 at a test section velocity of 30 ft per sec, in order to determine what effect lowering the velocity would have on the separation apparent at Station 4. At lower velocities (18 ft per sec), the velocity head at Station 4 would not have been great enough to permit making an accurate traverse. These traverses are presented in Fig. 2 and indicate that separation takes place at the downstream end of the diffuser and at the bottom (on the inside of the tunnel loop).

The prediction of the point of inception of separation caused by an adverse pressure gradient has long been one of the major problems in fluid mechanics. Separation occurs because the velocity of the fluid particle is being reduced while the particle is advancing into a region of higher pressure. The adverse pressure gradient is assisted by the friction drag of the walls in effecting a reduction in particle velocity, and these forces must be overcome by the kinetic energy gradient of the fluid if a reversal of flow direction is to be prevented. Although the axial pressure gradient and friction drag on a particle are probably nearly constant over the cross section of flow, fluid in the boundary layer has lower velocity and consequently will have a lower kinetic energy gradient, or come to rest sooner, than the fluid nearer the axis of the diffuser. If the boundary layer develops uniformly, every particle at an equal distance from the wall of a conical diffuser will have the same kinetic energy and come to rest at the same time. However, if there is some additional quality of the flow which causes one particle to have a lesser kinetic energy than other particles spaced symmetrically with respect to the walls, this particle will come to rest first.

An over-all analysis of the velocity traverses taken in the tunnel shows that the region of lowest velocity occurs at the inside of the tunnel loop. This means that the stream entering the contraction cone has a relatively low velocity near the bottom of the contraction and a high velocity near the top. This maldistribution is substantially but not entirely eliminated by the contraction cone; however, examination of the velocity traverses at the exit of the contraction cone indicates that the velocity in the boundary layer near the bottom of the cone is still slightly lower than at the top.

These traverses, however, were made with the vanes in Elbow IV, the elbow just preceding the contraction, set at a stagger of 96°. The studies of the vaned elbows have since indicated that the velocity deficiency in the region of low velocity will be lessened with the vanes at a stagger of 101°, as recommended in PART V, VANED ELBOW STUDIES. Consequently, one would expect a more uniform distribution of velocity at the exit of the contraction cone for the recommended vane stagger. In passing through the test section, the boundary layer grows thicker, but the sense of the distribution is not changed, as shown by the traverses at Station 2 at 49.62 ft per sec and 18 ft per sec test section velocity (PART III, TEST SECTION STUDIES). The traverse at Station 2 at 30 ft per sec velocity shows a reversed distribution with the highest velocity at the bottom of the section, but this may be due to an experimental error, since the upstream traverse (Station 1) shows a boundary layer velocity distribution at 30 ft per sec consistent with the traverses at 18 ft per sec and 49.62 ft per sec.

In view of these velocity measurements for the test section, it may be assumed that a tendency toward reversal of flow, or separation, will first occur near the bottom of the section. When this region of flow separates, or reverses its flow, the stream no longer conforms to the solid boundary but passes through a smaller area. The pressure gradient is thus made less severe and the tendency for separation to occur at the opposite wall will be alleviated, causing the stream to lose symmetry.

The work of Nikuradse [1] has shown that the flow through a diverging rectangular conduit (approximating two-dimensional flow ) will separate from the boundary at a certain critical angle of expansion. Nikuradse's experiments indicate that non-symmetrical velocity distribution first occurs when the half-angle of divergence ($\theta$) is $4^\circ$, and that backflow or separation will first occur when $\theta$ is from $4.8^\circ$ to $5.1^\circ$. When $\theta = 7^\circ$, the region of separation oscillates from one channel wall to the other, and this oscillation increases in frequency as $\theta$ increases.

In a comparison of the process of flow in a circular diffuser with flow in a two-dimensional diffuser (with an equal angle of divergence), it is not possible to assume that similar patterns of flow will prevail. The phenomenon of separation is caused by the pressure gradient and not by the flare of the walls. This has been demonstrated [2] by building a conduit with a rectangular cross section such that the width of one side increased and the width of the other side diminished, thereby keeping the area constant. When fluid flows through such a conduit, there is no separation from the walls. Even if the length and the initial and final end areas in two diffusers, one conical and the other rectangular (two-dimensional), are the same, the pressure gradients for equal discharges will not be the same, as can be demonstrated by Bernoulli's equation. Thus, it is seen that a direct comparison with Nikuradse's results cannot be made for the proposed conical diffuser.

The change in kinetic energy associated with the development of the boundary layer is another factor that must be considered in the action of a diffuser, since the pressure gradient in a diffuser must be overcome by the kinetic energy gradient of the stream. Even in a cylindrical conduit, the boundary layer has a kinetic energy gradient in the space in which it is developing.

The growth of the boundary layer is due to the retarding effect of the walls and the eddy viscosity of the fluid in the case of turbulent flow. Since these two factors vary with Reynolds number and thus vary with duct diameter, the whole problem becomes exceedingly complex. Therefore, the prediction of prototype velocity distribution from the model study can be only qualitative.

For equal velocities of flow in the model and the prototype, the pressure gradient in the prototype will be one-tenth that in the model because of the scale ratio. In other words, the same regain of pressure will be accomplished, but over a length ten times as great in the prototype as in the model. Also, if similar flow patterns obtain, the kinetic energy gradient will be one-tenth as large in the prototype. However, because of increased Reynolds number, the friction drag on a particle in the prototype will be less than on a homologous particle in the model, since the friction factor, and therefore the wall shear, decrease with increasing Reynolds number. This indicates that forces opposing the forward movement of the particles will be somewhat less severe in the prototype and that separation will not occur so readily. This conclusion is also borne out by the fact that the boundary layer will develop more slowly in the prototype. According to theory, the axial length of a cylindrical pipe over which the boundary layer will develop fully is given by the formula,

$$\frac{L}{d} = K\ Re^{n}$$

where

$L$ = length fully to develop boundary layer
$K$ = a constant
$d$ = pipe diameter
$Re$ = Reynolds number
$n$ = constant $> 0$

Thus, for increasing Reynolds number, the boundary layer develops more slowly, and the retardation of its particles is less severe.

It should be noted here again that the velocity distribution in the prototype diffuser will be profoundly influenced by the installation of a model in the test section. While the present design may not operate to perfection as regards velocity distribution, it does not seem feasible to increase the over-all dimension of the prototype circuit by using a diffuser of smaller angle of divergence, since the amount of separation will vary with the test installation.

## C. Energy Loss

Most of the experimental work done previously on diffuser design has centered on finding the efficiency of energy conversion for various angles of divergence, cross-sectional shapes, and patterns of inflow. In order to determine if the selected tunnel diffuser was operating efficiently under its particular pattern of inflow and outflow, the ratio of pressure energy gain to kinetic energy loss was determined from the experimental data in the following manner.

Assuming that the flow is purely axial, the total kinetic energy passing a section per unit time is

$$\int_0^A \frac{\rho}{2} V^3 \, dA,$$

and the kinetic energy per pound of fluid flowing is $\alpha \frac{\bar{V}^2}{2g}$ where $\alpha$ is defined as on page 4. Three values of $\alpha_2$, corresponding to the three different rates of flow at which the pressure gradient was measured, were computed from velocity traverses on horizontal and vertical diameters of Station 2 (Fig. 2). A similar traverse at Station 5 (Fig. 2) for a single test section velocity of 49.62 ft per sec was used to calculate $\alpha_5$.

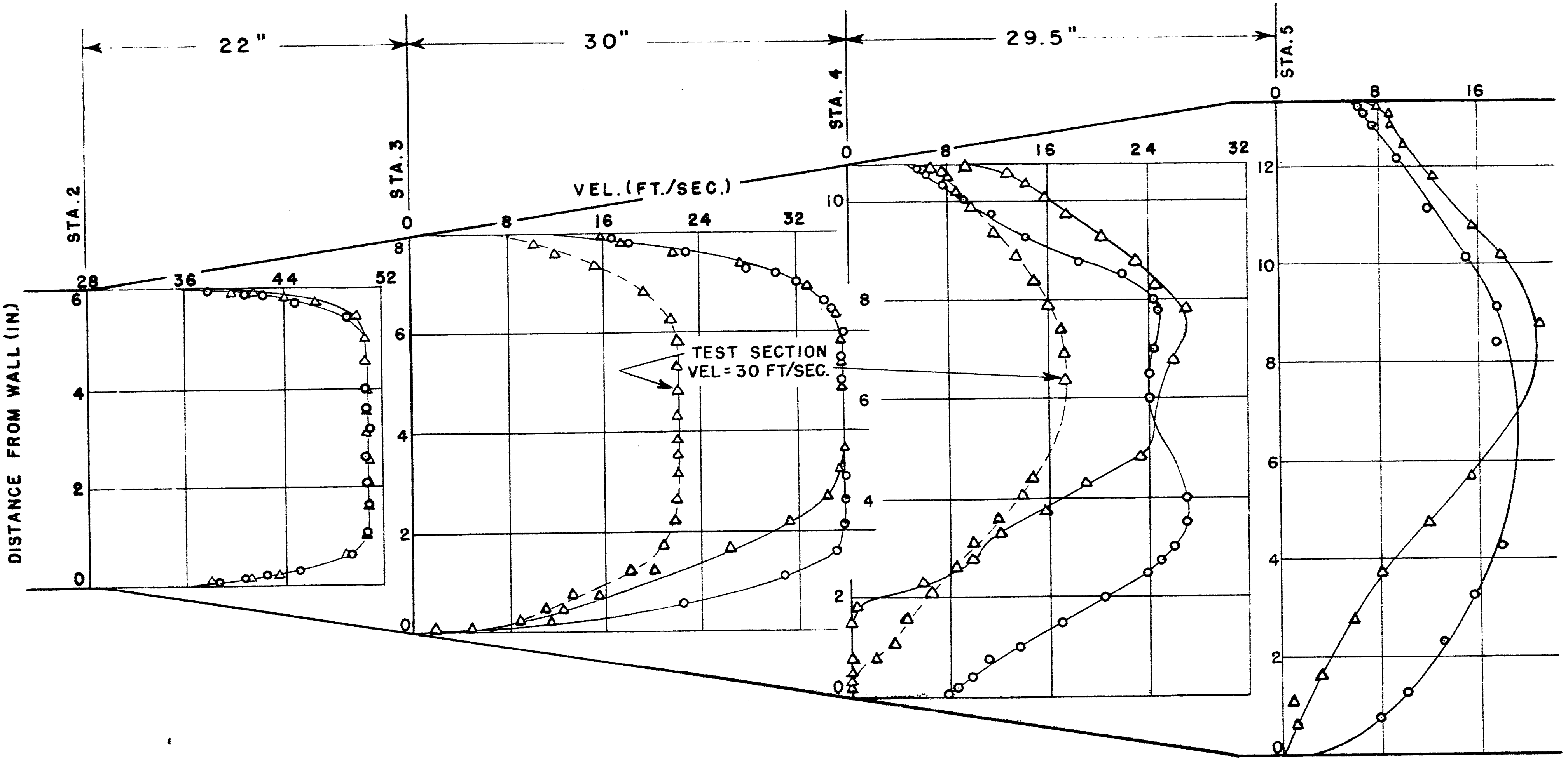



FIG. 2

VELOCITY TRAVERSES IN DIFFUSER

ON HORIZONTAL AND VERTICAL DIAMETERS

TEST SECTION VELOCITY = 49.6 FT/SEC. UNLESS NOTED
VANE STAGGER = 96°
TOTAL ANGLE OF DIVERGENCE = 5°
KEY: Δ VERTICAL DIAMETER — O HORIZONTAL DIAMETER

Table I

| | | Test Sec Vel = 18.0 ft/sec | | Test Sec Vel = 30.0 ft/sec | | Test Sec Vel = 49.62 ft/sec | |
|---|---|---|---|---|---|---|---|
| | | Station 2 | Station 5 | Station 2 | Station 5 | Station 2 | Station 5 |
| Model | Vel (ft/sec) | 18.00 | 3.72 | 30.00 | 6.19 | 49.62 | 10.25 |
| | $\alpha$ | 1.0185 | 1.812 | 1.0144 | 1.812 | 1.0151 | 1.812 |
| | $\alpha \frac{\bar{V}^2}{2g}$ (in.) | 61.49 | 4.67 | 170.15 | 12.93 | 465.7 | 35.5 |
| | $Re = V \cdot \frac{d}{\nu}$ | $0.9 \times 10^6$ | $0.4 \times 10^6$ | $1.5 \times 10^6$ | $0.7 \times 10^6$ | $2.5 \times 10^6$ | $1.1 \times 10^6$ |
| | $\lambda$ | 0.0198 | 0.0178 | 0.0198 | 0.0172 | 0.0195 | 0.0170 |
| | $\frac{p_5}{w} - \frac{p_2}{w}$ (in.) | 52.07 | | 141.61 | | 387.7 | |
| | $\frac{p_\lambda}{w}$ (in.) | 2.95 | | 8.03 | | 20.6 | |
| | $\eta$ | 0.9164 | | 0.9007 | | 0.9012 | |
| | $\eta_o$ | 0.9683 | | 0.9518 | | 0.9490 | |
| Prototype | $\eta'$ | 0.9280 | | 0.9280 | | 0.9280 | |
| | $\eta'^*$ | 0.8780 | | 0.8780 | | 0.8780 | |

*With test installation in test section.

The total efficiency of energy conversion is defined in the following equation in which the subscripts represent the station numbers:

$$\eta = \frac{\dfrac{p_5 - p_2}{w}}{\dfrac{\alpha_2 V_2^2 - \alpha_5 V_5^2}{2g}}$$

The percentage energy loss is $(1 - \eta)\,100$. Values of $\eta$ for $V_2 = 18.0$, 30.0, and 49.62 ft per sec are tabulated in Table I, line 8.

In order to extrapolate these energy data from the model to the prototype, it is first necessary to determine the manner in which scale effects influence the diffuser flow. An analysis of previous investigations of such flow indicates that it is safe to consider the total energy loss as being principally constituted of a wall friction element and a form loss element, where the former is subject to the conventional pipe roughness scale effects and the latter is substantially constant.

An evaluation of the wall friction element is herein based on a suitable integration of the expression,

$$d\left(\frac{p_\lambda}{w}\right) = \lambda \frac{dL}{d} \frac{V^2}{2g}$$

The integration is accomplished by the following procedure with reference to Fig. 3:

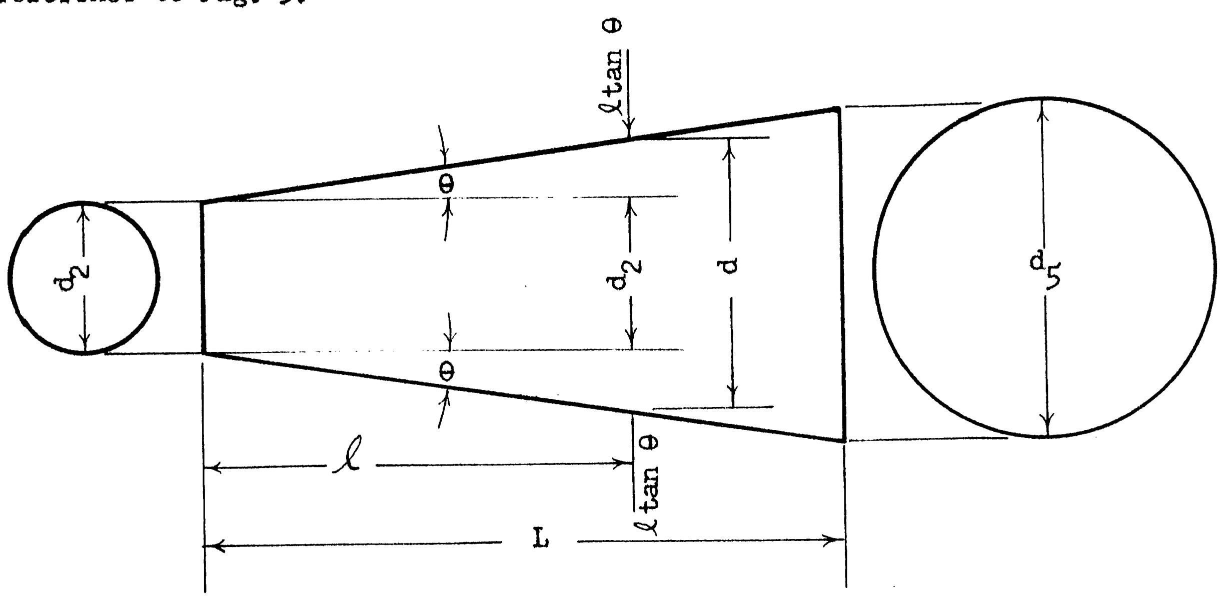



FIGURE 3

$$d = d_2 + 2\ell \tan\theta \;, \quad d\ell = \frac{d\,d}{2\tan\theta}$$

$$d\left(\frac{p_\lambda}{w}\right) = \lambda \cdot \frac{d\,d}{2\tan\theta \cdot d} \cdot \frac{V^2}{2g} \;, \quad V^2 = V_2^2 \cdot \frac{d_2^4}{d^4}$$

$$\frac{p_\lambda}{w} = \int_{d_2}^{d_5} \frac{\lambda V_2^2 \, d_2^4}{2\tan\theta \cdot 2g} \cdot \frac{d\,d}{5} = \frac{\lambda V_2^2 \, d_2^4}{2\tan\theta \cdot 2g} \left[\frac{1}{d_2^4} - \frac{1}{d_5^4}\right]$$

$$= \frac{\lambda}{8\tan\theta} \left[\frac{V_2^2 - V_5^2}{2g}\right]$$

for $V_2 = 6$ in., $V_5 = 13.2$ in., $\tan\theta = 2^\circ - 30'$

$$\frac{p_\lambda}{w} = 2.60\,\lambda\,\frac{V_2^2}{2g}$$

Conventional friction factor $\lambda$ was evaluated for the model from page 211 [3] using the Reynolds numbers shown in Table I, line 5 and considering the surface as galvanized iron. The resulting $\lambda$ values are shown in Table I, line 6, and the wall friction loss $\frac{p_\lambda}{w}$ on line 7.

The efficiency of form loss energy conversion can now be defined as:

$$\eta_o = \frac{\dfrac{p_5 - p_2}{w} + \dfrac{p_\lambda}{w}}{\dfrac{\alpha_2 \bar{V}_2^2 - \alpha_5 \bar{V}_5^2}{2g}}$$

The evaluation of $\eta_o$ for the test data is shown in Table I, line 9. It will be noted that $\eta_o$ varies somewhat for the three different test runs made. This variation in the value of $\eta_o$ is probably due to experimental error, since the form efficiency should tend to show a slight increase with increasing Reynolds number. The variation also may be due in part to the assumptions made in computing the wall friction losses. Actually, the friction drag in the retarded diffuser flow is probably somewhat different from that in a comparable section of straight pipe. In computing the efficiency of the prototype diffuser, a value of $\eta_o = 0.9500$ was assumed to apply. (The average of the three experimentally determined values of $\eta_o$ was 0.9564.)

The total efficiency $\eta'$ of the prototype energy conversion is then computed by correcting the above $\eta_o$ value as follows:

$$\eta' = \eta_o - \left[ \frac{\frac{p_\lambda}{w}}{\frac{\alpha_2 \bar{V}_2^2 - \alpha_5 \bar{V}_5^2}{2g}} \right]$$

In this expression, the last term is evaluated for the prototype conditions, assuming that the $\alpha$ values are substantially the same as for the model and that $\lambda = 0.0085$. The latter value was chosen for the prototype diffuser from measurements made by the Tennessee Valley Authority [4] on a tunnel of approximately the same size and surface. Values of $\eta'$ for the velocities used in the tests are shown in Table I, line 10.

It is of interest to compare the values of $\eta$ and $\eta_o$ with efficiencies for similar conical diffusers obtained by other investigators. In the original design report, the computed head loss for the prototype diffuser was $0.0803 \frac{\bar{V}_2^2}{2g}$, which would be the same as $\eta_1 = 0.9197$. Gibson [6] found $\eta = 0.865$ and $\eta_o = 0.955$ in a conical diffuser with $\theta = 2^o - 30'$. However, Gibson computed efficiencies assuming both initial and final values of $\alpha = 1.00$. This would seemingly prohibit making a comparison, since it can be seen that the computed $\alpha$ value, which can not be determined from Gibson's tests, is quite large at the downstream end of the tunnel diffuser.

Peters [7] carried out much more carefully controlled experiments, varying among other things the inlet velocity distribution by changing the length of a cylindrical pipe preceding the diffuser. For an inlet length of six diameters, as in the water tunnel, Peters found $\eta = 0.865$ and $\eta_o = 0.900$, as defined previously. It was also shown that the efficiency of the diffuser was lowered as the inlet length was increased.

Comparison of the values of $\eta$ and $\eta_o$, as determined in the model (Table I) with Peters' values, will indicate a somewhat greater efficiency in the model. The reason for this improvement is not apparent, but, as Peters points out in his paper, experimental determinations of efficiencies of small angle diffusers vary greatly. Values of $\eta_o$ varying from 0.89 to 0.97 have been found by various experimenters for $\theta$ in the neighborhood of $2^o - 30'$.

As mentioned before, inflow velocity distributions in the prototype will vary greatly, depending on the test installation. Accordingly, efficiency and head loss through the diffuser will vary greatly. In Peters' experiments, $\eta_o$ varied from 0.95 for no inlet length (uniform velocity distribution) to 0.90 for an inlet length of 6 diameters, and to 0.85 for an inlet length of 27 diameters (normal turbulent profile). However, Peters also proved that this drop in efficiency could be made much smaller by placing a length of straight conduit at the discharge end of the diffuser. It was shown that the pressure regain was not fully accomplished in the diffuser; the pressure continued to rise in the section of straight pipe for a distance of two to six diameters, depending on the inlet velocity distribution. The reason for this pressure rise is that the velocity distribution changes from a distorted profile at the exit of the diffuser to a normal turbulent profile in the conduit. As this change in velocity distribution is accomplished, the mean kinetic energy of the flow is decreased, that is, the value of $\alpha$ decreases. Consequently, a rise in pressure results if the friction and conversion losses are small enough. Just what energy will be lost in the conversion from kinetic to pressure energy is problematical. In all probability, it will be a major part of the total interchange of energy, since conversions of this type are inefficient. However, since the section of the tunnel following the diffuser is not treated in this section of the report, the final conversion loss will be assumed to be due to the vaned elbow following the diffuser, although in reality it is caused by the diffuser.

Estimating the energy loss in the diffuser with a model installed in the test section is a rather difficult task. As a conservative estimate based on Peters' work, it may be assumed that the form loss, $1 - \eta_o$, is doubled. Accordingly, values of $\eta'$ for a model installation are calculated and recorded in Table I, line 11. Also, it might be noted here that if the test model imparts appreciable rotation to the stream leaving the test section, the energy loss in the diffuser will be lessened.

## IV. CONCLUSIONS

On the basis of these experimental investigations, the following conclusions concerning the diffuser of the 60-in. prototype tunnel are drawn:

1. The $5^{\circ}$ diffuser as designed does not operate entirely satisfactorily in regard to velocity distribution. Some separation

occurs at the bottom of the diffuser (on the inside of the tunnel loop) near the downstream end. It is not considered advisable for two reasons to lessen the tendency for separation by making the angle of divergence smaller: (a) The magnitude and detrimental effect of the separation will depend on the model installation in the test section, and (b) the physical dimensions and cost of the water tunnel will be greatly increased. These tests do not permit an accurate evaluation of the velocity distribution and energy loss in support of the possible use of a diffuser of greater cone angle.

2. The energy loss in the diffuser is satisfactorily low in comparison with values of efficiency of energy conversion as determined by previous investigators. It is estimated that the efficiency of the diffuser will be 0.878 with a test body installed in the test section and 0.928 with no test body installed in the test section.

Cavitation indexes and performance characteristics of the entrance boundary transition are considered more properly evaluated as a part of the test section studies and are contained in the report concerning that component, PART III, TEST SECTION STUDIES.

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