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CONTENTS

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SOME FUNDAMENTAL CONCEPTS OF INCOMPRESSIBLE FLUID MECHANICS

PARTS I AND II

BY P. S. EAGLESON*

(Presented as two of the John R. Freeman Lectures on Fundamental Hydraulic Processes in Water Resources Engineering, Boston Society of Civil Engineers, October 1 and 8, 1963.)

INTRODUCTION

The purpose of this lecture series as well as the accompanying notes is to present current thinking and recent research results in certain specialized fields of hydraulics. These particular paragraphs on fundamentals will therefore be restricted to what is hoped is a logical development of some of the background material to be drawn upon later. They make no attempt to avoid mathematics and assume a knowledge of basic hydraulics and the operations of partial differentiation.

I. FORMULATION OF BASIC EQUATIONS

Three basic laws which we bring to bear in the solution of problems of incompressible fluid mechanics are:

a. Conservation of mass.
b. Conservation of momentum.
c. Conservation of energy.

These laws can be formulated in two ways:

a. By focusing our attention on an imaginary closed surface in the fluid field (called a control surface) and considering the net

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flux (rate of flow) of mass momentum or energy across this surface. This will be called the control volume approach.

b. By identifying a particular mass particle of infinitesimal size and considering what happens to it in terms of time and space coordinates. This will be called the differential approach.

When the control volume is of differential size, the two methods become indistinguishable. The relative advantage of either approach must thus lie with the scale of the information desired. When changes in average flow properties between two locations are desired with no concern for the details of the flow in between, then the control volume approach is ideal. When the spatial distributions of flow properties are being sought, the differential approach is required.

Coordinate System

The coordinate system chosen for these developments is shown in Fig. 1.1. Unless specified otherwise, x is horizontal, z is vertical

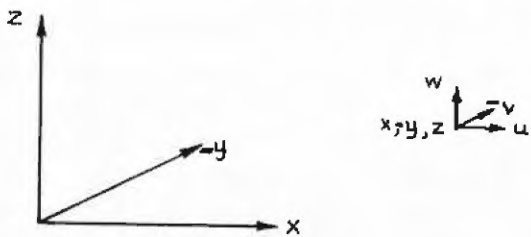

FIG. 1.1 COORDINATE SYSTEM

(positive upward) and y is perpendicular to the other two. At some instant of time, t, a fluid particle at x, y, z has the velocity components u, v, w, as is also shown in Fig. 1.1.

A. *Control Volume Approach*

Notation—

$\forall$ = control volume, ft^3
$d\forall$ = differential volume element, ft^3
A = control surface area, ft^2
dA = differential area element, ft^2

n = unit vector perpendicular to dA, positive outward
S = an extensive fluid property, i.e., some property possessed by each fluid element such as mass, linear momentum, energy, entropy, angular momentum, etc.
s = density of S or S per ft^3.

Let us consider a collection of particular fluid particles, those which completely fill the fixed boundaries of the control volume at time t_o. We will call this ensemble of particles the fluid system. If the fluid is flowing, the system and its cargo, S_s, will be convected across the control surface to some new position at later time, $t = t_o + \Delta t$. (Note: $S_{s_{t_o}} = S_{\mp}$). The cargo of S carried by the system may change with time, first due to convection into a new location (a purely spatial change) and second due to some inherent unsteadiness in the flow (a purely local change).

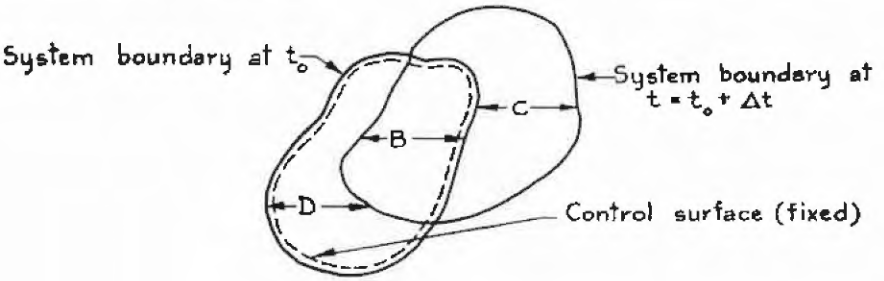

FIG. 1.2 CONTROL SURFACE AND SYSTEM BOUNDARIES

Referring to Fig. 1.2, the total time rate of change of system cargo may then be written:

$$\frac{dS_s}{dt} = \frac{S_{B_t} + S_{C_t} - S_{B_{t_o}} - S_{D_{t_o}}}{\Delta t}$$

or

$$\frac{dS_s}{dt} = \frac{S_{B_t} - S_{B_{t_o}}}{\Delta t} + \frac{S_{C_t} - S_{D_{t_o}}}{\Delta t}$$

Letting $\Delta t \to O$, $S_s \to S_{\mp}$, and

$$\frac{dS_s}{dt} = \frac{dS_{\mathcal{V}}}{dt} \equiv \frac{DS_{\mathcal{V}}}{Dt} = \left.\frac{\partial S_{\mathcal{V}}}{\partial t}\right|_{t_0} + \text{net outflow rate of S} \tag{1.01}$$

The symbol D/Dt is often referred to as the total, substantial or material derivative and represents the total rate of change of some quantity experienced by the particular system of masses under consideration as the system moves through the control volume. The total amount of S within the control volume at any time is given by:

$$S_{\mathcal{V}} = \int_{\mathcal{V}} s \, d\mathcal{V}$$

and the time rate of change of $S_{\mathcal{V}}$ by

$$\frac{\partial S_{\mathcal{V}}}{\partial t} = \frac{\partial}{\partial t} \int_{\mathcal{V}} s \, d\mathcal{V} \tag{1.02}$$

To evaluate the net rate of outflow of S from the control volume we may consider an element of surface area and the vector of the convecting velocity, q, at that point. Velocities directed outward from the control volume will be positive and inward will be negative.

Only the normal component, q_n, is effective in convecting S out through the control surface and thus the net rate of outflow of volume is

$$\int_A q_n \, dA$$

and the net rate of outflow of S is

$$\int_A s \, q_n \, dA \tag{1.03}$$

Thus in summary we can say:

Time rate of change of S for the system of masses =
Time rate of change of S within the control volume +
Net rate of outflow of S across the control surface.

Analytically

$$\frac{DS_s}{Dt} = \frac{\partial}{\partial t} \int_{\mathcal{V}} s \, d\mathcal{V} + \int_A s \, q_n \, dA \tag{1.04}$$

It should be noted that the quantity, s, may be a scalar such as energy or mass or a vector such as momentum as long as it depends upon the amount of fluid present.

Conservation of Mass

An example of the application of Eq. (1.04) is that in which the property, S, is mass. In this case $s = \rho$, the mass per unit volume.

Since system mass is to be conserved

$$\frac{DS_s}{Dt} = 0 = \frac{\partial}{\partial t}\int_{\forall} \rho \, d\forall + \int_A \rho \, q_n \, dA \qquad (1.05)$$

Green's theorem tells us that

$$\int_A \rho \, q_n \, dA = \int_{\forall} \left[\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} \right] d\forall$$

from which Eq. (1.05) can be written:

$$\int_{\forall} \left[\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} \right] d\forall = 0 \qquad (1.06)$$

As a simple example of these ideas consider the one dimensional steady flow of Fig. 1.3 in which the control volume is chosen so that

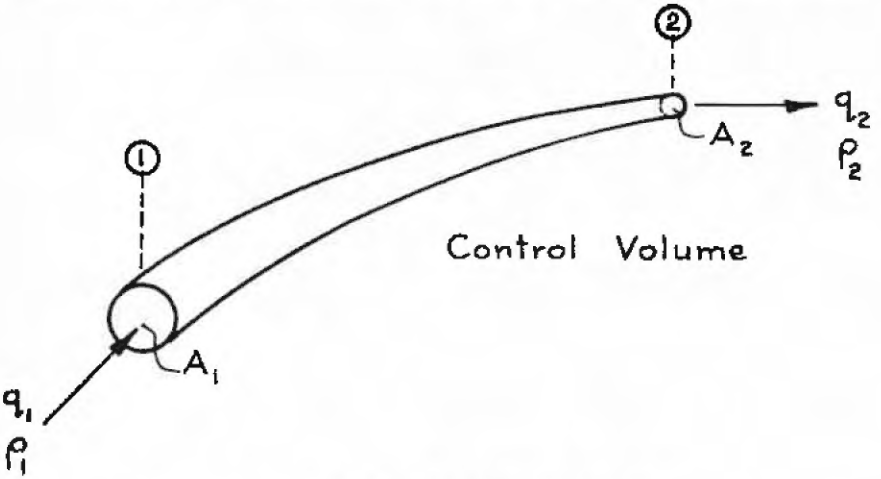

FIG. 1.3 STREAM TUBE

all elements of the lateral boundaries are streamlines for all time.

For this case Eq. (1.05) becomes:

$$\int_A \rho\, q_n\, dA = 0$$

or

$$\int_{A_1} \rho_1\, q_{1_n}\, dA + \int_{\text{side walls}} \rho\, q_n\, dA + \int_{A_2} \rho_2\, q_{2_n}\, dA = 0 \qquad (1.07)$$

Since the side walls are streamlines the middle integral of Eq. (1.07) is zero and we have

$$\int_{A_1} \rho_1\, q_1 \cos \alpha_1\, dA + \int_{A_2} \rho_2\, q_2 \cos \alpha_2\, dA = 0 \qquad (1.08)$$

Under the assumption of one-dimensional flow ρ, $\cos \alpha$ and q are constant across each end section, thus

$$\rho_1 V_1 \cos \alpha_1 \int_{A_1} dA + \rho_2 V_2 \cos \alpha_2 \int_{A_2} dA = 0 \qquad (1.09)$$

If areas A_1 and A_2 are selected so as to be cross-sectional areas (i.e., normal to the mean velocities V_1 and V_2) we have

$$\cos \alpha_2 \int_{A_2} dA = A_2$$

and

$$\cos \alpha_1 \int_{A_1} dA = A_1$$

in which case we obtain the steady, one-dimensional equation of continuity applicable to either compressible or incompressible flows

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \qquad (1.10)$$

The applicability of Eq. (1.10) is not restricted to truly one-dimensional flows, however, as long as ρ_1 and ρ_2 are constant across their respective sections and with the understanding that V_1 and V_2 are the components of the average velocity perpendicular to plane areas A_1 and A_2 respectively. For incompressible flows in which the lateral control surfaces are rigid boundaries Eq. (1.10) becomes

$$V_1 A_1 = V_2 A_2$$

which is applicable to steady or unsteady flow.

Conservation of Energy

Let us now consider the case in which S represents another scalar quantity, the fluid energy. Then

$$s = e\rho = \text{energy per unit volume}$$

in which e = energy per unit of mass. Eq. (1.04) then becomes

$$\frac{DE}{Dt} = \frac{\partial}{\partial t}\int_{\mathcal{V}} e\rho \, d\mathcal{V} + \int_A e\rho \, q_n \, dA \tag{1.11}$$

We know that energy can enter or leave the system either in the form of heat or of work. If we let Q be the net heat added to the system and W be the net work done by the system on its surroundings, we have from the First Law of thermodynamics and Eq. (1.11) that:

$$\frac{DE}{Dt} = \frac{\delta Q}{dt} - \frac{\delta W}{dt} = \frac{\partial}{\partial t}\int_{\mathcal{V}} e\rho \, d\mathcal{V} + \int_A e\rho \, q_n \, dA \tag{1.12}$$

in which the symbol δ is used to indicate incremental amounts of items which are not system properties.

Looking first at the term for rate of doing net work on the surroundings, $\delta W/dt$:

Work can be done by the system on its surroundings only at the control surface:

a. where fluid contacts fluid, and
b. where fluid contacts solid.

Remember that the definition of work requires not merely that a force be present at the boundary, but also that this force move through a distance. In other words, the boundary must be in motion under the application of a force which can be resolved into components normal or tangential to the boundary surface.

The normal differential force on an element of the control surface is given in terms of the pressure intensity, p, by

$$dF_p = p \, dA \tag{1.13}$$

and the rate at which work is done by these stresses is

$$\left.\frac{\delta W}{dt}\right|_p = \int_A p \, q_n \, dA = \int_A \rho\left(\frac{p}{\rho}\right) q_n \, dA \tag{1.14}$$

The tangential differential force on an element of the control surface is given in terms of the shearing stress, τ, by

$$dF_\tau = \tau \, dA \tag{1.15}$$

and the rate at which work is done by these stresses is

$$\left.\frac{\delta W}{dt}\right|_\tau = \int_A \tau \, q_t \, dA \tag{1.16}$$

For the time being at least, due to the difficulty of evaluating this term, we will refer to it simply as the "shear power," P_{shear}.

If there are one or more rotating mechanical elements comprising a portion of the control surface (pumps, turbines, fans, compressors, etc.) it is convenient to exclude from Eqs. (1.14) and (1.16) those portions of the normal or tangential work which are performed on these elements (since they are excluded from the control volume they become part of the surroundings) and lump them separately in a term called "shaft power," P_{shaft}.

Thus:

$$\frac{\delta W}{dt} = P_{shaft} + P_{shear} + \int_A \rho\left(\frac{p}{\rho}\right) q_n \, dA \tag{1.17}$$

Looking next at the right hand side of Eq. (1.12) we must consider what forms of energy are to be included:

a. kinetic energy $= E_k = \dfrac{mq^2}{2}$

or $e_k = \dfrac{q^2}{2}$ kinetic energy per unit mass,

b. gravitational potential energy $= E_g = mgz$
or $e_g = gz =$ gravitational potential energy per unit mass,

c. internal energy $= E_{U_*} = m\,u_*$
or $e_{U_*} = u_*$, internal energy per unit mass.

We will not consider here those energies due to electrical and magnetic fields, surface energies, etc. Thus

$$e = \frac{q^2}{2} + gz + u_* \tag{1.18}$$

and then

$$\frac{\delta Q}{dt} - P_{shaft} - P_{shear} = \frac{\partial}{\partial t}\int_{\forall} e\rho d\forall + \int_{A} \rho\left(\frac{q^2}{2} + gz + u_* + \frac{p}{\rho}\right) q_n \, dA \quad (1.19)$$

Steady, One-Dimensional Flow - -

If we make the following simplifying assumptions:

a. Flow is steady, i.e., $\frac{\partial}{\partial t}\int_{\forall} e\rho d\forall = 0$.

b. Control surface coincides with solid, stationary boundaries or crosses the flow at right angles to the stream lines in zones of uniform flow where ρ and u_* are constant and where the pressure is "hydrostatically" distributed.

Then —

$$P_{shear} = 0$$

and

$$\int_{A} \rho\left(\frac{q^2}{2} + gz + u_* + \frac{p}{\rho}\right) q_n \, dA = \left[\left(k_e \frac{V^2}{2} + g\bar{z} + u_* + \frac{p}{\rho}\right)\rho V A'\right]_{OUTFLOW} - \left[\left(k_e \frac{V^2}{2} + g\bar{z} + u_* + \frac{p}{\rho}\right)\rho V A'\right]_{INFLOW}$$

in which

$$k_e = \frac{\int_{A'} \rho \, q^3 \, dA'}{\rho \, V^3 \, A'}.$$

Letting

$Q' =$ mass flow rate $= \rho V A'$,

$h =$ enthalpy per unit mass $= u_* + \frac{p}{\rho}$,

$q' = \frac{\delta Q}{dt}$,

$k_e = 1$,

and indicating the outflow and inflow sections by subscripts 2 and 1 respectively, Eq. (1.19) may be written:

$$q' = P_{shaft} + \left[h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(\bar{z}_2 - \bar{z}_1) \right] Q' \tag{1.20}$$

To better appreciate the significance of this simple form of the general energy equation let us look at some specific cases.

a. Suppose the shaft work is zero and we have a freely-falling jet of fluid. If the fluid is real we know that the mechanical energies as represented by $V^2/2$ and $g\bar{z}$ are not conserved. That is all of the loss of potential energy does not appear in increased kinetic energy. The missing energy has been degraded and will appear as a change in enthalpy, h and/or in heat transfer, q'.

b. Again for no shaft work if the fluid is real and incompressible and if the temperature is maintained a constant, the effects of fluid friction appear as a heat transfer, q'. This can be easily seen by considering flow in a horizontal conduit of uniform section. Since the internal energy u_* depends solely on temperature, Eq. (1.20) reduces to

$$q' = \left[\frac{p_2}{\rho} - \frac{p_1}{\rho} \right] Q'$$

or, more generally, for real, incompressible flows with no shaft work

$$\Delta_{1\,2} \left[\frac{p}{\rho} + \frac{V^2}{2} + g\bar{z} \right] Q' = \text{Rate of energy "loss"} = Q' \, \Delta u_* - q'$$

c. When the rate of energy "loss" is zero

$$q' = Q' \, \Delta u_*$$

and

$$\frac{p}{\rho} + g\bar{z} + \frac{V^2}{2} = \text{constant}$$

which is the familiar equation of Bernoulli.

Conservation of Momentum

We will now consider the case in which S represents a vector quantity, the linear fluid momentum, P. Then

$$\frac{s}{\rho} = \frac{P}{m} = q = \text{linear momentum per unit mass or velocity}$$

Equation (1.04) is then a vector equation:

$$\left[\frac{DP}{Dt}\right]_i = \frac{\partial}{\partial t}\int_{\forall} \rho\, q_i\, d\forall + \int_A \rho\, q_i\, q_n\, dA \qquad (1.21)$$

in which the subscript, i, refers to a particular coordinate direction.

We can expand the left hand side of Eq. (1.21) in terms of Newton's second law of motion which states, for a fixed (i.e. inertial) frame of reference:

$$F_i = \left(\frac{DP}{Dt}\right)_i \qquad (1.22)$$

or, in words — The resultant external force acting on a system of masses is equal to the total rate of change of linear momentum of the system.

Equation (1.21) then becomes

$$F_i \frac{\partial}{\partial t}\int_{\forall} \rho\, q_i\, d\forall + \int_A \rho\, q_i\, q_n\, dA \qquad (1.23)$$

Looking again at the stream tube of Fig. 1.3 we can write for one-dimensional, steady flow

$$F_i = \rho_1 V_1 V_{1_1} \cos\alpha_1 \int_{A_1} dA + \rho_2 V_2 V_{1_2} \cos\alpha_2 \int_{A_2} dA \qquad (1.24)$$

If A_1 and A_2 are selected normal to the mean velocities V_1 and V_2 and the density is uniform, Eq. (1.24) reduces to the familiar form

$$F_i = \rho V_2 A_2 V_{2_i} - \rho V_1 A_1 V_{1_i} = \rho VA\ (\Delta V_i) \qquad (1.25)$$

Another and independent vector equation similar to Eq. (1.21) can be written for conservation of angular momentum. For a discussion of this case see Shames (4).

B. *Differential Approach*

In order to study the details of the fluid motion at some point in space we will consider control volumes and mass elements of differential size.

Taylor's Series

If some fluid characteristic, f, such as velocity, pressure, temperature, density, etc., is known at point (x_o, y_o, z_o) and all spatial deriva-

tives of f are known at this point, then the value of f at some new point can be determined. As an example, considering a variation, Δx, in the x coordinate direction, we can use Taylor's series to write

$$f(x_0 + \Delta x, y_0, z_0) = f(x_0, y_0, z_0) + \left.\frac{\partial f(x, y, z)}{\partial x}\right|_{(x_0, y_0, z_0)} \frac{\Delta x}{1!} + \left.\frac{\partial^2 f(x, y, z)}{\partial x^2}\right|_{(x_0, y_0, z_0)} \frac{(\Delta x)^2}{2!} + \ldots \quad (1.26)$$

One important characteristic of this equation is that as we let Δx become very small, the quantities $(\Delta x)^2$, $(\Delta x)^3$, . . . become even smaller and Eq. (1.01) can be approximated by:

$$f(x_0 + \Delta x, y_0, z_0) = f(x_0, y_0, z_0) + \left.\frac{\partial f(x, y, z)}{\partial x}\right|_{(x_0, y_0, z_0)} \Delta x \quad (1.27)$$

Conservation of Mass

Consider the small rectangular parallelepiped with sides of length Δx, Δy and Δz as shown in Fig. 1.4. Conservation of mass requires that the net mass of fluid flowing across the boundaries into the volume element in a certain time, Δt, be equal to the amount by which the mass of the element has increased in the same Δt.

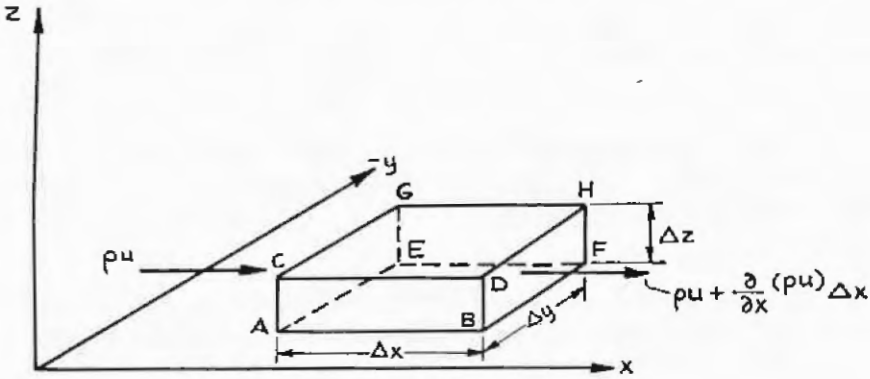

FIG. 1.4 FLOW INTO ELEMENT

The inflow of mass across face AECG in time Δt is

$$\rho u \Delta y \Delta z \Delta t$$

where ρ is the mass density of the fluid.

According to Taylor's series, the inflow of mass across the opposing face BFDH, in time Δt is

$$-\left[\rho u\, \Delta y\, \Delta z\, \Delta t + \frac{\partial(\rho u)}{\partial x}\, \Delta x\, \Delta y\, \Delta z\, \Delta t\right].$$

Adding the above two expressions yields the net inflow of mass in the x-direction during Δt:

$$-\frac{\partial(\rho u)}{\partial x}\, \Delta x\, \Delta y\, \Delta z\, \Delta t.$$

In a similar fashion it can be shown that the net inflow of mass during Δt in the y and z directions is respectively

$$-\frac{\partial(\rho v)}{\partial y}\, \Delta x\, \Delta y\, \Delta z\, \Delta t$$

$$-\frac{\partial(\rho w)}{\partial z}\, \Delta x\, \Delta y\, \Delta z\, \Delta t$$

The net inflow of mass into the volume element is the sum of the contributions of the three coordinate directions, i.e.:

Net inflow of mass across all faces in Δt

$$= \left[-\frac{\partial(\rho u)}{\partial x} - \frac{\partial(\rho v)}{\partial y} - \frac{\partial(\rho w)}{\partial z}\right] \Delta x\, \Delta y \Delta z\, \Delta t \qquad (1.28)$$

If the mass present at time t is $\rho \Delta x \Delta y \Delta z$, then at time $t + \Delta t$, according to Taylor's formula, the mass present will be

$$\rho \Delta x \Delta y \Delta z + \frac{\partial}{\partial t}(\rho \Delta x \Delta y \Delta z)\, \Delta t.$$

The net increase is thus:

$$\frac{\partial \rho}{\partial t}\, \Delta x \Delta y \Delta z \Delta t \qquad (1.29)$$

In the absence of any creation or destruction of mass within the

volume element, this must be equal to the inflow of mass across the boundaries, i.e., Eq. (1.28) = Eq. (1.29), or

$$\frac{\partial \rho}{\partial t} = -\left[\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z}\right] \tag{1.30}$$

which is seen to be the differential form of Eq. (1.06).

The right hand side of this equation can be expanded to yield

$$\frac{\partial \rho}{\partial t} + u\frac{\partial \rho}{\partial x} + v\frac{\partial \rho}{\partial y} + w\frac{\partial \rho}{\partial z} + \rho\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) = 0 \tag{1.31}$$

Examining the first four terms of Eq. (1.31) we recognize the expanded form of the material derivative of the density, $D\rho/Dt$, in which

$$\frac{D}{Dt} \equiv \underbrace{\frac{\partial}{\partial t}}_{\text{local term}} + \underbrace{u\frac{\partial}{\partial x} + v\frac{\partial}{\partial y} + w\frac{\partial}{\partial z}}_{\text{convective terms}} \tag{1.32}$$

We can now see clearly that when focusing our attention on a particular particle we must consider the rate of increase of a certain quantity due to two effects:

(1) a "local" effect independent of the motion of the particle. This effect is the rate of change that a motionless particle would experience at a certain point.

(2) a "convective" effect which is the rate of change of the property due to the particle moving in a field where gradients of the property exist.

A simplified case which may clarify the physical significance of these effects is that of a person traveling across country by car. Consider that throughout the area there is a uniform increase in the mean daily temperature of 1 degree/day. This is the "local" effect and a person would experience this rate of temperature change even if he were not moving. In addition, suppose there is a mean daily temperature increase of 1 degree/1000 miles in the positive x direction. If the car is traveling in the x direction with velocity u then the *total* rate of increase experienced by the car would be

$$\frac{DT}{Dt} = \frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} = 1 + \frac{u}{1000}$$

where T is the mean daily temperature. Furthermore if the rate of travel, u, is 500 miles/day in the positive x direction, the total rate of increase experienced by the car is

$$\frac{DT}{Dt} = 1 + \frac{500}{1000} = 1.5 \text{ degrees/day}$$

Conversely, if one were traveling 1000 miles/day in the negative x direction, the total rate of temperature increase would be:

$$\frac{DT}{Dt} = 1 + (-1000)\frac{1}{1000} = 0$$

or the two effects would exactly compensate and one would experience no increase in the mean daily temperature.

Granted, now that D/Dt of a quantity does represent the total derivative or total rate of increase of that particular quantity experienced by a certain moving particle, let us return and express Eq. (1.04) in the form:

$$\frac{D\rho}{Dt} + \rho\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) = 0 \qquad (1.33)$$

which is the continuity equation for a compressible or incompressible fluid in steady or unsteady motion. If we further idealize for the case of an incompressible fluid, the mass density of a particle must be constant, i.e., $D\rho/Dt = 0$ and hence Eq. (1.08) becomes:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \qquad (1.34)$$

Conservation of Momentum

When working in differential form, the momentum equations are very often called equations of motion. These equations will be derived for the case of a real (viscous) fluid of constant density by considering all of the forces which can act on the element of fluid shown in Fig. 1.5. Two types of forces must be considered, surface forces and body forces.

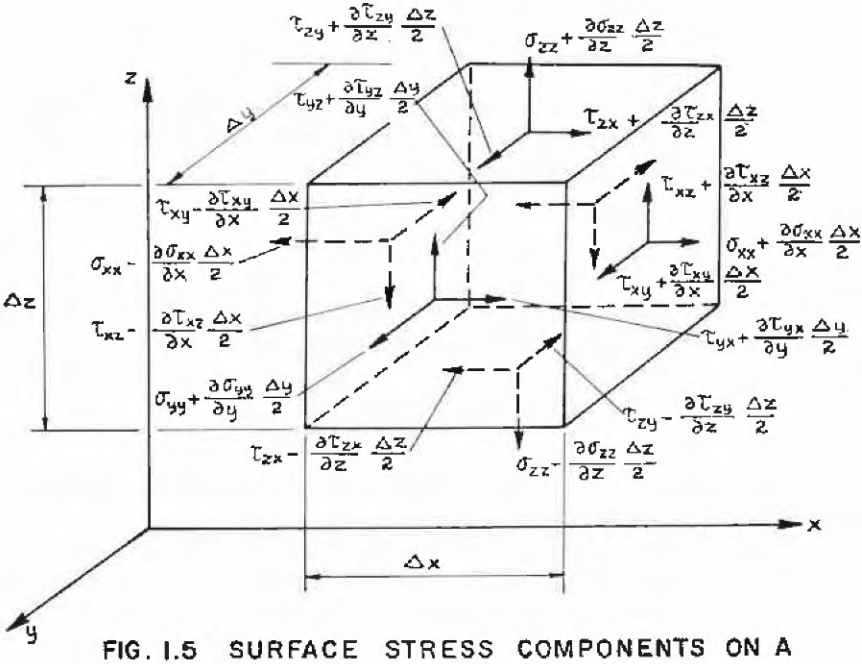

FIG. 1.5 SURFACE STRESS COMPONENTS ON A VOLUME ELEMENT

1. Surface forces—Any surface force may be resolved into components perpendicular and tangent to the surface. In terms of force per unit area, these become the direct stress, σ, and shear stress, τ, respectively.

In this development each stress component will bear two subscripts. The first defines the axis which is perpendicular to the element face upon which the stress acts, the second defines the coordinate direction in which the force produced by the stress acts. The sense of the stress is positive if both of these directions are positive or negative and negative if either one is negative. There are nine of these stress components.

2. Body forces—A body force depends only upon the bulk or mass of fluid included in the volume element. Gravity is the only body force to be considered in this study.

The nine surface stress components can be written as indicated in Fig. 1.5. The net normal force on the element in the positive x direction is thus

$$\left[\left(\sigma_{xx}+\frac{\partial\sigma_{xx}}{\partial x}\frac{\Delta x}{2}\right)-\left(\sigma_{xx}-\frac{\partial\sigma_{xx}}{\partial x}\frac{\Delta x}{2}\right)\right]\Delta y\Delta z=\frac{\partial\sigma_{xx}}{\partial x}\Delta x\Delta y\Delta z$$

and the net tangential force in the positive x direction is

$$\left[\left(\tau_{yx}+\frac{\partial\tau_{yx}}{\partial y}\frac{\Delta y}{2}\right)-\left(\tau_{yx}-\frac{\partial\tau_{yx}}{\partial y}\frac{\Delta y}{2}\right)\right]\Delta z\Delta x$$
$$+\left[\left(\tau_{zx}+\frac{\partial\tau_{zx}}{\partial z}\frac{\Delta z}{2}\right)-\left(\tau_{zx}-\frac{\partial\tau_{zx}}{\partial z}\frac{\Delta z}{2}\right)\right]\Delta x\Delta y$$
$$=\left(\frac{\partial\tau_{yx}}{\partial y}+\frac{\partial\tau_{zx}}{\partial z}\right)\Delta x\Delta y\Delta z$$

Designating X as the x component of the body force per unit of volume, the equation of motion in the x-direction can then be written:

$$d\,F_x=\left(\frac{DP}{Dt}\right)_x=\rho\cdot\frac{Du}{Dt}$$

$$\text{surface forces}+\text{body forces}=\text{mass}\cdot\text{acceleration}$$

or

$$\frac{\partial\tau_{yx}}{\partial y}+\frac{\partial\tau_{zx}}{\partial z}+\frac{\partial\sigma_{xx}}{\partial x}+X=\rho\frac{Du}{Dt}$$

and similarly for the other coordinate directions

$$\frac{\partial\tau_{xy}}{\partial x}+\frac{\partial\tau_{zy}}{\partial z}+\frac{\partial\sigma_{yy}}{\partial y}+Y=\rho\frac{Dv}{Dt} \qquad (1.35)$$

and

$$\frac{\partial\tau_{xz}}{\partial x}+\frac{\partial\tau_{yz}}{\partial y}+\frac{\partial\sigma_{zz}}{\partial z}+Z=\rho\frac{Dw}{Dt}$$

Stresses and Strains

In a fashion similar to the above we may write equations of angular motion about coordinate axes through the mass center of the volume element. For example, adding moments about an axis parallel to the x axis and equating them to the moment of inertia times the angular acceleration, $\dot{\alpha}$, around this axis

$$\left[\left(\tau_{zy}+\frac{\partial\tau_{zy}}{\partial z}\frac{\Delta z}{2}\right)+\left(\tau_{zy}-\frac{\partial\tau_{zy}}{\partial z}\frac{\Delta z}{2}\right)\right]\Delta x\Delta y\frac{\Delta z}{2}$$
$$+\left[\left(\tau_{yz}+\frac{\partial\tau_{yz}}{\partial y}\frac{\Delta y}{2}\right)+\left(\tau_{yz}-\frac{\partial\tau_{yz}}{\partial y}\frac{\Delta y}{2}\right)\right]\Delta z\Delta x\frac{\Delta y}{2}$$
$$=\rho\,\Delta x\Delta y\Delta z\,\frac{(\Delta y^2+\Delta z^2)}{12}\,\dot{\alpha} \qquad (1.36)$$

In the limit, as Δx, Δy and Δz are shrunk to zero, the right hand side of Eq. (1.36) must vanish and we have

$$\tau_{yz}=\tau_{zy}$$

Repeating this about the other centroidal axes yields

$$\tau_{yx}=\tau_{xy} \qquad (1.37)$$

and

$$\tau_{xz}=\tau_{zx}$$

which reduces the number of scalar surface stress components to six

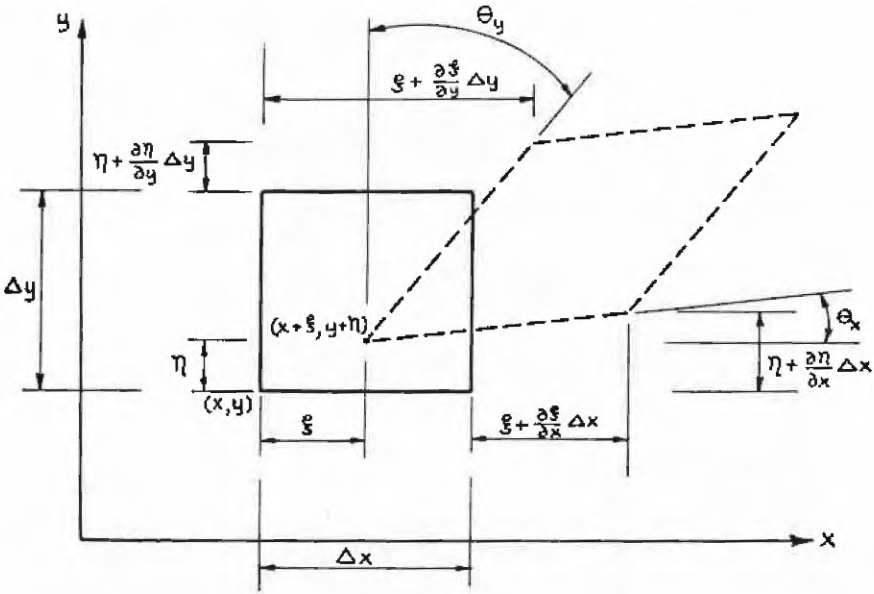

FIG. 1.6 PLANE STRAIN AND DEFORMATION

The negative average of the three direct stresses is that quantity which we know as fluid pressure, p. Thus

$$p = -\bar{\sigma} = -\frac{1}{3}(\sigma_{xx} + \sigma_{yy} + \sigma_{zz}) \tag{1.38}$$

Stress-Strain Relations

When stresses are applied to a volume element of any continuous medium, whether it be solid, liquid or gas, the element will deform. Let the coordinates of a point before deformation be given by x, y, z and after deformation by $x + \xi$, $y + \eta$, $z + \zeta$. This is illustrated for two dimensions in Fig. 1.6.

Defining normal strain as the change in length of a side of the element divided by its original length, the normal strain component, ε_x, can be written (see Fig. 1.6):

$$\left.\begin{aligned} \varepsilon_x &= \frac{\xi + \dfrac{\partial \xi}{\partial x}\Delta x - \xi}{\Delta x} = \frac{\partial \xi}{\partial x}. \\ &\text{Similarly} \\ \varepsilon_y &= \frac{\partial \eta}{\partial y} \\ &\text{and} \\ \varepsilon_z &= \frac{\partial \zeta}{\partial z} \end{aligned}\right\} \tag{1.39}$$

The volumetric strain (dilatation), e, is then given by:

$$e = \varepsilon_x + \varepsilon_y + \varepsilon_z \tag{1.40}$$

Defining shear strain as the change in angle between pairs of axes perpendicular to surfaces of the undeformed element, the shear strain component, γ_{xy}, can be written (see Fig. 1.6):

$$\left.\begin{aligned} \gamma_{xy} &= \theta_x + \theta_y = \frac{\dfrac{\partial \eta}{\partial x}\Delta x}{\Delta x} + \frac{\dfrac{\partial \xi}{\partial y}\Delta y}{\Delta y} = \frac{\partial \eta}{\partial x} + \frac{\partial \xi}{\partial y} \\ &\text{Similarly} \\ \gamma_{yz} &= \frac{\partial \zeta}{\partial y} + \frac{\partial \eta}{\partial z} \\ \gamma_{zx} &= \frac{\partial \xi}{\partial z} + \frac{\partial \zeta}{\partial x} \end{aligned}\right\} \tag{1.41}$$

Elastic Solid

When the medium is an elastic solid, we know from experiment that Hooke's law provides a linear proportionality between stress and the magnitude of the strain. This proportionality is defined by the shear modulus of elasticity, G, the bulk modulus of elasticity, E, and their interrelationship through the Poisson ratio, n, where

$$n = \frac{\text{unit strain in direction normal to applied force}}{\text{unit strain in direction of applied force}} = \frac{E}{2G} - 1 \tag{1.42}$$

Using these relations we can write, for an isotropic elastic solid:

$$\left.\begin{aligned} \varepsilon_x = \frac{\sigma_{xx}}{E} - n\frac{\sigma_{yy}}{E} - n\frac{\sigma_{zz}}{E} &= \frac{1}{E}\left[\sigma_{xx} - n(\sigma_{yy} + \sigma_{zz})\right] \\ \varepsilon_y &= \frac{1}{E}\left[\sigma_{yy} - n(\sigma_{zz} + \sigma_{xx})\right] \\ \varepsilon_z &= \frac{1}{E}\left[\sigma_{zz} - n(\sigma_{xx} + \sigma_{yy})\right] \end{aligned}\right\} \tag{1.43}$$

and

$$\left.\begin{aligned} \gamma_{xy} &= \frac{\tau_{xy}}{G} \\ \gamma_{yz} &= \frac{\tau_{yz}}{G} \\ \gamma_{zx} &= \frac{\tau_{zx}}{G} \end{aligned}\right\} \tag{1.44}$$

From Eqs. (1.40) and (1.43) the volumetric strain becomes

$$e = \frac{1 - 2n}{E}(\sigma_{xx} + \sigma_{yy} + \sigma_{zz}) \tag{1.45}$$

Equations (1.38), (1.42), (1.43), (1.44) and (1.45) can now be combined to give

$$\left.\begin{aligned} \sigma_{xx} - \bar{\sigma} &= 2G\left[\frac{\partial\xi}{\partial x} - \frac{1}{3}\left(\frac{\partial\xi}{\partial x} + \frac{\partial\eta}{\partial y} + \frac{\partial\zeta}{\partial z}\right)\right] \\ \sigma_{yy} - \bar{\sigma} &= 2G\left[\frac{\partial\eta}{\partial y} - \frac{1}{3}\left(\frac{\partial\xi}{\partial x} + \frac{\partial\eta}{\partial y} + \frac{\partial\zeta}{\partial z}\right)\right] \\ \sigma_{zz} - \bar{\sigma} &= 2G\left[\frac{\partial\zeta}{\partial z} - \frac{1}{3}\left(\frac{\partial\xi}{\partial x} + \frac{\partial\eta}{\partial y} + \frac{\partial\zeta}{\partial z}\right)\right] \end{aligned}\right\} \tag{1.46}$$

$$\left.\begin{aligned}\tau_{yx} = \tau_{xy} &= G\left(\frac{\partial\eta}{\partial x} + \frac{\partial\xi}{\partial y}\right)\\ \tau_{zy} = \tau_{yz} &= G\left(\frac{\partial\zeta}{\partial y} + \frac{\partial\eta}{\partial z}\right)\\ \tau_{xz} = \tau_{zx} &= G\left(\frac{\partial\xi}{\partial z} + \frac{\partial\zeta}{\partial x}\right)\end{aligned}\right\} \tag{1.47}$$

The units of the shear modulus of elasticity G are seen to be those of stress per unit of strain which is equivalent to stress since the strain is dimensionless.

Newtonian Fluids—

Experiments have shown that for fluids, stresses are related to the time rate of strain rather than to the strain magnitude as in the case of the solid. In analogy with Hooke's law, the most simple form of this relationship is the simple linear proportion

$$\frac{\text{stress}}{\text{time rate of strain}} = \mu \tag{1.48}$$

in which μ is a fluid property called the dynamic viscosity and has the units of stress · time. Fluids which obey this simple proportion are called Newtonian fluids.

Eqs. (1.46) and (1.47) can now be rewritten for Newtonian fluids by replacing G by μ and by taking the time derivative of all the strains. For example:

$$\sigma_{xx} - \bar{\sigma} = \sigma_{xx} + p = 2\mu\frac{\partial}{\partial t}\left[\frac{\partial\xi}{\partial x} - \frac{e}{3}\right]$$

Interchanging the order of differentiation and recognizing that

$$u = \frac{\partial\xi}{\partial t},\ v = \frac{\partial\eta}{\partial t},\ w = \frac{\partial\zeta}{\partial t}$$

we can write

$$\left.\begin{aligned}\sigma_{xx} + p &= 2\mu\frac{\partial u}{\partial x} - \frac{2}{3}\mu\frac{\partial e}{\partial t}\\ \sigma_{yy} + p &= 2\mu\frac{\partial v}{\partial y} - \frac{2}{3}\mu\frac{\partial e}{\partial t}\\ \sigma_{zz} + p &= 2\mu\frac{\partial w}{\partial z} - \frac{2}{3}\mu\frac{\partial e}{\partial t}\end{aligned}\right\} \tag{1.49}$$

and

$$\left.\begin{aligned}
\tau_{xy} = \tau_{yx} &= \mu\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right) \\
\tau_{zy} = \tau_{yz} &= \mu\left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}\right) \\
\tau_{xz} = \tau_{zx} &= \mu\left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)
\end{aligned}\right\} \tag{1.50}$$

Equation of Motion for Newtonian Fluid

Substituting Eqs. (1.48) and (1.49) into Eqs. (1.35) we obtain

$$\left.\begin{aligned}
\rho\frac{Du}{Dt} &= X - \frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left[\mu\left(2\frac{\partial u}{\partial x} - \frac{2}{3}\frac{\partial e}{\partial t}\right)\right] \\
&+ \frac{\partial}{\partial y}\left[\mu\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)\right] + \frac{\partial}{\partial z}\left[\mu\left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)\right] \\
\rho\frac{Dv}{Dt} &= Y - \frac{\partial p}{\partial y} + \frac{\partial}{\partial y}\left[\mu\left(2\frac{\partial v}{\partial y} - \frac{2}{3}\frac{\partial e}{\partial t}\right)\right] \\
&+ \frac{\partial}{\partial z}\left[\mu\left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)\right] + \frac{\partial}{\partial x}\left[\mu\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)\right] \\
\rho\frac{Dw}{Dt} &= Z - \frac{\partial p}{\partial z} + \frac{\partial}{\partial z}\left[\mu\left(2\frac{\partial w}{\partial z} - \frac{2}{3}\frac{\partial e}{\partial t}\right)\right] \\
&+ \frac{\partial}{\partial x}\left[\mu\left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z}\right)\right] + \frac{\partial}{\partial y}\left[\mu\left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)\right]
\end{aligned}\right\} \tag{1.51}$$

which are the famous Navier-Stokes equations which govern the dynamic behavior of Newtonian fluids. Along with these three equations we also have the equation of conservation of mass already derived:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \tag{1.30}$$

In the case of compressible fluids it is necessary to add an equation of state (relating pressure, density and temperature, T) an equation of energy (relating mechanical work and temperature distribution) and

an empirical relation between viscosity and temperature. This gives seven equations in the variables u, v, w, p, ρ, T, μ.

For the incompressible case of primary interest here, the density of a given fluid particle is a function of neither time nor spatial position, thus:

1. The conservation of mass relation becomes

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \equiv \frac{\partial e}{\partial t} = 0 \tag{1.34}$$

2. Pressure variations will not cause large temperature changes through compression, thus the isothermal conditions necessary for the viscosity to be spatially invariant are more commonly approached.

Under these conditions the equations of motion for an isothermal incompressible Newtonian fluid can be written

$$\left.\begin{aligned} \rho \frac{Du}{Dt} &= X - \frac{\partial p}{\partial x} + \mu \nabla^2 u \\ \rho \frac{Dv}{Dt} &= Y - \frac{\partial p}{\partial y} + \mu \nabla^2 v \\ \rho \frac{Dw}{Dt} &= Z - \frac{\partial p}{\partial z} + \mu \nabla^2 w \end{aligned}\right\} \tag{1.52}$$

in which

$$\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2} \tag{1.53}$$

Equations (1.34) and (1.52) provide 4 relations in the 4 variables u, v, w, p and constitute the theoretical foundation for incompressible, isothermal fluid mechanics. Unfortunately these equations are so complex that no general solution of them has been found. Particular solutions have been found, however, for certain special cases but before looking at these it is instructive to examine the dimensionless form of Eqs. (1.52).

Dimensionless Form of Equations of Motion

We will restrict ourselves to the case in which the only body force is that of gravity. We know that the gravitational force field is a poten-

tial field, that is, there exists a scalar function of space, Ω, such that with the axis z oriented vertically upward:

$$\left.\begin{aligned} \frac{\partial\Omega}{\partial x} &= -X = 0 \\ \frac{\partial\Omega}{\partial y} &= -Y = 0 \\ \frac{\partial\Omega}{\partial z} &= -Z = -\rho g \end{aligned}\right\} \tag{1.54}$$

in which g is the local gravitational constant.

Let us define a reference value for each basic variable:

$$\begin{aligned} l_o &= \text{reference length} \\ V_o &= \text{reference velocity} \\ l_o/V_o &= \text{reference time.} \end{aligned}$$

It is convenient to decompose the pressure intensity, p, into a static and a dynamic component, the static component, p_s, being that which would be present if the fluid was suddenly frozen and the dynamic pressure, p_d, that deviation from the static due to fluid acceleration, i.e.

$$p = p_s + p_d \tag{1.55}$$

in which p_s is written in terms of the "static" piezometric head, h_s, as

$$p_s = \gamma(h_s - z). \tag{1.56}$$

There will be two reference pressures:

$$\rho V_o^2 = \text{reference dynamic pressure}$$

and

$$\gamma l_o = \text{reference static pressure.}$$

To be selected as a reference value, the quantity should be important in the flow problem. For example, l_o might be the thickness of a bridge pier or the wave height in deep water. Using the reference values we can define the dimensionless variables:

$$\left.\begin{aligned} U &= u/V_o & l_x &= x/l_o & H_s &= h_s/l_o \\ V &= v/V_o & l_y &= y/l_o & P_d &= p_d/\rho V_o^2 \\ W &= w/V_o & l_z &= z/l_o & T &= tV_o/l_o \end{aligned}\right\} \tag{1.57}$$

Substituting Eqs. (1.54) through (1.57) into Eqs. (1.34) and (1.52) gives:

$$\left.\begin{aligned}\frac{DU}{DT} &= -\frac{\partial P_d}{\partial l_x} - \mathbb{F}^{-2}\frac{\partial H_s}{\partial l_x} + \mathbb{R}^{-1}\nabla^2 U\\ \frac{DV}{DT} &= -\frac{\partial P_d}{\partial l_y} - \mathbb{F}^{-2}\frac{\partial H_s}{\partial l_y} + \mathbb{R}^{-1}\nabla^2 V\\ \frac{DW}{DT} &= -\frac{\partial P_d}{\partial l_z} + \mathbb{R}^{-1}\nabla^2 W\end{aligned}\right\} \tag{1.58}$$

$$\frac{\partial U}{\partial l_x} + \frac{\partial V}{\partial l_y} + \frac{\partial W}{\partial l_z} = 0 \tag{1.59}$$

in which

$$\left.\begin{aligned}\mathbb{F} &= \text{Froude number} = \frac{V_o}{\sqrt{gl_o}} = \frac{\text{inertia forces}}{\text{gravity forces}}\\ \mathbb{R} &= \text{Reynolds number} = \frac{V_o l_o \rho}{\mu} = \frac{\text{inertia forces}}{\text{viscous forces}}\end{aligned}\right\} \tag{1.60}$$

We have introduced a new variable, H_s, and thus need a new equation in order to have enough information available for problem solution. This equation comes from the statement that a fluid particle lying in the static piezometric surface, h_s, must always remain there. If the equation of this surface is

$$F_{h_s}(x, y, z, t) = 0 \tag{1.61}$$

this condition is written

$$\frac{DF_{h_s}}{Dt} = 0$$

Dynamic Similarity

We are now in a position to answer the very important question—What are the necessary and sufficient conditions for dynamic similarity?

By dynamic similarity we mean that condition under which the vector polygons of both force and velocity (the latter of these implies kinematic similarity) are geometrically similar at corresponding points in two geometrically similar systems.

In terms of Eqs. (1.58), (1.59) and (1.62) this means

(1) for the same values of T, l_x, l_y, l_z the two systems will have identical U, V, W, P_d and H_s.

(2) Corresponding terms in the dynamical equations (Eqs. 1.58) must be equal in the two systems.

These conditions are both met:

(1) By assuring geometric similarity of the two systems.

(2) By observing the time scale specified by equal T's in the two systems.

(3) By making the dynamic coefficients, $\mathbb{F}$ and $\mathbb{R}$, of Eqs. (1.58) equal in the two systems. (Note: If the original formulation of the Navier-Stokes equations had included other forces such as surface tension etc., other dynamic coefficients would be present.)

For closed systems the "static" piezometric head is everywhere constant and the problem solution becomes independent of H_s and $\mathbb{F}$.

For open systems (containing a free surface or other interface), H_s may vary and thus $\mathbb{F}$ is important as well as $\mathbb{R}$.

Problems in fluid mechanics are now reduced to determining, by theory, experiment or both, one or more of the dependent variables, U, V, W, P_d, H_s as functions of the independent parameters l_x, l_y, l_z, T and of the dynamic coefficients $\mathbb{F}$ and $\mathbb{R}$. Familiar examples are the many drag coefficient plots which present:

$$P_d = \frac{p_d}{\rho V_o^2} = \frac{F_D/A}{\rho V_o^2} = \frac{C_D}{2} = \phi(\mathbb{R}, \text{geometry})$$

or velocity distribution plots which usually show

$$U = \frac{u}{U_{max}} = \phi\left(\frac{z}{l_o}, \mathbb{R}\right)$$

etc.

Conservation of Energy

The differential form of the energy equation will not be developed here. The interested reader is referred to Rouse (3) for this material.

II. Reduction of Basic Equations in Special Cases

Rotationality vs. Irrotationality

Returning to the two-dimensional representation of Fig. 1.6 (p. 18) we note that one of the possible responses of a fluid particle to stress is a rotation of the particle about its mass center. This rotation is defined

as the average angular velocity of two infinitesimal orthogonal line elements which lie in a plane perpendicular to the axis about which the rotation is desired. Analytically, from Fig. 2.1 the mean rate of change of angles α and β can be written:

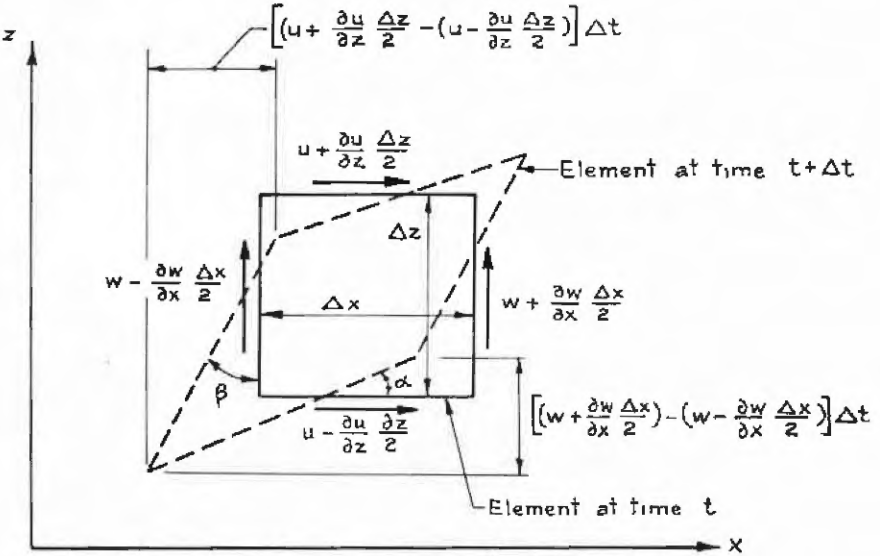

FIG 2.1 ROTATION OF A FLUID PARTICLE

$$\frac{\Delta\alpha}{\Delta t}=\frac{\left[\left(w+\frac{\partial w}{\partial x}\frac{\Delta x}{2}\right)-\left(w-\frac{\partial w}{\partial x}\frac{\Delta x}{2}\right)\right]\Delta t}{\Delta x\,\Delta t}=\frac{\partial w}{\partial x}$$

$$\frac{\Delta\beta}{\Delta t}=\frac{-\left[\left(u+\frac{\partial u}{\partial z}\frac{\Delta z}{2}\right)-\left(u-\frac{\partial u}{\partial z}\frac{\Delta z}{2}\right)\right]\Delta t}{\Delta z\,\Delta t}=-\frac{\partial u}{\partial z}$$

in which counter-clockwise rotations are taken to be positive. The mean angular velocity about the y axis is thus

$$\omega_y=\frac{1}{2}\left[\frac{\Delta\alpha}{\Delta t}+\frac{\Delta\beta}{\Delta t}\right]=-\frac{1}{2}\left[\frac{\partial u}{\partial z}-\frac{\partial w}{\partial x}\right] \qquad (2.01)$$

in which the bracketed quantity is termed the vorticity. In order for a fluid to possess vorticity there must be a rotation of the particle. For this reason, flows in which the vorticity is zero or negligible are called irrotational flows.

Euler Equations

Making use of the two-dimensional equation of continuity:

$$\frac{\partial u}{\partial x} + \frac{\partial w}{\partial z} = 0 \tag{2.02}$$

we can show that

$$\nabla^2 u = \frac{\partial}{\partial z}\left[\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right]$$

and

$$\nabla^2 w = \frac{-\partial}{\partial x}\left[\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right] \tag{2.03}$$

thus, if we specify a flow to have constant vorticity, $\nabla^2 u = \nabla^2 w = 0$ and, regardless of the fluid viscosity, the isothermal, incompressible equations of motion for a Newtonian fluid (Eqs. 1.52) reduce to (for two dimensions):

$$\rho \frac{Du}{Dt} = X - \frac{\partial p}{\partial x}$$

$$\rho \frac{Dw}{Dt} = Z - \frac{\partial p}{\partial z} \tag{2.04}$$

These equations are called the Euler equations. It is seen from Eqs. (1.52) that these same equations apply exactly to the hypothetical case of a zero viscosity fluid regardless of the intensity or distribution of vorticity.

We must ask ourselves whether Eqs. (2.04) have any practical use. Is the vorticity ever constant or nearly so? If so, where and under what circumstances?

Origin and Distribution of Vorticity

It is obvious that a fluid at rest has zero vorticity everywhere. The fluid in all systems was once at rest thus if a fluid has vorticity now where did it come from? Since body forces act through the mass center of a fluid particle they can produce no rotation. Vorticity must thus result from the application of surface forces. Such forces are by defini-

tion applied only at the boundaries of the fluid system. Furthermore, when we compare Eqs. (1.50) and (2.03) we find that in certain cases zero shear stress in a real fluid means zero vorticity. In these cases only those boundaries at which a shear stress exists can generate vorticity. We recognize the solid boundary as just such a vorticity generator due to the "no slip" condition which a real fluid satisfies at these surfaces. Let's look at the two-dimensional distribution of vorticity near a surface of this kind in a flow having an ambient velocity, U_o.

Eliminating pressure between the first and last of Eqs. (1.52) and using Eqs. (2.01) and (2.02) we obtain the equation of vorticity transport

$$\frac{D\omega_y}{Dt} = \nu \nabla^2 \omega_y \tag{2.05}$$

in which $\nu = \frac{\mu}{\rho}$ = kinematic fluid viscosity. Eq. (2.05) is the differential form of a conservation of angular momentum relation and tells us that the total rate of change of vorticity of a fluid particle equals the rate at which vorticity is dissipated by friction. The left hand side of this equation represents convection of vorticity while the right hand side represents conduction at a rate governed by the molecular momentum diffusivity, ν, of the fluid. Since at this time we are only interested in an approximate solution to Eq. (2.05) we will make some very crude assumptions which are totally unacceptable in a quantitative sense but which will not distort the qualitative nature of the phenomenon. These are:

(1) The vertical velocity, w, is zero everywhere.

(2) Vorticity gradients in the z direction are much larger than in the x direction.

With these approximations Eq. (2.05) becomes, for steady flow

$$\nu \frac{\partial^2 \omega_y}{\partial z^2} = u \frac{\partial \omega_y}{\partial x} \tag{2.06}$$

We will now assume that $\partial\omega_y/\partial x = -k_1 \frac{\partial \omega_y}{\partial z}$. The minus sign is crucial but logical. In the absence of w the z transport of vorticity is solely by diffusion and it is the fundamental nature of diffusive processes to have first and second derivatives of opposite sign. With this assumption Eq. (2.06) may be integrated once to obtain

$$\ln\left(\frac{\partial\omega_y}{\partial z}\right) = -\frac{k_1}{\nu}\int u\,dz + f_1(x)$$

which may be approximated

$$\frac{\partial\omega_y}{\partial z} = f_1(x)\,e^{-k_2U_0z/\nu}$$

Integrating again

$$\omega_y \cong -\frac{1}{2}\frac{\partial u}{\partial z} = f_2(x) - f_3(x)\,e^{-k_2U_0z/\nu} \qquad (2.07)$$

and again

$$u = \frac{-2\nu}{k_2U_0}f_3(x)\,e^{-k_2U_0z/\nu} - 2f_2(x)\,z + f_4(x) \qquad (2.08)$$

Under the conditions

(1) $u = 0, \quad \omega_y = \omega_0 \quad \text{at} \quad z = 0$
(2) $u \to U_0 \quad \text{as} \quad z \to \infty$

Eqs. (2.07) and (2.08) become

$$\frac{\omega_y}{\omega_0} = e^{-\mathbb{R}\,z/l_0} \qquad (2.09)$$

$$\frac{u}{U_0} = [1 - e^{-\mathbb{R}\,z/l_0}] \qquad (2.10)$$

in which

$$\mathbb{R} = \frac{k_2U_0l_0}{\nu} \qquad (2.11)$$

Eq. (2.09) demonstrates clearly that while ω_y varies with z close to the surface, as z/l_0 gets large for a given $\mathbb{R}$ or as $\mathbb{R}$ gets large for a given z/l_0, ω_y approaches zero. We thus see that the Euler equations are valid beyond a certain distance, δ/l_0, from a solid boundary. The distance, δ, is called the boundary layer thickness and is usually defined by

$$z = \delta \quad \text{when} \quad \frac{u}{U_0} = 0.99$$

This boundary layer, containing essentially all of the viscous effects, decreases in thickness as the Reynolds number increases and vice-versa.

We may redefine the Reynolds number of interest in these problems as:

$$\mathbb{R}_z = \frac{k_2 U_0 z}{\nu} \tag{2.12}$$

which has the special value, $\mathbb{R}_\delta$, at $z = \delta$. The velocity distribution of Eq. (2.10) can then be written

$$\frac{u}{U_0} = [1 - e^{-\mathbb{R}_z}] \tag{2.13}$$

which is sketched in Fig. 2.2.

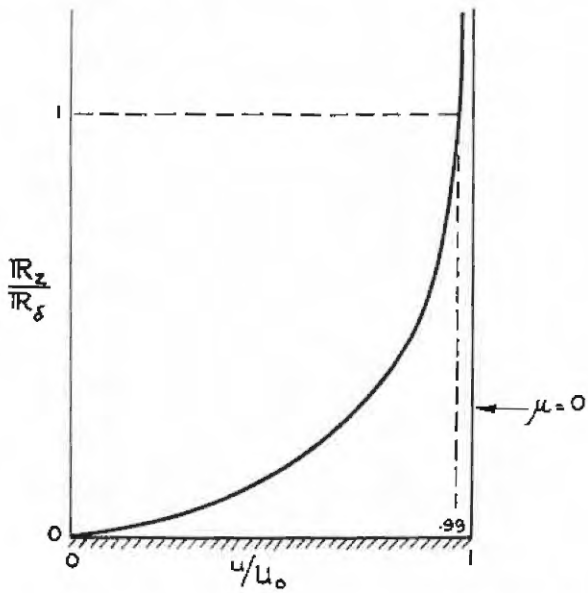

FIG. 2.2 VELOCITY DISTRIBUTION NEAR SOLID BOUNDARY

Recognition of the fact that the Reynolds number, $\mathbb{R}_z$, represents the ratio of inertia to viscous forces allows us to identify three regimes of flow:

(1) *Large* $\mathbb{R}_z$—Inertia forces predominate throughout the flow field. This is satisfied in a real fluid only where the vorticity is constant and thus cannot apply near a vorticity-generating surface. Eqs. (2.04),

the Euler equations, govern this regime. If the vorticity is zero this type of flow is known as irrotational flow.

(2) *Very low* $\mathbb{R}_z$—Viscous forces predominate throughout the flow field. Vorticity transport occurs principally by diffusion and Eq. (2.05) may be approximated by:

$$\nabla^2 \omega_y = 0 \tag{2.14}$$

This is called creeping flow.

(3) *Intermediate* $\mathbb{R}_z$—Viscous and inertia forces are both important and the flow obeys the complete Navier-Stokes equations. This type of flow is called boundary layer flow.

Utilizing Fig. 2.2, this categorization can be summarized:

Location	*Type of Flow*	*Applicable Dynamic Equations*
($z = 0$ at shearing surface)		
$\mathbb{R}_z > \mathbb{R}_\delta$	Irrotational Flow	Euler Equations (2.04)
$\mathbb{R}_z \ll \mathbb{R}_\delta$	Creeping Flow	$\nabla^2 \omega = 0$
Elsewhere	Boundary Layer Flow	Complete Navier-Stokes Eq. (1.52)

Irrotational Flow

Let us look first at the case of very large Reynolds numbers where

$$\nabla^2 u = \nabla^2 v = \nabla^2 w = 0$$

In two-dimensions, the applicable equations of motion are thus Eq. (2.04). If we take the special but most common case of constant vorticity, that of irrotational motion, we have from the definition of ω_y:

$$\frac{\partial u}{\partial z} = \frac{\partial w}{\partial x} \tag{2.15}$$

It can be shown (1) that Eq. (2.15) is a necessary and sufficient condition for the existence of a scalar function of space and time, Φ, called the velocity potential which satisfies the relationships:

$$\nabla^2 \Phi = 0, \tag{2.16}$$

and

$$\left.\begin{aligned}\frac{\partial\Phi}{\partial x} &= -u\\ \frac{\partial\Phi}{\partial z} &= -w\end{aligned}\right\} \tag{2.17}$$

Considering only gravitational body forces we can use Eqs. (1.54), (2.15) and (2.17) to rewrite the Euler Eqs. (2.04):

$$\left.\begin{aligned}\frac{\partial}{\partial x}\left[-\frac{\partial\Phi}{\partial t}+\frac{1}{2}(u^2+w^2)+\frac{p}{\rho}\right]&=0\\ \frac{\partial}{\partial z}\left[-\frac{\partial\Phi}{\partial t}+\frac{1}{2}(u^2+w^2)+\frac{p}{\rho}+gz\right]&=0\end{aligned}\right\} \tag{2.18}$$

in which form they are readily integrated to yield

$$\left.\begin{aligned}-\frac{\partial\Phi}{\partial t}+\frac{1}{2}(u^2+w^2)+\frac{p}{\rho}&=F_1(z,t)\\ -\frac{\partial\Phi}{\partial t}+\frac{1}{2}(u^2+w^2)+\frac{p}{\rho}+gz&=F_2(x,t)\end{aligned}\right\} \tag{2.19}$$

Subtracting equations (2.19) gives

$$gz = F_2(x,t) - F_1(z,t)$$

Since g is not a function of x, it is apparent that F_2 is a function of time alone and thus $F_1 = F_2(t) - gz$. Eqs. (2.19) are then reduced to the single relationship

$$-\frac{\partial\Phi}{\partial t}+\frac{1}{2}(u^2+w^2)+\frac{p}{\rho}+gz=F_2(t) \tag{2.20}$$

For steady flows Eq. (2.20) reduces to

$$\frac{1}{2}(u^2+w^2)+\frac{p}{\rho}+gz=\text{constant} \tag{2.20a}$$

which is the usual form of the steady state Bernoulli equation.

Equations (2.16) and (2.20) provide the means for solving the irrotational flow problem. In the general case, Eq. (2.16) is solved first for the only Φ which also satisfies the applicable boundary conditions. With Φ and thus the velocity components known, Eq. (2.20)

may be solved for the pressure intensity, p, in terms of the unknown time function, $F_2(t)$. Since fluid motion is affected only by pressure gradients and $F_2(t)$ is a constant throughout the fluid at any time, t, the choice of $F_2(t)$ is evidently arbitrary. We may thus say

$$F_2(t) = 0 \tag{2.21}$$

without loss of generality.

This technique permits solution of many practical flow problems of interest to the civil engineer such as the mechanics of water waves, and pressure distribution on submerged objects.

Creeping Flow

When the viscous forces far outweigh the inertial forces (very low $\mathbb{R}_e$) we have seen that the Navier-Stokes equations reduce to a form in which the vorticity is a potential function, i.e.

$$\nabla^2 \omega = 0$$

We can also write Eqs. (1.52) for this case:

$$\left.\begin{aligned} \frac{\partial p}{\partial x} &= \mu \nabla^2 u \\ \frac{\partial p}{\partial y} &= \mu \nabla^2 v \\ \frac{\partial p}{\partial z} + \gamma &= \mu \nabla^2 w \end{aligned}\right\} \tag{2.22}$$

Differentiating the first of these by x, the second by y and the third by z we see that Eqs. (2.22) may be added to obtain

$$\nabla^2 p = \mu \nabla^2 \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \tag{2.23}$$

but from continuity

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

thus, in creeping flow the pressure is also a potential function, i.e.

$$\nabla^2 p = 0 \tag{2.24}$$

The theory of this flow classification is applicable to the solution

of such civil engineering problems as flow in porous media and the settling of very fine sediments.

Boundary Layer Flow

It is indeed unfortunate that in the vast majority of engineering problems both inertia and viscous forces are important. One reason for this dismay has already been pointed out—no general solution to the very complex complete Navier-Stokes equations has been found. However, many practical problems involve conditions which permit simplification of the equations. Let us look first at an example of the type of simplification which will allow exact solutions.

Parallel Flow—

If two of the three velocity components are everywhere zero, the flow is called "parallel." If these two are v and w, Eqs. (1.34) and (1.52) become

$$\frac{\partial u}{\partial x} = 0 \tag{2.25}$$

$$\rho \frac{\partial u}{\partial t} = -\frac{\partial p}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \tag{2.26}$$

$$\left.\begin{aligned} \frac{\partial p}{\partial y} &= 0 \\ \frac{\partial p}{\partial z} &= -\gamma \end{aligned}\right\} \tag{2.27}$$

If the flow is steady and occurs between flat parallel walls a distance 2 b apart, Eq. (2.26) may be further reduced to

$$\frac{dp}{dx} = \mu \frac{d^2 u}{dy^2} \tag{2.28}$$

with boundary conditions $u = 0$ when $y = \pm b$. Since $\frac{\partial p}{\partial y} = 0$, $\frac{dp}{dx}$ is not a function of y and Eq. (2.28) can be integrated to give the velocity distribution

$$\left.\begin{aligned} u &= -\frac{1}{2\mu} \frac{dp}{dx} (b^2 - y^2) \\ \text{or} \quad \frac{u}{U_o} &= \left(1 - \frac{y^2}{b^2}\right) \end{aligned}\right\} \tag{2.29}$$

In this solution the Navier-Stokes equations were solved exactly since the boundary conditions permitted linearization without approximation. It should be noted that the velocity distribution obtained is parabolic rather than exponential as was given by the crude approximations leading to Eq. (2.13). Other practical examples of this type of exact solution can be found. More often however, the simplification which permits solution must involve an approximation. A classical and extremely important example of this technique is that originated by Prandtl for the study of boundary layers on a flat plate.

Boundary Layer Growth on a Flat Plate—

Considering an enclosed two-dimensional flow, the dimensionless Navier-Stokes equations (1.58) and the continuity equation (1.59) become:

$$\frac{\partial U}{\partial T} + U\frac{\partial U}{\partial l_x} + W\frac{\partial U}{\partial l_z} = -\frac{\partial P_d}{\partial l_x} + \frac{1}{\mathbb{R}}\left(\frac{\partial^2 U}{\partial l_x^2} + \frac{\partial^2 U}{\partial l_z^2}\right) \tag{2.30}$$

$$1 \qquad 1\cdot 1 \qquad \delta\cdot\frac{1}{\delta} \qquad\qquad \delta^2 \qquad 1 \qquad \frac{1}{\delta^2}$$

$$\frac{\partial W}{\partial T} + U\frac{\partial W}{\partial l_x} + W\frac{\partial W}{\partial l_z} = -\frac{\partial P_d}{\partial l_z} + \frac{1}{\mathbb{R}}\left(\frac{\partial^2 W}{\partial l_x^2} + \frac{\partial^2 W}{\partial l_z^2}\right) \tag{2.31}$$

$$\delta \qquad 1\cdot\delta \qquad \delta\cdot 1 \qquad\qquad \delta^2 \qquad \delta \qquad \frac{1}{\delta}$$

$$\frac{\partial U}{\partial l_x} + \frac{\partial W}{\partial l_z} = 0 \tag{2.32}$$

$$1 \qquad 1$$

The no slip boundary condition requires

$$U = W = 0 \quad \text{at} \quad l_z = 0$$
$$U = 1 \quad \text{at} \quad l_z \to \infty$$

We will select the reference length, l_o, of Eqs. (1.57) such that $\partial U/\partial l_x$ has a magnitude of order 1. From Eq. (2.32) $\partial W/\partial l_z$ must then also be of order 1 and since W must go from 0 to its maximum value across the vertical distance, $l_z = \delta$, W must be of order δ. Assuming the local accelerations to be of the same order of magnitude as the

convective accelerations we can then assign the orders of magnitude shown under the equations (2.30), (2.31) and (2.32). We note that $\frac{\partial P_d}{\partial l_z}$ is of order δ which means that the pressure has essentially its static distribution perpendicular to the boundary. Furthermore the pressure at $z = \delta$ may be determined from irrotational flow, thus it is regarded as a known function of l_x and T which is impressed upon the boundary layer by the external flow.

In the light of these arguments, Eqs. (2.30)-(2.32) can be approximated by the relations (returning to dimensional form):

$$\rho \frac{Du}{Dt} = -\frac{\partial p}{\partial x} + \mu \frac{\partial^2 u}{\partial z^2} \tag{2.33}$$

$$\frac{\partial u}{\partial x} + \frac{\partial w}{\partial z} = 0 \tag{2.34}$$

with the boundary conditions

$$u = w = 0 \qquad \text{at} \quad z = 0$$
$$u = U_o(x,t) \qquad \text{at} \quad z \to \infty$$

The pressure intensity, $p(x,t)$, in Eq. (2.33) may be determined from solution of the irrotational relation (i.e. Euler Eq.), approximately applicable beyond $z = \delta$:

$$\rho \frac{DU_o}{Dt} = -\frac{\partial p}{\partial x} \tag{2.35}$$

With Eqs. (2.33), (2.34) and (2.35) the variation in velocity from that given at some initial section can be determined as a result of the impressed potential flow.

This very important technique forms the basis for all analytical evaluations of viscous drag.

Even when legitimate approximations cannot be found which permit an analytic solution of a problem there may be a way out. As long as a problem can be formulated, it can (in principle) be solved for a specific set of conditions using numerical techniques and high-speed digital computation. It is the problem formulation which presents the real difficulty. We must remember that the Navier-Stokes equations were based upon an assumed stress-strain relationship modeled after that for an elastic solid but employing rate of strain instead of strain

and substituting the molecular viscosity, μ, for the shear modulus, G. As Reynolds number increases, this so-called laminar relationship ceases rather abruptly to provide a sufficient definition of the stress-rate of strain phenomenon.

Stability of Laminar Flows

A physical system is said to be in a stable state when an arbitrary perturbation to the state is damped with time. It is unstable when the perturbation grows continuously, thereby changing the state of the system. Stability of a flow system such as is described by the Navier-Stokes equations depends upon the relative rate at which the energy of a disturbance is dissipated by viscosity and the rate at which energy is transferred to the disturbance from the mean flow.

If the velocities and pressures of a two-dimensional parallel flow are assumed given by

$$\left.\begin{aligned} u &= U + u' \\ w &= w' \\ p &= P + p' \end{aligned}\right\} \tag{2.36}$$

in which the primed components represent perturbations from the time average values U and P, then the Navier-Stokes equations become:

$$\left.\begin{aligned} &\frac{\partial u'}{\partial t} + U\frac{\partial u'}{\partial x} + w'\frac{dU}{dz} + \frac{1}{\rho}\frac{\partial P}{\partial x} + \frac{1}{\rho}\frac{\partial p'}{\partial x} \\ &\qquad\qquad = \nu\left[\frac{d^2U}{dz^2} + \nabla^2 u'\right] \\ &\frac{\partial w'}{\partial t} + U\frac{\partial w'}{\partial x} + \frac{1}{\rho}\frac{\partial P}{\partial z} + \frac{1}{\rho}\frac{\partial p'}{\partial z} = \nu\nabla^2 w' \end{aligned}\right\} \tag{2.37}$$

The only way energy can get from the mean flow to the perturbation is if mathematical coupling of the two motions exists in one or more terms of these equations. If no such coupling exists, then the two motions are independent. This coupling can be seen to exist in the non-linear inertia terms, $U\,\partial u'/\partial x$, $w'\,dU/dz$ and $U\,\partial w'/\partial x$. Damping of the perturbations can occur only through the viscous terms $\nu\nabla^2 u'$ and $\nu\nabla^2 w'$. It is thus expected that as the Reynolds number of the flow increases (and thus the ratio of inertia to viscous forces increases) a stable flow may become unstable. Furthermore, it should be noted that the inertial

reaction varies directly with the temporal and spatial gradient of the perturbations, thus the stability should be sensitive to their frequency as well as to the Reynolds number of the mean flow.

Mathematical methods exist for the analysis of hydrodynamic stability and have been applied successfully (see Schlichting (2)) to the prediction of the stability relationship of the flow in a laminar boundary layer. The result of this analysis which has been confirmed experimentally, is indicated qualitatively in Fig. 2.3.

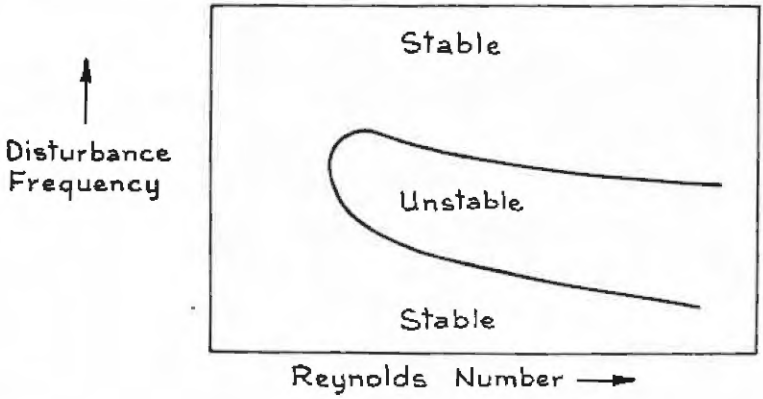

FIG. 2.3 STABILITY OF FLOW IN A LAMINAR BOUNDARY LAYER

Turbulence

In most practical situations, the flow is subjected continuously to a whole spectrum of disturbance frequencies arising from structural vibrations and geometrical irregularities. We have just seen that when the flow is in shear, these disturbances may be amplified. As Fig. 2.3 shows us, there is a Reynolds number below which none of these disturbances will be amplified. This value is so low, however, as to make the existence of laminar flows in civil engineering situations the rare exception rather than the rule.

The unstable disturbances grow in magnitude until something occurs to increase their rate of viscous dissipation sufficiently to create a new equilibrium state called a turbulent flow. This added dissipation

occurs due to the molecular viscosity of the fluid, just as does the direct dissipation of the mean motion, however, the strain rates involved depend upon the kinematic structure of the disturbances rather than on that of the mean flow alone. The equations of motion governing the turbulent state of flow are therefore evidently different from those of Navier and Stokes which govern the laminar state of Newtonian fluids. It should be noted that the turbulent velocity fluctuations are generally smaller than the mean velocity, nevertheless they have a very important effect upon such fundamental characteristics as energy loss, drag and mixing.

Because of the probabilistic nature of the turbulent perturbations it is convenient to adopt an analytical model for their behavior which can take advantage of statistical methods of analysis. As we have already done, all dependent variables, f, (i.e. velocity, pressure, force, piezometric head, etc.) will be written in terms of a time average or mean quantity $\bar{f}$ and a random fluctuating component (or, deviation), f', such that

$$f = \bar{f} + f' \tag{2.38}$$

where

$$\overline{f'} = \lim_{T\to\infty} \frac{1}{T}\int_0^T f'\,dt = 0 \tag{2.39}$$

A convenient statistical measure of the deviation, f', is given by the root-mean-square (RMS):

$$\text{RMS } f' = \sqrt{\overline{(f')^2}} = \left[\lim_{T\to\infty}\frac{1}{T}\int_0^T (f')^2\,dt\right]^{1/2} \tag{2.40}$$

In operating with the equations that are to follow it will be helpful to remember the following rules for taking the time average:

$$\left.\begin{aligned}
\overline{(\bar{f}+f')} &= \bar{f} + \overline{f'} = \bar{f} \\
\overline{(\bar{f}+f')(\bar{g}+g')} &= \bar{f}\cdot\bar{g} + \overline{f'\cdot\bar{g}} + \overline{g'\cdot\bar{f}} + \overline{f'\cdot g'} = \bar{f}\cdot\bar{g} + \overline{f'\cdot g'} \\
\overline{\frac{\partial(\bar{f}+f')}{\partial s}} &= \overline{\frac{\partial\bar{f}}{\partial s}} + \overline{\frac{\partial f'}{\partial s}} = \frac{\partial\bar{f}}{\partial s} + \frac{\partial\overline{f'}}{\partial s} = \frac{\partial\bar{f}}{\partial s}
\end{aligned}\right\} \tag{2.41}$$

Reynolds Equations

Since conversion of mechanical flow energy into heat (i.e. dissipation) can ultimately occur only through the action of the molecular viscosity, all motions, if examined in fine enough detail, must actually be laminar and hence must satisfy the Navier-Stokes equations. However, to make use of this fact the Navier-Stokes equations must be written in terms of the microstructure (turbulence) of the flow and not just in terms of the mean quantities. Let us substitute the quantities

$$\left.\begin{aligned} u &= \bar{u} + u' \\ v &= \bar{v} + v' \\ w &= \bar{w} + w' \\ p &= \bar{p} + p' \end{aligned}\right\} \tag{2.42}$$

into the Navier-Stokes (1.52) and continuity (1.34) equations. We will carry this out in detail for the x component of motion in order to generate some familiarity with the reasoning involved:

Using Eqs. (2.42), the first of Eqs. (1.52) can be expanded to give:

$$\left.\begin{aligned} &\frac{\partial \bar{u}}{\partial t} + \frac{\partial u'}{\partial t} + \bar{u}\frac{\partial \bar{u}}{\partial x} + \bar{u}\frac{\partial u'}{\partial x} + u'\frac{\partial \bar{u}}{\partial x} + u'\frac{\partial u'}{\partial x} \\ &\quad + \bar{v}\frac{\partial \bar{u}}{\partial y} + \bar{v}\frac{\partial u'}{\partial y} + v'\frac{\partial \bar{u}}{\partial y} + v'\frac{\partial u'}{\partial y} \\ &\quad + \bar{w}\frac{\partial \bar{u}}{\partial z} + \bar{w}\frac{\partial u'}{\partial z} + w'\frac{\partial \bar{u}}{\partial z} + w'\frac{\partial u'}{\partial z} \\ &\quad = \frac{\bar{X}}{\rho} + \frac{X'}{\rho} - \frac{\partial \bar{p}}{\rho \partial x} - \frac{\partial p'}{\rho \partial x} \\ &\quad + \nu \nabla^2 \bar{u} + \nu \nabla^2 u' \end{aligned}\right\} \tag{2.43}$$

Let us now consider only those flows in which the mean values do not vary with time. Then, if we take the time average of Eq. (2.43) using the rules of Eq. (2.41), we obtain:

$$\bar{u}\frac{\partial \bar{u}}{\partial x} + \bar{v}\frac{\partial \bar{u}}{\partial y} + \bar{w}\frac{\partial \bar{u}}{\partial z} + \overline{u'\frac{\partial u'}{\partial x}} + \overline{v'\frac{\partial u'}{\partial y}} + \overline{w'\frac{\partial u'}{\partial z}} = \frac{\bar{X}}{\rho} - \frac{\partial \bar{p}}{\rho \partial x} + \nu \nabla^2 \bar{u} \tag{2.44}$$

Subtracting Eq. (2.44) from Eq. (2.43) gives (again for "steady" flows) a similar equation in terms of the fluctuating components:

$$\frac{\partial u'}{\partial t} + \bar{u}\frac{\partial u'}{\partial x} + u'\frac{\partial \bar{u}}{\partial x} + u'\frac{\partial u'}{\partial x} - \overline{u'\frac{\partial u'}{\partial x}}$$

$$+ \bar{v}\frac{\partial u'}{\partial y} + v'\frac{\partial \bar{u}}{\partial y} + v'\frac{\partial u'}{\partial y} - \overline{v'\frac{\partial u'}{\partial y}}$$

$$+ \bar{w}\frac{\partial u'}{\partial z} + w'\frac{\partial \bar{u}}{\partial z} + w'\frac{\partial u'}{\partial z} - \overline{w'\frac{\partial u'}{\partial z}}$$

$$= \frac{X'}{\rho} - \frac{\partial p'}{\rho \partial x} + \nu \nabla^2 u' \tag{2.45}$$

Performing similar operations on the continuity equation we obtain:

$$\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} + \frac{\partial \bar{w}}{\partial z} = 0 \tag{2.46}$$

and

$$\frac{\partial u'}{\partial x} + \frac{\partial v'}{\partial y} + \frac{\partial w'}{\partial z} = 0 \tag{2.47}$$

Turbulent flows can often be handled for engineering purposes by understanding only the mean flow in detail (although real advances can come only through an understanding of the fluctuations). For this reason we will restrict our attention to Eq. (2.44) and the reader is referred to the work of Rouse (3) for discussion of the equations governing the fluctuating components. Multiplying Eq. (2.47) by the fluctuating component, u', and averaging we get

$$\overline{u'\frac{\partial u'}{\partial x}} + \overline{u'\frac{\partial v'}{\partial y}} + \overline{u'\frac{\partial w'}{\partial z}} = 0 \tag{2.48}$$

which may be added (since it is equal to zero) to Eq. (2.44) to give for the mean flow:

$$\rho\frac{D\bar{u}}{Dt} + \rho\left[\overline{u'\frac{\partial u'}{\partial x}} + \overline{u'\frac{\partial u'}{\partial x}}\right] + \rho\left[\overline{u'\frac{\partial v'}{\partial y}} + \overline{v'\frac{\partial u'}{\partial y}}\right]$$

$$+ \rho\left[\overline{w'\frac{\partial u'}{\partial z}} + \overline{u'\frac{\partial w'}{\partial z}}\right] = \bar{X} - \frac{\partial \bar{p}}{\partial x} + \mu \nabla^2 \bar{u} \tag{2.49}$$

Note for example that:

$$\overline{u'\frac{\partial v'}{\partial y}} + \overline{v'\frac{\partial u'}{\partial y}} = \overline{u'\frac{\partial v'}{\partial y} + v'\frac{\partial u'}{\partial y}} = \overline{\frac{\partial (u'v')}{\partial y}} = \overline{\frac{\partial u'v'}{\partial y}}$$

We can then move the turbulent convective accelerations to the right hand side and write Eq. (2.49) in the revealing form:

$$\left.\begin{aligned}\rho\left(\bar{u}\frac{\partial \bar{u}}{\partial x} + \bar{v}\frac{\partial \bar{u}}{\partial y} + \bar{w}\frac{\partial \bar{u}}{\partial z}\right) = \bar{X} - \frac{\partial \bar{p}}{\partial x} \\ + \frac{\partial}{\partial x}\left(\mu\frac{\partial \bar{u}}{\partial x} - \rho\,\overline{u'u'}\right) \\ + \frac{\partial}{\partial y}\left(\mu\frac{\partial \bar{u}}{\partial y} - \rho\,\overline{u'v'}\right) \\ + \frac{\partial}{\partial z}\left(\mu\frac{\partial \bar{u}}{\partial z} - \rho\,\overline{u'w'}\right)\end{aligned}\right\} \quad (2.50)$$

When written in the form of Eq. (2.50), the fluctuating terms in the mean flow equation of motion can be interpreted as stresses. These stresses are known as apparent or virtual stresses of turbulent flow or more commonly perhaps as Reynolds stresses. They may be very large when compared with their companion terms for the mean viscous shear, but they are still inertia forces and cannot account for energy dissipation since ultimately this must involve viscous action.

Remembering that we are considering only flows which are steady in the mean we can write the three Reynolds equations

$$\left.\begin{aligned}\rho\frac{D\bar{u}}{Dt} &= \bar{X} - \frac{\partial \bar{p}}{\partial x} + \mu\nabla^2\bar{u} - \rho\left[\frac{\partial \overline{u'^2}}{\partial x} + \frac{\partial \overline{u'v'}}{\partial y} + \frac{\partial \overline{u'w'}}{\partial z}\right] \\ \rho\frac{D\bar{v}}{Dt} &= Y - \frac{\partial \bar{p}}{\partial y} + \mu\nabla^2\bar{v} - \rho\left[\frac{\partial \overline{u'v'}}{\partial x} + \frac{\partial \overline{v'^2}}{\partial y} + \frac{\partial \overline{v'w'}}{\partial z}\right] \\ \rho\frac{D\bar{w}}{Dt} &= \bar{Z} - \frac{\partial \bar{p}}{\partial z} + \mu\nabla^2\bar{v} - \rho\left[\frac{\partial \overline{u'w'}}{\partial x} + \frac{\partial \overline{v'w'}}{\partial y} + \frac{\partial \overline{w'^2}}{\partial z}\right]\end{aligned}\right\}$$

(2.51)

These equations, together with the continuity equations (2.46) and (2.47), form the tools for the exact solution of a turbulent flow problem. Unfortunately however, three new dependent variables, u′, v′ and w′, have been added with the addition of only one extra equation, (2.47). Therefore, the mean flow cannot be determined, for a given set of boundary conditions, without some relationship between the mean and fluctuating components of velocity. The search for such relationships has formed the basis for the bulk of analytical and experimental research on turbulent flows.

Classification of Turbulent Flows

Consider a turbulent flow near a solid boundary. As a consequence of the no slip condition there can be no turbulent fluctuation in the plane of the boundary at the boundary. Furthermore, the fluctuating component normal to the boundary must vanish unless the latter is permeable. Turbulence is thus zero at a solid boundary and will be small in its immediate vicinity. It follows that at and very near a solid boundary the viscous stresses predominate over the Reynolds stresses and the flow is laminar. The zone in which this occurs is called the laminar sub-layer. As distance from the wall increases, the effect of the wall in suppressing turbulence decreases and the Reynolds stresses

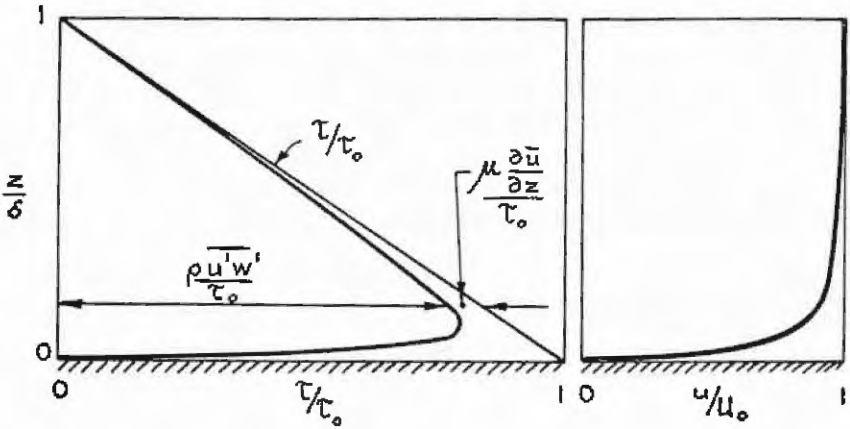

FIG. 2.4 VELOCITY AND SHEAR DISTRIBUTION IN TURBULENT FLOW NEAR A WALL

grow in magnitude. This is indicated schematically in Fig. 2.4. At the same time the viscous stresses are decreasing due to the reduction in velocity gradient with distance from the wall. Eventually the Reynolds stresses predominate and this zone of the boundary layer is fully turbulent.

A turbulent flow such as just described is classified as wall turbulence due to action of the boundary in suppressing the fluctuations. It is further classified as a turbulent shear flow due to the non-zero value of the Reynolds stresses.

Turbulence in the early wake of an object would also be a turbulent shear flow since the turbulence originates at a shearing surface as in the case of the wall boundary layer. In the case of the wake, however, the shear surface is a fluid surface, the separation streamline, at which the fluctuations need not vanish. Such flows are called free turbulence due to the absence of boundary effects.

Flows in which the turbulence has the same structure at all points are called homogeneous. Two-dimensional flows are homogeneous in at least one direction.

A special class of homogeneous flows is that in which the turbulence structure does not vary with direction at any point. These are called isotropic. It is the nature of free turbulent fields which are decaying under the action of viscosity to approach the condition of isotropy.

If a flow is truly isotropic, the fluctuation components must be statistically independent. The mean value of the product of two statistically independent random variables is the product of their mean values or zero. Thus, for isotropic turbulence

$$\left.\begin{aligned} u'^2 &= v'^2 = w'^2 \\ \text{and} \qquad \overline{u'v'} &= \overline{u'w'} = \overline{v'w'} = 0 \end{aligned}\right\} \tag{2.51a}$$

and the fluctuating terms vanish from the Reynolds equations. In actuality however, even though the correlations, $\overline{u'v'}$, etc. may be zero, the flow decays in the direction of the mean velocity, thus the gradient of the $\overline{u'^2}$ term is finite.

Turbulent Transport Processes

To fully understand the importance of turbulence in every aspect of human life, some attention must be paid to the manner in which energy, momentum and mass are transferred from one spot to another

in a fluid medium. The mechanisms for this transfer are radiation, conduction and convection.

Radiation, a wavelike phenomenon, is of interest in civil engineering problems which involve the compressibility of the fluid or the propagation of gravity waves. The modes of major interest are conduction and convection.

Conduction—The conduction process is molecular in scale. The molecules of any solid, liquid or gaseous substance are in continual random motion, the intensity of which is determined by the temperature of the given substance. Because of this random molecular motion there will be an interchange of molecules between layers. This interchange will be effective, in the average, over a length which is known as the molecular mean free path. These conduction processes are sometimes termed molecular diffusion.

If adjacent layers are at different temperatures, this exchange will produce a net transport of heat; if they are in motion at different mean velocities, this will produce a net transverse transport of momentum. If we are dealing with impure (multicomponent) fluid systems, different concentrations of the impurity in adjacent layers will produce a net transport of mass (called ordinary diffusion).

Convection—The convection process is macroscopic in comparison with the microscopic nature of conduction. Once again the transport depends upon a gradient. That is, if a static body of fluid is subjected to a gradient of piezometric head, flow will take place which convects (i.e. carries with it) energy, mass and momentum. For purposes of this discussion we will restrict ourselves to turbulent convection or turbulent diffusion which is that transport which occurs due to the turbulent fluctuations. It is conceptually convenient (although not strictly correct physically) to look upon turbulent motion as a system of eddies (vortexes) of varying scale (size) and intensity (rotational velocity) superimposed on the mean flow. Under such a model macroscopic packets of fluid are moved about at random thereby convecting their cargoes of energy, mass and momentum from spot to spot at a rate which is, for the same mean flow conditions, many times greater than that which occurs due to the molecular conduction processes just discussed.

A similarity is observed in the above descriptions of molecular and turbulent diffusion which provides the key to a phenomenological model relating the mean flow and the Reynolds stresses.

Let us call the amount of the property (per unit mass of fluid) whose transfer is to be studied, s, and let us assume (only for convenience) that in our given flow this quantity varies in only the z direction. This is indicated in Fig. 2.5. Because of molecular motion

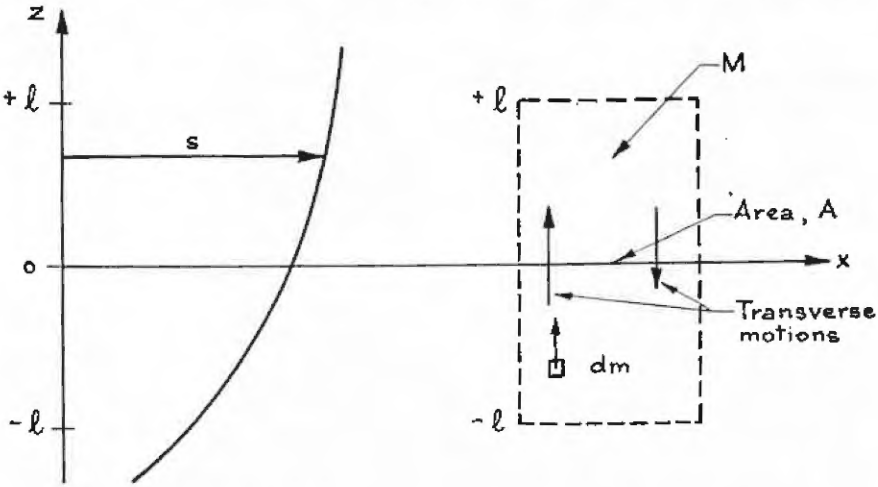

FIG. 2.5 DEFINITION SKETCH FOR CONDUCTION AND TURBULENT CONVECTION

and also turbulence (if present) a vertical exchange of fluid packets is taking place through any horizontal area, A, at elevation, z. The distance traveled by these fluid packets before taking on the character of their new surroundings is assumed to be a random quantity, the mean value of which we will call l. The amount of s carried by a parcel of mass, dm, and passing through A will depend upon the point of origin of the motion of the parcel. This is because of the assumed gradient of s in the direction of the random motion. Statistically speaking no particle lying outside the limits $+l > z > -l$ will pass through A. The average time needed for the parcel to travel the distance l will be called t.

The net upward transport, S, of s per unit of time and area may then be written

$$S = \frac{M/2}{Atl}\left[\int_{-l}^{0} s(z)\,dz - \int_{0}^{+l} s(z)\,dz\right]$$

Expanding s(z) in a Taylor series and integrating

$$S = -\frac{\rho l^2}{t}\left(\frac{\partial s}{\partial z}\right)_{z=0} = -K\left(\frac{\partial s}{\partial z}\right)_{z=0} \tag{2.52}$$

When speaking of molecular processes the quantity $\frac{l^2}{t}$ is a property of the medium and in general varies with both temperature and pressure. If the process is heat conduction then $\partial s/\partial z$ becomes $\partial T/\partial z$ and K is the thermal conductivity. If the process is molecular momentum transfer, s becomes the momentum per unit mass (i.e. the velocity) and S, the momentum flux per unit of area. The constant K is then the molecular viscosity, $\nu = \mu/\rho$. For $s = \bar{u}$, Eq. (2.52) becomes:

$$S = \tau_{zx} = -\mu\frac{\partial \bar{u}}{\partial z}$$

The minus sign signifies that the net transfer of momentum takes place "down" the velocity gradient thereby applying a force in the positive x direction to fluid below surface A.

Eq. (2.52) could be applied to the case of turbulent convection by imagining the length, l, to be a mixing length. In this case the time required for the particle to travel the distance, l, would be

$$t = \frac{l}{v'}$$

and the change in x component of mean velocity experienced by an excursion in the positive z direction would be

$$u' = -l\frac{\partial \bar{u}}{\partial z}.$$

Using these two relations Eq. (2.52) becomes, in the average

$$\tau_{zx} = +\rho\,\overline{u'v'}$$

which is the Reynolds stress. It is clear now that K is not a property of the fluid for the case of Reynolds stresses but must reflect the local kinematics of the flow. We know that for $\overline{u'v'} \neq 0$, u′ and v′ must be related. Assuming this relationship to be a simple proportion we include the proportionality factor in the definition of l to write:

$$\tau = \rho l^2\left|\frac{\partial \bar{u}}{\partial z}\right|\frac{\partial \bar{u}}{\partial z} = \rho\varepsilon\frac{\partial \bar{u}}{\partial z} \tag{2.53}$$

The quantity ε is often called the kinematic eddy viscosity. From arguments of dynamic similarity, Kármán introduced the assumption that

$$1 = \varkappa \frac{\partial \bar{u}/\partial z}{\partial^2 \bar{u}/\partial z^2} \tag{2.54}$$

in which the coefficient, $\varkappa$, has come to be known as the Kármán Universal Constant. Combining (2.53) and (2.54) we can write

$$\sqrt{\frac{\tau}{\rho}} = -\varkappa \frac{(\partial \bar{u}/\partial z)^2}{\partial^2 \bar{u}/\partial z^2} \tag{2.55}$$

which can be integrated once the function $\tau(z)$ is known.

To summarize this discussion of turbulent transport processes:

1. Diffusion, either molecular or turbulent, requires a gradient of the diffusant and takes place in the direction of decreasing concentration of the diffusant.
2. For a given gradient in a given medium the "depth of penetration" of the diffusion process will be directly proportional to the diffusivity, K.
3. The rate of transfer under a given gradient in a given medium is directly proportional to the diffusivity, K.
4. Experiments have shown the eddy diffusivity (eddy viscosity) to be of the order 10^3 to 10^7 times larger than the molecular diffusivity (molecular viscosity).
5. The total transfer under any circumstances is equal to the sum of the transfers by molecular and turbulent means.

It is also interesting to note that experiments indicate a difference in the turbulent transfer coefficients for mass and momentum under the same flow conditions.

Turbulent Energy Spectrum

The energy in a random process such as the turbulent velocity fluctuations is distributed over a continuous spectrum of frequencies, f, or alternatively wave numbers, k, where

$$f(\text{cps}) \sim k \sim \frac{2\pi}{L}$$

It has been established experimentally that the wave number of the turbulence generated by the mean flow in the process of doing work

against the Reynolds stresses is of the order of the boundary scale. For example, as an irrotational flow of a real fluid passes through a screen, a free turbulent shear flow is created by the multitude of wakes which has a predominant wave number determined by the bar size of the screen. Imagine this initial disturbance to be a simple harmonic motion. We can imagine this sinusoidal disturbance as the input to a black box the output of which is the turbulent velocity fluctuation at some point further downstream of the screen. If the black box performs a linear operation on the input, the output will be monochromatic also. We know, however, that the equation of motion is highly non-linear, thus it is reasonable to expect this spectral transfer function to be non-linear also. In this case a monochromatic input is distorted so that the output must contain the fundamental (input) frequency in addition to other, higher, harmonics. Thus, as these original eddies move downstream, they continually give up energy to the generation of eddies at slightly

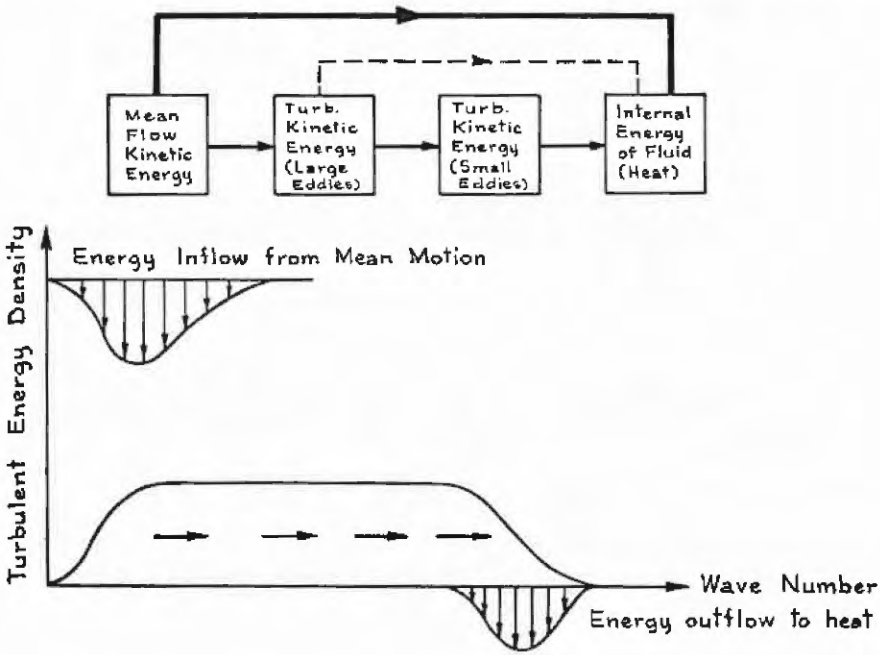

FIG. 2.6 SPECTRAL PATH OF TURBULENT ENERGY (AFTER CORRSIN (5))

higher wave number. These eddies then generate still smaller eddies and so on. Eventually the scale of the eddies becomes so small (thus decreasing the turbulence Reynolds number, ε/ν) that the viscous forces predominate and the kinetic energy of the fluctuations is converted ("dissipated") into heat. This energy path is indicated schematically in Fig. 2.6 which is taken from the superb discussion by Corrsin (5).

Turbulent Shear Flows

Wall Turbulence—

Turbulence due to the influence of solid boundaries is probably of primary interest to civil engineers. Under the assumption that τ is constant with z and is equal to the boundary value, τ_0, Eq. (2.55) can be integrated to give the familiar logarithmic distribution of mean velocity:

$$\frac{\bar{u}}{\sqrt{\tau_0/\rho}} = \frac{1}{\varkappa} \ln z + C$$

Experiments in Newtonian fluids indicate $\varkappa$ to be about 0.4 and the constant C to vary with the geometry of the channel. It is curious that the relationship holds well in parallel flows where τ varies linearly with z.

Free Turbulence—

Free turbulent shear flows are commonly found in wakes or jets where a separation stream line exists. This separation streamline is initially a line of abrupt discontinuity in mean velocity. The high shear at this discontinuity generates intense turbulence which then diffuses laterally. In the case of wall turbulence an equilibrium mean velocity profile was eventually reached because the turbulence-generating surface was continuous in the direction of the mean motion. In the case of free turbulence, however, the lateral transport of momentum acts to soften the discontinuity in mean velocity which is serving as the turbulence generator. The diffusion process thus causes, in the direction of the mean flow, a continuous modification of the mean velocity distribution and a continuous widening of the wake or jet. This is illustrated by the sketch of Fig. 2.7.

Analytical solutions proceed as in the case of boundary layer analysis. The general procedure is as follows:

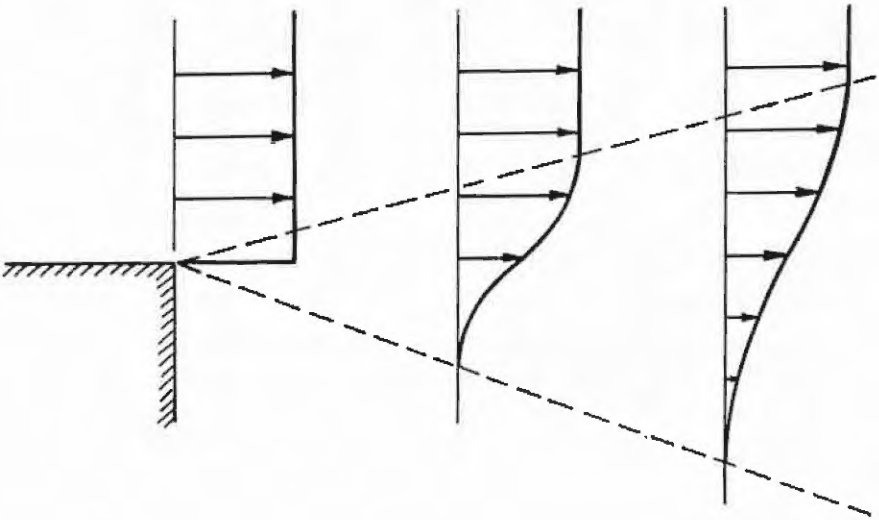

FIG. 2.7 MEAN VELOCITY GRADIENTS IN FREE TURBULENT SHEAR FLOW

1. Neglect longitudinal velocity variations in comparison with lateral.

2. Assume rate of wake spreading is very gradual. (With 1 and 2 the pressure can be assumed constant everywhere.)

3. Neglect viscous stresses in comparison with Reynolds stresses. For a two-dimensional flow we then have (Eqs. (2.51) and 2.46)):

$$\bar{u}\frac{\partial \bar{u}}{\partial x} + \bar{w}\frac{\partial \bar{u}}{\partial z} = -\frac{\partial(\overline{u'w'})}{\partial z}$$

and

$$\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{w}}{\partial z} = 0$$

4. Assume similar distribution of mean velocity at all x.

5. Assume the lateral distribution of some characteristic of the mean flow or of the turbulence.

6. Assume some relation between mean flow and turbulence.

If two diffusing shear layers meet, such as will happen at some point downstream in a jet of finite diameter or behind a bluff body of finite thickness, the diffusion process is altered and a separate analytical solution must be sought. The early stage of such diffusion is called the zone of flow establishment and holds up to the intersection of the two processes. Downstream of this lies the zone of established flow.

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REGISTRATION AND PROFESSIONALISM

By Frank L. Heaney,* *Member*

(Presented at a meeting of the Boston Society of Civil Engineers, December 16, 1964.)

Significance of Registration

Increasing Recognition

The obtaining of registration as a professional engineer has become a magic touchstone by which a technician becomes a professional. This rather startling statement is becoming true in industry and in government as well as in the private practice of engineering. If the public does not always recognize the certificate of engineering registration as a mark of the professional, usually it is the fault of the engineers and of the laxity of their engineering registration laws rather than the intelligence of the layman.

Many of the industrial and electronic firms in Massachusetts, some of which expressed strong opposition to mandatory registration of engineers, are now encouraging their qualified employees to secure their licenses. This is in spite of the fact that in many cases their activities are exempt from the provision of the registration laws. One of the few items where registration actually is involved in industrial work, is the requirement that engineers' professional stamps be placed on certain test certificates.

Heads of departments in plants which employ engineers, holders of engineering titles, and engineers employed in research are now told that engineering registration is becoming necessary for a promotion to engineering positions and is being taken into consideration in awarding salary increases.

The values to industry are manifold. Registration has become a clear demarkation between professional and nonprofessional personnel. This is an important factor in labor relations problems. Surprisingly, the unions also welcome registration as making a clear identification of certain employees. Needless to say, the unions would like to bring the professional employees in under their tent. The special qualification of registration also makes it more difficult to replace such an

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employee during labor strife. The most important effect, however, is that the professional engineering license encourages an assumption of greater responsibility on the part of the employee.

Government, including the military, is one of the largest employers of civil engineers. Federal agencies are exempt from mandatory registration laws of the states. Therefore, the advantages of professional engineering registration were slow to be realized by federally employed engineers. However, it is in this group that great strides are now being taken in encouraging engineers to become registered.

Many state, county and municipal governments require that the responsible, higher level, top-paying positions be filled by registered engineers. Some states require registration for all persons with "engineer" titles. For example, in Texas all highway employees with engineering titles must be registered as professional engineers.

The attitude of the military services can be expressed best by the following quotation from Lieutenant General W. K. Wilson, Jr., Chief of Engineers, U.S. Army, and himself a registered professional engineer:

> "As Chief of the Army Engineers, I am emphasizing a policy, which I have always supported, to encourage all civilian employees and officers engaged in engineering within the Corps of Engineers to become registered members of their profession.
>
> "There are a great many advantages in registration, some of which I consider to be particularly rewarding. One of these is the increased self-confidence it gives any man in the Government to know he can measure up fully to the requirements for the practice of his profession which those in private life must meet.
>
> "Another is the added prestige that registration gives him in dealing day to day with the registered private engineers of the firms which design and build so much of the construction he is called upon to administer and supervise.
>
> "Perhaps most important of all is the fact that every engineer, whether in private or government practice, has an obligation actively to contribute to the preservation and enhancement of the public stature of his profession. Such contributions are essential to ensure that the profession is responsive to the growing demands society is placing on the engineer."

Similar statements have been made by Major General A. M. Minton, P. E., Director of Civil Engineering for the Air Force, and by Rear Admiral P. Corradi, Chief of Navy Civil Engineers.

There are now more than 17,000 professional engineers registered in Massachusetts and more than 280,000 registered in all states. Regis-

tration is mandatory in all of the 50 states as well as in many foreign countries. These figures indicate the extent of recognition being afforded engineering registration.

Reasons for Registration and Licensing

Over 100 occupations, both professional and nonprofessional, are licensed in one or more states. Job titles run from architects, doctors, opticians, and professional engineers; to barbers, beauticians, and plumbers.

A license has been defined as "authority to do some act or carry on some trade, profession or business, in its nature lawful but prohibited by statute except with the permission of the civil authority."

The advantages to the engineer have been very well stated in the foregoing quotation from General Wilson. These apply to the civilian engineer as well as the military engineer.

Public acceptance and recognition of the professional caliber of engineering are based upon a feeling of confidence. This confidence can be assured by the engineer having demonstrated his competence and character through passing examinations for registration. If we consider, for a moment, the public attitude towards a certified public accountant as compared to a bookkeeper or accountant without this distinction, we are bound to agree to this premise.

Principles of Engineering Registration Laws

Purpose of Laws

Registration as stipulated in the various state statutes is intended to provide two services:

(1) Protect the public from a possibility of being duped unknowingly by an incompetent into practices which are unsafe, unhealthy, or not in the interest of the public's welfare.

(2) Provide a means of legal recourse in the cases of malpractice or misrepresentation.

History

Registration was made mandatory for the practice of engineering in Wyoming in 1907. This was followed by all the states, with Massachusetts being the last to adopt the mandatory provisions in 1958. In several of the states failure of structures caused by inadequate design accelerated the acceptance of their licensing statutes.

The present Massachusetts engineering registration law is a compilation of amendments to Chapters 13 and 112 of the General Laws as effected by Chapters 643 and 722 of the Acts of 1941, and Chapter 584 of the Acts of 1958. Copies of this compilation are available at the office of the Registration Board in Room 34 at the State House.

The requirements for registration in Massachusetts have been briefed by Mr. Harrison I. Dixon, a member of the Board of Registration, in the November, 1964 issue of the "Massachusetts Professional Engineer." The following is based on his article:

As a professional engineer,

A. Graduate with B.S. degree in engineering from an accredited ECPD college. He must also have had at least four years of recognized professional engineering practice.
B. A non-graduate who has had eight years' professional experience and who is permitted to take the examination held by the Board twice yearly.
C. "Eminence" or 12 years of professional engineering practice if NOT less than 35 years of age.
D. "Reciprocity" by registration in another state under methods equal to those of Massachusetts. The Board also recognizes a "National" certificate of registration available to those properly qualified who may be practicing in more than one state.

As an Engineer-in-training,

A. Graduation in accredited engineering curriculum of four scholastic years or more from an accredited ECPD college, and who is permitted to take a written examination in the basic engineering subjects.
B. A specific record of four years of experience satisfactory to the Board and who is permitted to take written examination in the basic engineering subjects.

As a land surveyor,

A. Graduate with B.S. degree in civil engineering, including a course in surveying, from an accredited ECPD college plus two years of professional experience.

B. A non-graduate with 6 years' professional experience, if he passes the land surveyor's examination.
C. Ten years' professional experience, if not less than 30 years of age.

Engineer-in-Training Program

There are few in the profession of engineering who consider a man is qualified to practice immediately upon graduation. This is truer today than previously. Our science oriented colleges fit the student with a good basis for adapting readily to our changing technology. However, the virtual abandonment of design courses leaves a gap for the how and why of design.

Most state laws require him to spend 4 years under the guidance of a professional engineer before being admitted to responsible charge of engineering work. This is similar to the internship required of medical doctors and is for the same reason.

The student is encouraged in his last year of college to take the Engineer-in-Training examination (E.I.T.). The questions are intended to test his competency in basic engineering fundamentals. It is the first part of the professional engineers' examination. The second part is given 4 years later when he is questioned in regard to actual practice.

The importance in taking the E.I.T. examination as early as possible instead of at the end of the 4-year period is readily apparent. Those who have waited have had much greater difficulty recollecting fundamental details.

Massachusetts does not require an examination at the present time. It is hoped the law will be changed soon to bring us up to the many other states in this regard. Civil engineering has never been a profession for the stay-at-homes. The young engineer should by all means aim at qualifying in any state.

The E.I.T. status also makes the young engineer aware of his professional responsibilities. It tends to keep him out of the ranks of the technicians and away from joining a union.

American Society of Certified Engineering Technicians

N.S.P.E. in 1961 established an "Institute for Certification of Engineering Technicians." It authorized the Institute to "concern itself entirely with technicians who work for and under the direction of engineers."

Certificates are issued in 3 grades:

A. Senior Engineering Technician-35 years old, 7 years' experience and endorsement of 3 professional engineers.
B. Engineering Technician-25 years old, 7 years experience, endorsement of two professional engineers and a written examination for those who are not graduates of accredited engineering curriculum.
C. Junior Engineering Technicians, no age limit, 2 years experience or graduation from ECPD accredited program, and endorsement of one professional engineer.

A national society, American Society of Certified Engineering Technicians, was formed in April, 1964. It is expected that it will have a rapid growth.

Recognition of the engineering technicians should not dilute or lower the status of the professional engineer. It will, however, provide him with competent assistants who will have a pride in their work. The technician will now know that he is an important member of the engineering team.

Model Law

In any discussion of engineering registration laws we hear a great deal about conformity with the Model Law. The first Model Law was drafted in 1911 by the American Society of Civil Engineers. It soon gained the support of other so-called "founder societies." It was intended "to serve as a model to be followed in the filing of all new registration laws and in the amending of existing laws, with a view to attaining a uniform, high standard throughout the United States."

The Model Law has been revised in 1925, 1927, 1929, 1937, 1943, 1946, and 1960. The present Massachusetts statute is in general conformity with the 1946 revision.

The 1960 revision has not been approved as yet by the American Society of Civil Engineers or by the Consulting Engineers' Council. The major objection is the provision for practice by corporations.

Corporate Practice

Section 81R (f) of the Massachusetts Statute provides as follows:

"Practices or performance of work not prevented or affected. Nothing in said sections shall be construed to prevent or to affect:—(f) The practice of engineering or land surveying in the Commonwealth by a firm, co-partnership, corporation or joint stock association; provided that the person in

charge of such practice by such firm, co-partnership, corporation or joint stock association is a professional engineer or land surveyor holding a certificate of registration under said section."

The 1960 version of the Model Law contains this exemption clause:

"This act shall not be construed to prevent or to affect:—(d) Corporate and Partnership Obligations. The practice or offer to practice engineering, as defined by this Act, by individual registered professional engineers through a partnership, joint stock company, or corporation, as agents, employees, officers or partners, provided they shall be individually liable for their professional acts, and further provided that all personnel of such partnership, joint stock company or corporation, who act in its behalf as engineers in the State are registered under this Act or are persons practicing lawfully or are exempt under paragraph (b) or (c) of this section. Each such partnership, joint stock company, or corporation providing engineering services shall be jointly and severally liable with such individual registered professional engineers. And all final plans, designs, drawings, specifications and reports involving engineering judgment and discretion, when issued, shall be dated and bear the seals and signatures of the engineers who prepare them."

The point of issue seems to be the key word "through" instead of "by."

The states of Rhode Island and Washington have recently made sweeping revisions of their registration laws and have adopted most of the 1960 Model Law provisions. Both of these states provide, as Massachusetts does, for the practice of engineering by corporations rather than by individuals through the business medium of a corporation. The State of Washington even goes so far as to license the corporation to practice engineering. This latter question brings up the point as to whether only natural persons may be admitted to the practice of a profession. The writer believes that our Massachusetts version is satisfactory to corporations and to most engineers.

Registration Required of Employees and Subordinates

There is a great deal of confusion about who is legally required to be registered. Many uninformed persons consider that it is only those engineers who offer their services to the public, in other words, the owners of an engineering firm or the heads of other engineering organizations.

In this regard, attention is directed to the following wording of the Model Law, Section 22 (c), and of the Massachusetts statute, Section 81R (d):

"This act shall not be construed to prevent or to affect: 'The work of an employee or a subordinate of a person holding a certificate of registration under this Act,' . . . '*provided such work does not include final engineering designs or decisions and is done under the direct responsibility, supervision, and checking of a person holding a certificate of registration under this act.*'" (Italics added by the writer.)

It seems clear that this does not exempt any engineers doing responsible work either in the office or in the field. This general ignoring of the clause is causing one of the major violations of the principles of the engineering registration law both in Massachusetts and in several other states.

The new Model Law calls for registration of *all* engineers whether in private practice or in other forms of engineering endeavor. The Consulting Engineers Council in an open letter to all consulting engineers dated October 10, 1962, questions whether the engineering profession as a whole is ready for this step.

There is considerable opposition on the part of some consulting engineers to the concept that all engineers should be required to be registered. There is no question that it will result in higher salary levels and so raise the cost of engineering. Many also question the public benefit as the consultant now bears all of the financial and contract responsibility for the engineering work. They state that it is merely up to the head of the firm to employ persons in whom he has confidence.

The writer agrees with the framers of the new Model Law. This is the nub of the problem:—An engineer is only a technician if he is not registered. There is no middle ground if we are to have a profession.

Policing of the Profession

As in the medical and legal professions, it is necessary to set up some means within the profession to police the members. Until recently we had no counterpart of the regional medical society or bar association. Committees have now been formed to investigate reported infractions of the registration laws. Reports of unethical practices are being handled by separate committees.

A Massachusetts corporation was formed in February, 1963, entitled "Committee to Uphold the Principles of Engineering Registration Laws" (CUPERL, Inc.). The stated purposes of this corporation are:

"To safeguard and advance the interests of the general public by improving the practice of professional engineering and of land surveying through advocating measures for increasing the efficiency and effectiveness of engineering registration laws; to receive and investigate complaints of illegal practice of engineering, of negligence, incompetency and misconduct of registered and unregistered engineers and land surveyors and report on same to the proper authorities, including the presentation of complaints to the Board of Registration of Professional Engineers and of Land Surveyors, District Attorneys, or Attorney General, as well as the making of recommendations toward expulsion from professional and technical societies, and to this end to provide liaison with the many engineering societies and the appropriate governmental agencies."

This type of activity is supplemented by local chapters of the Massachusetts Society of Professional Engineers. This work is coordinated by CUPERL, Inc. In several other states, similar activities are carried on entirely by the State Society of Professional Engineers. However, in Massachusetts, many engineering societies wished to have a broader base of operation than that afforded by MSPE.

In order to separate the responsibility from any particular society, a corporation was formed. This has obvious legal advantages both as to tax status and as to the incurring of any possible liabilities for slander and libel.

The corporation, CUPERL, Inc. provides little or no protection to its members if sued for damaging the reputation or destroying the livelihood of the parties under investigation. Great care must be used in the handling of the cases. Fortunately, the disclosure of data relating to alleged transgressors can be confined to the members of the corporation who have no obligation to report to the various societies or to anyone outside of their group.

The various engineering societies in the state may nominate one of their members for consideration for membership in the Corporation. The trustees of CUPERL usually follow these recommendations but there is no obligation to do so. The Corporation may select any registered professional engineer as a member. This serves to keep the corporation independent in fact from the engineering societies. There are nine members presently and another nine may be elected bringing the total to a maximum of 18.

The type of case which has arisen most frequently is a non-registered person offering engineering services to the public. In the recent state election several persons included a statement in their

political advertisements that they were engineers although they were not registered. In fact, some of them were a "far cry" from being engineers. Action was taken by CUPERL to bring about the satisfactory corrections in the above cases.

An alleged fraud in presenting surveying data to a public agency was brought to the attention of the Corporation. This has been discussed with the Attorney General's office and is presently being investigated.

A program is now nnderway by the various chapters of MSPE and by CUPERL to make sure that the various county, city and town engineers are registered. We have been informed that because of rulings on "Home Rule" the state cannot prevent towns and cities from assigning any title to their employees. We can, however, make sure that they do not practice or profess to practice professional engineering or land surveying without being registered. We have been successful in inducing some Boards of Selectmen to make registration a requirement in appointing Town Engineers.

In some States, the Board of Registration of Professional Engineers passes upon questions of ethical behavior. This provision is not included in the Massachusetts statutes nor has it been recommended as yet. These cases are brought to the attention of the Joint Committee on Professional Conduct. This Committee is sponsored by the American Society of Civil Engineers, Massachusetts Section; Boston Society of Civil Engineers; and the Massachusetts Society of Professional Engineers.

We have the basic machinery to keep a guiding hand on the profession. It is being utilized to a limited extent. Any person who is not satisfied with the efforts being made by the engineering societies should bring his complaints on violation of the engineering registration law or on a breach of ethics to the aforementioned committees.

Corrective Action Necessary

Lack of Professionalism in Engineers' Behavior

Many young engineering graduates ask why they should bother to take examinations for registration. The time, effort and expense involved is considerable. They observe that many more mature engineers working near them who, although registered, never are permitted to sign their name to the work, never have occasion to use their professional seals and seldom, if ever, are referred to in their places of

work as "engineers." This is a situation peculiar to the engineering profession and is almost never observed in the medical or legal professions.

One of the principal reasons for this lack of appreciation is in the use of titles. Many persons with full professional capabilities and often registered professional engineers are referred to in their places of work "designer," "chief draftsman," "stress analyst", "clerk-of-the-works," "inspector," and other designations which make no reference to engineer. On the other hand the titles "resident engineer," "project engineer," "safety engineer" and "right-of-way engineer" are applied to persons who many times are not registered and do not possess the qualifications for registration. We do not find architects making this mistake. They do not give a field inspector an architectural title, they refer to him as a "clerk-of-the-works."

It is enlightening to read the definition of "practice of architecture" as stated in the architects' registration law and this is as follows:

> "Performing or agreeing to perform or holding one's self out as able to perform professional services in connection with the design, construction, enlargement or alteration of a building including consultations, investigations, evaluations, preliminary studies, aesthetic design, preparation of plans, specifications and contract documents, the coordination of structural and mechanical design, and site development, supervision of construction and other similar service or combination of service in connection with the design and construction of buildings regardless of whether one or all of these services are being performed regardless of whether these services are performed in person or as the directing head of an office or organization performing them, provided that the practice of architecture shall not include the practice of engineering as defined in this chapter but a registered architect may perform such engineering work as is incidental to the practice of architecture."

It is evident from the preceding that the architects intend to protect their profession and to become the leader in any grouping of professionals in the construction industry.

One of the most serious and damaging conditions is one where professional engineers work in a department or firm headed by a non-professional. The architects have discouraged this practice by refusing membership in the American Institute of Architects to persons employed by engineering firms or other non-architectural firms.

A bill was filed in the Massachusetts legislature, last year, to require that persons placed in charge of departments in state, county,

and municipal governments where professional engineers or land surveyors are employed, be, themselves, registered in the field which is practiced in that department. This bill was supported by many engineers including the writer. Unfortunately it failed to pass. Non-qualified men making final engineering determinations, in many cases, can be the cause of inferior or defective work as well as damage to the professional status of engineering. We can find many examples of major engineering works in and about Boston where this sort of decision making has raised havoc with our transportation system.

Recognition of Other Professionals

The very limited use of the stamp or seal of the professional engineer in Massachusetts is a major factor in the evaluating the importance of professional standing to engineers. It is not uncommon in thumbing through a set of possibly fifty construction drawings to find them all stamped and signed by the same engineer on the same day. Many of the drawings may be of a specialty with which he may be almost totally uninformed and may freely admit to having little or no competence. The engineer may give the reason that he has confidence in the man who prepared the drawings or that he bears the contract and financial responsibility for the work involved.

Certain parts of the Model Law state that drawings shall be stamped by the engineers who prepare them. The wording of the statutes is not sufficiently clear. The references to responsible charge of preparation of engineering work needs to be clarified in the law. A similar statement to that previously described for the work of employees and subordinates would be satisfactory in this respect. The writer believes it to be a gross violation of the principles of engineering registration. It also results in shoddy work when qualified professionals are not engaged and held responsible for the heating, ventilating, structural, electrical or other specialties involved.

This work of other professionals has been a sore point with many building inspectors and other regulatory officials reviewing plans. They have in many cases made it a point to call on the person whose seal appeared on the drawing for explanation and defense of the designs appearing over their signature and seal. In addition, they have refused to talk to any others whose names and seals did not appear on the documents.

These regulatory officials have also sought to obtain legislation by

which it would be necessary to state the field of specialty alongside the professional stamp. This is a misinterpretation of the present law. Engineers, of course, are not registered in their specialty. They are registered as professional engineers to practice within the profession to the limit of their capabilities. This is the same in the medical profession and in the legal profession. Specialties are listed in the roster only to signify the fields of practice under which the engineers listed their experience for consideration in their registration application.

There is no question that the engineer has the moral obligation of employing co-professionals in fields in which he himself does not profess competence. How much better it would be if this were evidenced by the engineers in the various specialties in electrical, mechanical, structural and the like being recognized by having their seal appear on the drawings for which they are responsible.

Fortunately, the practice is growing among engineering offices to have two seals on engineering drawings and documents involving specialties other than the field of the person in responsible charge of the entire project. In other words, the section of the drawings dealing with electrical work would bear the seal of the engineer specializing in this field as well as the seal of the engineer in responsible charge of the entire project.

P. E. Designation

In many states it is common and sometimes required that registered engineers write PE after their signature. This is in the same manner as a medical doctor writes MD after his signature. Engineers, particularly in New England, are reluctant to engage in this practice which touches on a sort of self-aggrandizement. It may well be that overcoming this modesty is a practical means of assuring better recognition as a professional. It is certainly worthy of very serious consideration.

Effect of the Grandfather Clause

There is no question that one of the most serious obstacles to the advance of professionalism and to recognition of registration as a symbol of professionalism is the great number of engineering registrations obtained by the use of the "grandfather" provision. Although this escape clause was in effect for only six months at the outset of the mandatory registration law of 1958, in Massachusetts, it resulted in

a large number of unqualified persons assuming the title of registered professional engineers.

These persons taking advantage of the "grandfather" clause needed to demonstrate to the Board that they actually practiced engineering at the time the law was passed and also they had to furnish endorsements from at least three registered professional engineers. It is enlightening that about 1,500 of these applications are still pending. The applicants have been unable to obtain any registered professional engineers who are willing to perjure themselves in stating these men were competent. The professional responsibility involved in endorsing others for participation in his profession, should always be kept in mind.

The requirement of the "grandfather" clause was brought about by the constitutional provision that a person cannot be deprived of his property without due process of the law. The right of a person to earn his living through the practice of a profession or to engage in a trade is one of the most valuable of property rights. However this may be justified from a legal standpoint, its practical effects have been harmful to the profession. Fortunately, it is not required to grant reciprocity to persons who obtain their license by this method.

Attempts have been made to have this "grandfather" clause reinstated; it has been and should be fought strenuously each time by the engineering societies. This condition will remove itself from the scene in a single generation. It should not be a "crutch" on which to base our reasons that nothing can be done about improving the registration laws and upgrading the practice of engineering.

Corrections Necessary in Registration Law

Efforts to have Massachusetts Registration Law brought up to the standards of the 1960 Model Law have been unsuccessful to date. Apparently engineers are not willing to agree on the provisions. This is not unexpected as you seldom find two lawyers who agree on the exact wording of a bill or for that matter any group of engineers in a discussion rarely agree. It has been reported in the National Council of State Boards of Engineering Examiners that there were fifty engineers present at a discussion of the provisions of the model law and there were fifty different expressions of opinion. However, explanations and discussions should bring about sufficient agreement so that an improved measure can be filed in Massachusetts, at least by December,

1965. Other states have upgraded their engineering laws to this 1960 standard. More than half the states have provisions more stringent than Massachusetts.

The writer has been on a BSCE committee charged with recommendations for revisions to the Massachusetts statutes. Even this committee was not in complete agreement. However, a draft of the proposed revisions has been circulated among many interested engineers for discussion purposes. The suggested revisions are as follows:

1. New definitions of land surveyors and engineers-in-training.
2. Provisions of a paid full-time executive secretary for the Board of Registration.
3. Requirement for written examination, without exception, for both professional engineers and land surveyors.
4. Set passing grade of examination at 70 per cent.
5. Provide for work of employee or subordinate for work not done under direction of registrant. This includes Field Engineers and Resident Engineers as well as work in specialties other than that of registrant.
6. Provide that board may employ counsel.

All of these items have been drafted but the committee is not yet in agreement as to Item 5.

The Metropolitan Chapter of the Massachusetts Society of Professional Engineers has filed a bill to provide for only the full-time executive secretary and for an increase in dues to pay for same. No general agreement could be reached on the other revisions.

Summary and Conclusions

There is no question that the problems as usual are national in scope. Referring everything to the National Societies only results in a further stagnation. However, Massachusetts has led the way before and should again.

BSCE should join with ASCE Mass. Section and MSPE to form a formal discussion group with regular meetings to discuss and formulate courses of action on the following:

1. Revision of the Engineering Registration Laws to conform to the 1960 Model Law and to go beyond this if necessary to accomplish the desired goals as established in the areas of discussion.

2. Clarify the relationship which exists between engineering and science in the colleges, between technicians, graduate engineers, engineers-in-training and professional engineers.
3. Establish greater rapport between the practicing professional engineers and the professors and students in the colleges of engineering.

PHYSIOLOGICAL AND PSYCHOLOGICAL EFFECTS OF NOISE ON MAN

By Alexander Cohen, Ph.D.*

(Presented at a meeting of the Boston Society of Civil Engineers, Sanitary Section, on December 2, 1964.)

Introduction

Technological advancements in our daily living situations have given rise to at least one undesirable byproduct—excessive noise. Job situations have become noisier as the result of more mechanized equipment being used in plants and offices. Indeed, not too long ago, a noise survey of 200 workplaces in 40 different industrial plants found 50 per cent of the machines in use to produce noise levels believed intense enough to pose some hazard to the hearing of the exposed worker (1). Other probable adverse effects (e.g., interference in speech communication) were not considered in the context of this survey. In six industries alone (saw and planing, wood products, furniture and fixtures, fabricated metals, textiles, transportation) it has been estimated that the number of workers exposed to potentially damaging noise is 6,000,000 (2). Some experts even feel that the number of workers subjected to potentially harmful noise levels probably exceeds the number exposed to any other significant hazard in the occupational environment (3, p. iii).

Besides job situations, communities have become noisier. This is due to the spill-over of increased factory noise, the flyovers of jet and heavy propeller aircraft, and the increased volume of traffic along the nation's rapidly expanding highway system. The problem of excessive community noise is documented in terms of reports of localities taking action to curtail airport and aircraft activities (4, 5, 6) as well as to halt other real and potential sources of community noise (7).

Lastly, home situations have become noisier. This condition has been caused by the upsurge in the use of power appliances such as garbage disposals, dishwashers, and power lawnmowers. Although not in the same category as the aforementioned items, the booming hi-fi

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set and excited children at play have also contributed to the over-all noise problem in homes.

It is the aim of this presentation to describe and discuss the physiological and psychological effects stemming from this noise exposure.

Noise and Hearing Loss

By definition, noise is unwanted sound. It can cause hearing loss and other undesired physiological changes, interference in speech communication, and, of course, annoyance. As already noted, noise-induced hearing loss constitutes a major health problem in industry and for this reason will be given extensive treatment in this paper.

Exposing the ear to an intense noise will most probably cause hearing loss. This loss may be temporary, permanent, or a combination of the two. Temporary hearing loss, sometimes called temporary threshold shift or auditory fatigue, represents loss in hearing acuity which can occur after a few minutes exposure to an intense noise and is recoverable following a period of time away from the noise. Fig. 1

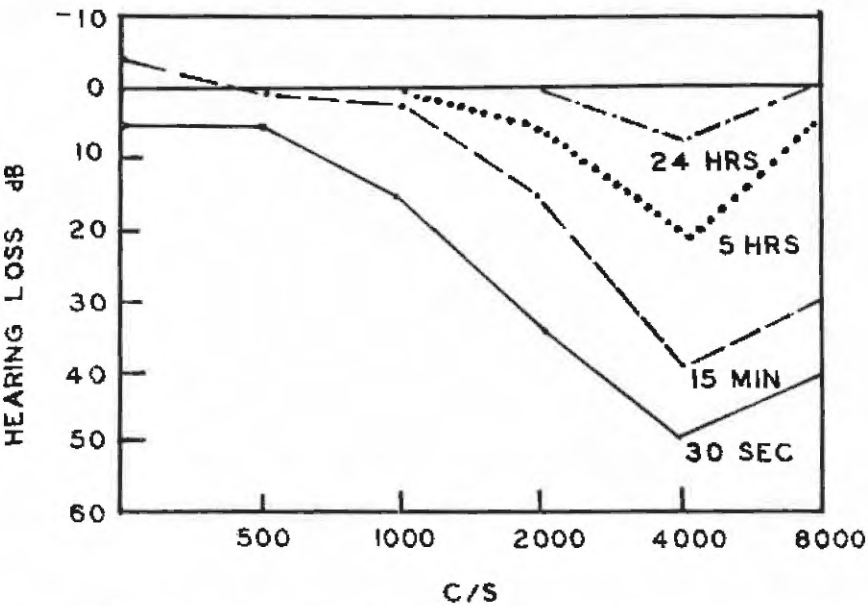


Fig. 1.—Changes in Hearing Threshold Levels, Relative to Pre-exposure Values, for Different Test Frequencies Measured at Various Times After a 20-Minute Exposure to 115 dB (Sound Pressure Level re 0.0002 Dyne/cm^2) Broadband Noise. (Unpublished Data).

shows hearing threshold levels, i.e., minimum audible sound levels, for different pure tone test frequencies measured at various times following a 20 minute exposure to an intense broad-band noise. The horizontal line drawn through the "O" hearing level value on the vertical axis represents a listener's pre-exposure threshold levels for hearing the different test frequencies. Differences between these hearing levels and those plotted on the remaining curves indicate the extent of the listener's loss at specified times following the noise exposure. Note how the differences between the pre- and post-exposure hearing levels diminish with increasing time away from the noise, therein depicting the recovery of the ear from this noise exposure.

With daily continuous exposures for months or years to intense noise, there may be only partial recovery of the observed loss, the non-recoverable or residual loss being indicative of a permanent noise-induced hearing impairment. Fig. 2 describes apparent permanent hear-

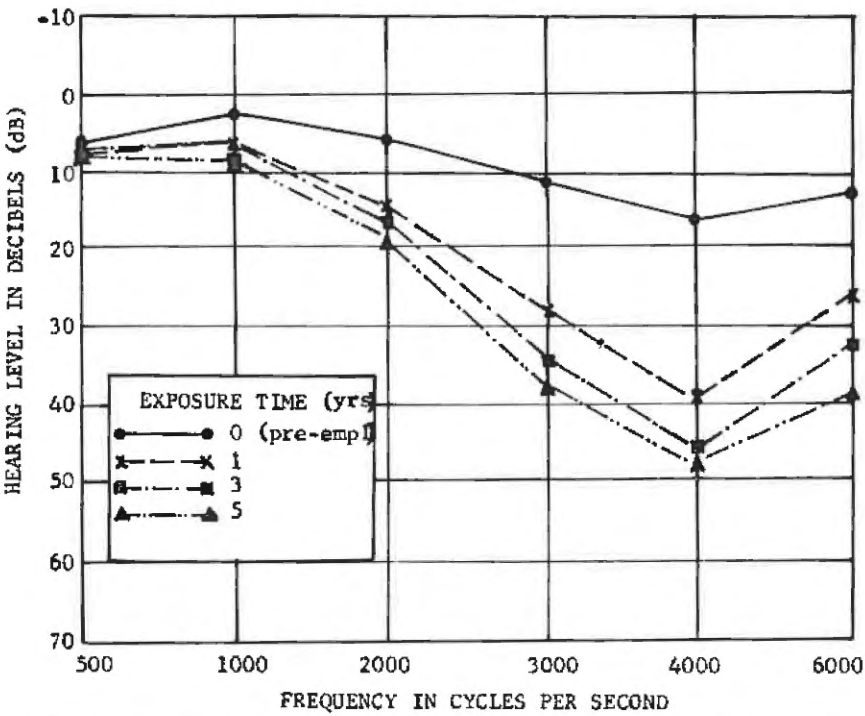


FIG. 2.—MEDIAN HEARING LEVELS FOR WEAVERS OBTAINED JUST PRIOR TO EMPLOYMENT (0 YEARS) AND AFTER VARIOUS YEARS OF EXPOSURE TO WEAVING AREA NOISE. DATA TAKEN FROM YAFFE AND JONES (3, 8).

ing losses for weavers as a function of their years of exposure to weaving room noise (3, 8). It is indicated that significant hearing losses occur first in the frequency range 3000 to 6000 cycles per second (cps) on the audiogram, with 4000 cps showing the most sizeable impairment. Losses in this frequency range are not believed critical to speech reception so that the individual may be completely unaware of the first signs of a noise-induced hearing loss. With longer exposure time significant hearing loss will also occur at frequencies below 3000 cps which can affect speech perception. Workmen's compensation laws for industrial hearing loss in several states presently regard only the hearing losses in the speech frequency range 500 to 2000 cps as being compensable (9, 10).

Complicating the evaluation of hearing loss due to noise is the fact that hearing acuity normally decreases with increasing age (11, 12). Further, the losses associated with age are quite similar to those caused by undue noise in that the hearing for high frequency sounds is most affected in both instances (compare Figs. 2 and 3). Consequently, how much of a given worker's hearing loss is caused by occupational noise exposure?—and how much is due to his age? Hearing data for different age and sex groups having had negligible noise exposure, such as shown in Fig. 3, are subtracted from the total hearing loss values obtained for noise exposed persons in order to leave a "purer estimate" of the amount of noise-induced loss. The hearing loss data plotted in Fig. 2 for the weavers has not been corrected for the age effects on hearing.

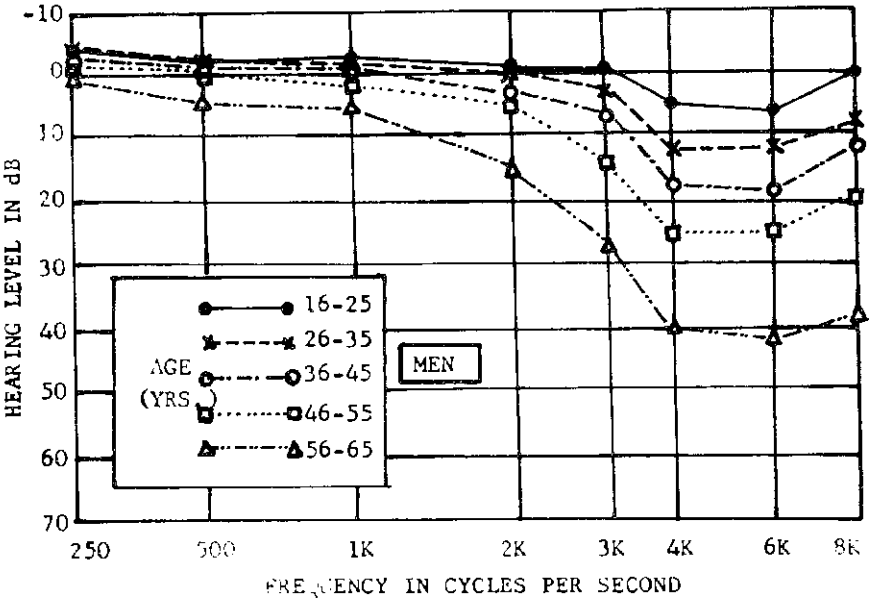
The factors believed to be critical in assessing the severity of noise exposure on hearing are the following:

1. The over-all sound level of the noise.
2. The spectrum of the noise.
3. The total duration of noise exposure.
4. The frequency and time distribution of noise exposure.
5. The susceptibility of the exposed individual's ears to noise-induced hearing loss.

Over-all Sound Level of the Noise

Airborne sound refers to alternate increases and decreases in atmospheric pressure. The amplitude of such changes relative to the resting or normal atmospheric pressure provides an indication of the strength of the sound. These pressure amplitudes are averaged[1] and

[1] A root-mean-square (RMS) average is used in these instances in which the positive and negative



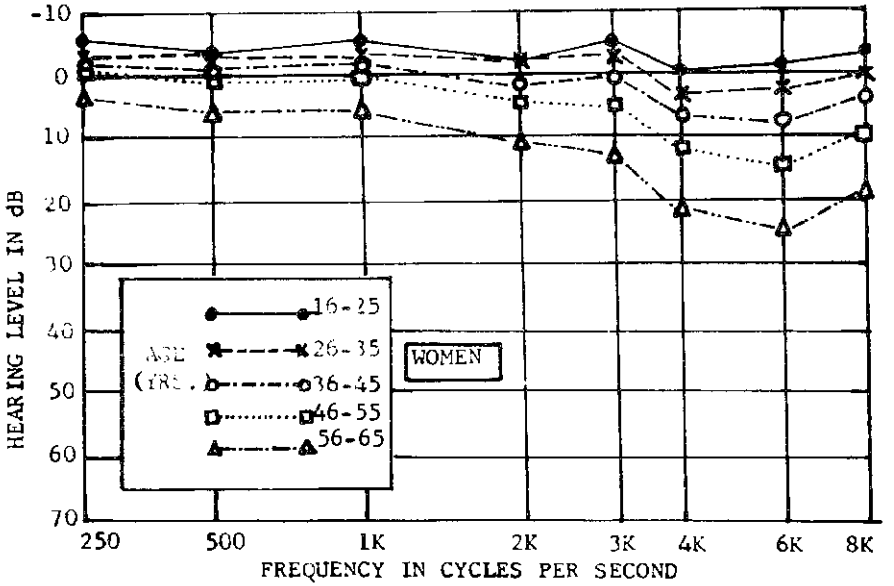


FIG. 3.—MEAN HEARING LEVELS FOR PERSONS HAVING MINIMAL NOISE EXPOSURE AS A FUNCTION OF AGE AND SEX. DATA ARE REPLOTTED FROM RILEY ET AL. (12).

quantified on a decibel scale in accordance with the formula:

$$N_{dB} = 20 \log_{10} \frac{P}{P_0}$$

where N_{dB} = number of decibels, P = the average of the pressure changes underlying the sound being measured, P_0 = a reference pressure, usually 0.0002 dyne/cm^2, which is the weakest audible pressure change that a healthy young ear can detect under ideal listening conditions.

Two important points to remember about sound pressure measurements on a decibel scale are as follows:

1. The decibel scale is a logarithmic scale such that while a change from 0 dB to 10 dB represents a 10-fold increase in sound energy, a 0 dB to 20 dB change represents a 100-fold increase in sound energy, a 0 dB to 30 dB change represents a 1000-fold increase in energy, and a 0 to 40 dB change corresponds to a 10,000-fold energy increase (see Table I).

2. Two decibel values cannot be added together directly. The combined decibel level when adding one 80 dB sound to a second 80 dB sound is not 160 dB. Actually, it is 83 dB.

Sound pressure levels in dB (re .0002 dyne/cm^2) are indicated for some typical noise sources in Table I. Also shown are the RMS pressure values equivalent to the specified decibel notations, and the relative changes in energy.

With reference to over-all sound levels of noise and hearing loss, it is believed that any amount of exposure to unprotected ears to noise in excess of 135 dB is hazardous and should be avoided (13). At the other extreme, exposure to noise whose over-all pressure level falls below 78 dB[2] will not generally produce significant temporary hearing loss in unprotected ears and therein is not assumed to cause any permanent hearing loss. Most industrial noise conditions fall between these two limits and require other information such as the noise spectrum and the length of exposure before a judgment can be made as to the potential harmfulness of the noise to hearing.

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pressure amplitudes (measured relative to the resting pressure level) are first squared and then added together. This sum is then divided by the number of pressure changes involved and a square root extracted from the result. In actuality, instruments designed to measure sound pressure (i.e., sound pressure level meters) carry out this computation. Sound levels measured by these instruments give readings in decibels based on comparing the RMS pressure change for the sound under study with the stated reference RMS pressure value of .0002 dyne/cm^2.

[2] Temporary threshold losses have also been found to occur for over-all noise levels below this value. Such changes, however, are not believed to be a fatiguing of the ear but rather a process of adaptation. See Selters (14) for a discussion of this point.

TABLE I

RELATIONSHIPS BETWEEN SOUND PRESSURE IN DYNE/CM^2, SOUND PRESSURE LEVEL IN DECIBELS (dB), AND ENERGY CHANGE

SOUND PRESSURE (DYNE/CM^2)	SOUND PRESSURE LEVEL IN dB		SOUND PRESSURE LEVEL IN dB	RELATIVE CHANGE IN ENERGY
200	120	PNEUMATIC CHIP HAMMER @ 5 FT.	120	10^{12}
100				
50	110	AUTOMATIC PUNCH PRESS @ 3 FT.	110	10^{11}
		CUT-OFF SAW @ 3 FT.		
20	100		100	10^{10}
10		SUBWAY TRAIN @ 20 FT.		
5	90		90	10^{9}
		SMALL TRUCKS ACCELERATING @30 FT.		
2	80		80	10^{8}
1		OFFICE WITH TABULATING MACHINES		
0.5	70		70	10^{7}
		CONVERSATIONAL SPEECH @ 3 FT.		
0.2	60		60	1,000,000
0.1				
0.05	50	PRIVATE BUSINESS OFFICE	50	100,000
0.02	40		40	10,000
0.01				
0.005	30		30	1,000
		BROADCAST STUDIO (MUSIC)		
0.002	20		20	100
0.001				
0.0005	10		10	10
0.0002	0	THRESHOLD OF HEARING	0	1

Noise Spectrum

Common types of noise as well as music and speech sounds are each composed of many different frequencies within the audible frequency range. The spectrum of a sound or noise refers to the manner in which the energy contained in the noise or sound is distributed across the component frequencies. In obtaining spectral measurements of noise for most purposes, the frequencies comprising the noise are filtered into eight frequency bands, each one octave in width, and sound pressures are determined for each band. The octave band frequency limits usually used are 37.5-75, 75-150, 150-300, 300-600, 600-1200, 1200-2400, 2400-4800 and 4800-9600 cps. Two noises having the same over-all level may differ in terms of the distribution of such energy when analyzed into octave bands. For example, Fig. 4 shows a forging ham-

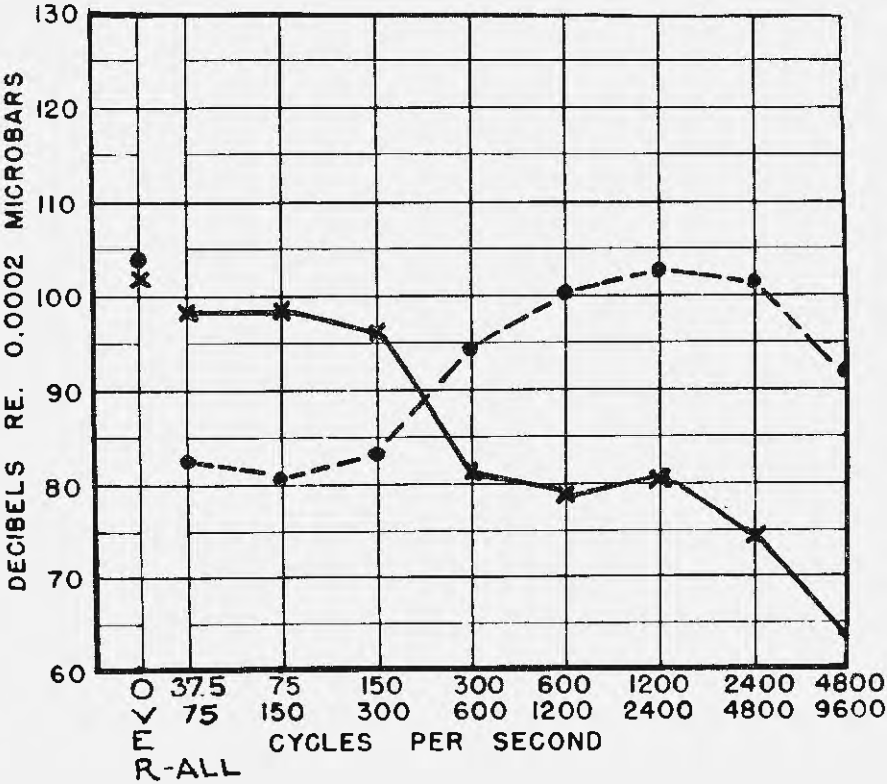

FIG. 4.—OCTAVE-BAND ANALYSES OF ANNEALING FURNACE (FILLED CIRCLES) AND PUNCHPRESS (X-POINTS) NOISE. DATA TAKEN FROM NOISE MANUAL—AMERICAN INDUSTRIAL HYGIENE ASSOCIATION, DETROIT, MICHIGAN, APRIL, 1958, p. 1-5.

mer and annealing furnace to produce noise having the same over-all sound level—about 102 dB. The annealing furnace, however, has most of its energy in the octave bands above 1200 cps while the forging hammer has most of its energy in those bands below 1200 cps.

With reference to noise spectra and hearing loss, it is believed that noises containing large concentrations of energy in the higher octave bands (above 1200 cps) are more damaging to hearing than those noises which have energy concentrations in the lower octave bands (15, 16). Fortunately, noise control procedures are quite effective in reducing the more damaging high frequency sounds (17, 18). They are least effective in suppressing the less harmful low frequencies.

Total Duration of Exposure

As the total exposure time to a noise increases so too does the likelihood that the noise will cause a permanent hearing impairment. Recently, however, it was shown that the amount of permanent hearing loss at 4000 cps stemming from daily exposures of five to eight hours to noise reached a maximum at about 12 years of exposure (19). Further losses at this frequency with continuing exposure appeared to be due to the aging process (presbycusis). This finding is believed particularly significant since 4000 cps on the audiogram usually shows the greatest loss from industrial noise exposure. An incidental finding here was that the maximum permanent hearing loss at 4000 cps after 10 years' exposure was correlated to the temporary hearing loss noted at the same frequency for a new employee group after their first day's exposure to the same noise conditions. This would suggest that temporary hearing loss might be correlated with permanent hearing loss and therein serve as an indicator of the susceptibility of a given ear to noise-induced hearing loss. The relationship between permanent and temporary hearing loss is still not firmly established, however, and to some experts appears untenable (20).

Frequency and Time Distribution of Noise Exposure

In many instances industrial noise exposure for a given worker is not continuous or of a constant nature. More typically, the worker is intermittently exposed to noise conditions which are fluctuating in level as well as spectra. Under these conditions, it is difficult to quantify accurately the amount of noise impinging upon the worker so that some judgment can be made about the harmfulness of the exposure. Much like a radiation badge, noise dosimeters are now being developed which can be worn by an individual for purposes of totalizing the amount of sound energy to which he is exposed. Through read-out devices and some routine computation, the dosimeter provides decibel readings for sound energy averaged over any desired time period. These more definitive estimates of intermittent noise exposure, when correlated with hearing loss, will provide a basis for establishing more realistic noise limits for such exposure situations. At the present time there is little agreement as to the establishment of noise exposure limits for intermittent exposure conditions (21). Most experts do agree, however, that less hearing loss will occur from an intermittent noise exposure than from a continuous one of the same total energy (22). This

would suggest that rotating a worker in and out of a continuous type noise situation will provide some protection to his hearing.

Susceptibility of the Individual

The magnitude of both temporary as well as permanent hearing loss for the same amount of noise exposure may vary greatly from one person to another. For this reason, some effort has been directed toward developing a technique which will identify the noise-susceptible or weak ears. Several investigators have considered the nse of temporary threshold losses following a standardized noise exposure test as a susceptibility indicator (23, 24). Those persons showing the greatest temporary losses from the exposure would be considered as most susceptible to permanent hearing losses and consequently would be placed in non-noisy work areas or otherwise protected. This procedure is not fully accepted at this time because of insufficient data concerning the relationship between temporary and permanent hearing loss. Measures of temporary hearing loss are being used, however, to check whether workers in noisy areas are using the ear protection issued to them. Individual workers may be selected for spot audiometric tests. If excessive threshold losses are noted relative to some previously obtained hearing values, it is considered as evidence that ear protection is not being used.

At present, attempts are being made to establish valid and acceptable noise tolerance criteria for preventing hearing damage. The need for such criteria in industry cannot be over-emphasized. They could serve as guidelines for (1) determining the noise reduction needed to create an acoustically safe work environment, (2) the establishment of hearing conservation programs which would include monitoring audiometry so as to identify early those ears which are unusually susceptible to noise-induced loss, and (3) fair rulings in court cases involving compensation claims for noise-induced hearing loss.

Many proposals for damage risk criteria to noise exposure have been made which are intended to minimize the risk of noise-induced hearing loss in a vast majority of the exposed worker population for their working lifetime (13, 24, 25, 26, 27, 28). Several of those criteria governing 5 – 8 hour continuous exposure to noise on a daily basis are shown in Fig. 5.

Each of the plotted curves represents maximum octave band limits for noise exposure and, if exceeded, would prompt recommendations for

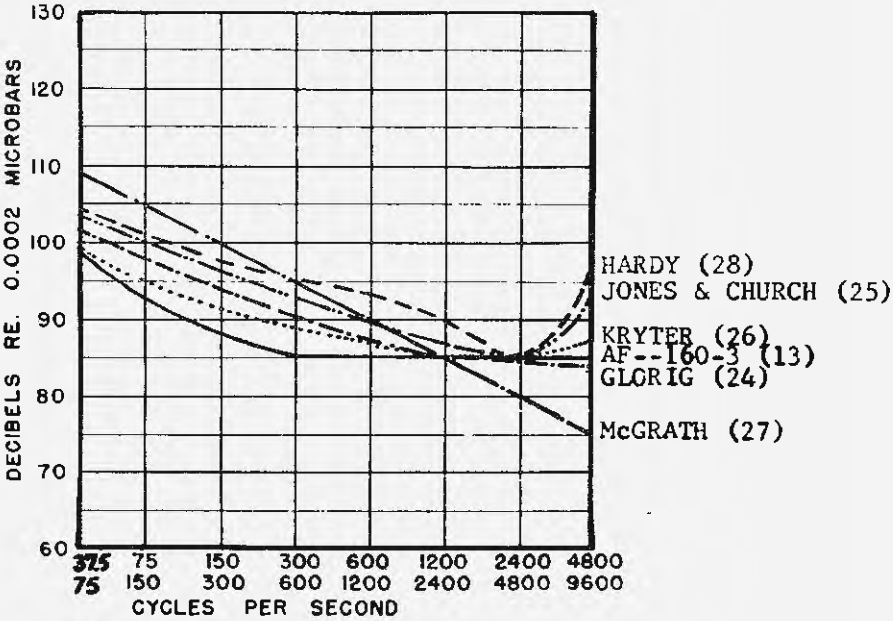

FIG. 5.—PROPOSED LIMITING OCTAVE-BAND PRESSURE LEVELS (IN dB) FOR PREVENTING HEARING LOSSES FROM CONTINUOUS 5-8 HOUR DAILY EXPOSURES TO NOISE FOR A WORKING LIFETIME.

ear protection or other means of noise control. While not too different, a given noise condition could be considered safe in applying one criterion and unsafe in applying another. Which of these various criteria has most merit in reducing the risk of noise-induced hearing loss, while not, of course, requiring needless amounts of noise control, remains to be completely determined. Answers to this question are sought in studies correlating hearing loss with long term exposure to noise conditions defined as safe or unsafe by the various criteria. The U.S. Public Health Service (3) conducted such a study in prison industries where groups of inmates experiencing known occupational noise conditions were given pre-employment and follow-up hearing tests at regular intervals to determine apparent hearing changes with duration of exposure. Actually, the prison situation was ideal for controlling not only the industrial noise exposure but also the noise occurring when the men were not working. On the other hand, exposures to the industrial noise could not be studied on the same men for more than 5 years because of the turnover of inmates in the prisons.

Table II summarizes the prison study data, showing the typical shifts in hearing threshold levels (relative to the pre-exposure values) for test frequencies 500 to 6000 cps following 2 to 5 years of exposure to the noise found in the specified work situations. Coupled to the hearing data obtained for each industrial noise situation is a notation (right hand margin) of whether that noise condition exceeded any of the criteria shown in Fig. 5. In all instances, a criterion was considered to be exceeded if any one octave band level of the specific noise under observation was greater than the octave limits imposed by the criterion. It is indicated that when no criteria are exceeded, the threshold losses at any frequency rarely are more than 10 dB. On the other hand, noise conditions which exceed all of the criteria are associated with quite sizeable shifts, especially for frequencies 3000, 4000 and 6000 cps. As already noted, severe losses at these frequencies mark the early stages of noise-induced hearing loss. With longer exposure to the same condition, significant losses in hearing would also be expected to occur at lower frequencies where hearing for speech could be impaired. Note that comparatively large threshold losses at high frequencies occur for several noise conditions where all criteria were exceeded with the exception of one (Hardy). This suggests that the limits imposed by this criterion may not always provide sufficient protection against noise-induced hearing loss.

Based on a recent poll of experts in the noise and hearing field (21), it is generally contended that a noise whose sound pressure levels fall below 85 dB in the octave bands 300-600, 600-1200, 1200-2400, 2400-4800, 4800-9600 cps, poses no significant risk of hearing damage for fairly continuous daily noise exposures for a working lifetime. For these same exposure durations, the experts believe that the permissible sound pressure levels for the lower octaves 37.5-75, 75-150, 150-300 cps can be somewhat higher than 85 dB. They disagree markedly, however, on criteria for intermittent noise exposure and also on limits for noise conditions in which a strong tone may be present in the noise field (e.g., compressor whine, transformer hum). More data relating noise exposure to hearing loss, given more accurate characterization of the intermittency and nature of the noise condition, will be needed before the acceptable criteria for the latter types of situations can be prescribed. The lack of information regarding the effects of impact noise on hearing also make no judgment possible at this time regarding limits for these types of exposure.

TABLE II

RELATIONSHIP OF PROPOSED NOISE TOLERANCE CRITERIA TO MEDIAN HEARING LEVEL SHIFTS AFTER YEARS OF EXPOSURE TO NOISE FOUND IN PRISON INDUSTRIES

Department	Exposure in Years	Median Hearing Level Shifts in Decibels*						Criterion Exceeded
		500	1000	2000	3000	4000	6000	
Cotton Mill (Spin)	4	0.5	0.5	1.0	3.0	2.5	1.0	All except Hardy
Cotton Mill (Twist)	2	1.0	3.5	3.0	12.5	18.0	12.5	All except Hardy
Cotton Mill (Weave)	5	1.5	5.0	6.5	26.0	28.5	20.5	All
Woolen Mill (Spin, Finish)	2	—1.5	—2.5	0.5	8.5	1.0	5.0	None
Woolen Mill (Weaving)	2	1.5	2.0	9.5	14.5	21.5	11.0	All except Hardy
Shoe Factory (Fitting)	4	0.5	4.0	3.0	4.0	6.5	7.0	None
Shoe Factory (Lasting cutting)	3	2.5	1.5	2.0	1.0	1.0	2.5	None
Shoe Factory (Making, Tracing)	4	2.5	2.5	3.0	6.0	5.0	3.5	Glorig
Brush Factory	5	3.0	3.0	5.0	4.5	0.5	20.0	None
Furniture Mills	3	5.5	2.0	2.5	28.0	26.5	25.0	All except Hardy
Furniture (Cabinets)	2	1.0	—1.5	0.0	3.5	10.5	—0.5	None
Printing Factory	3	—0.5	2.0	4.0	2.5	—1.0	2.0	None
Clothing (Tailoring)	3	1.5	2.0	0.5	1.0	—1.0	—1.0	None

NOTE: This table was previously published in an article by Cohen (21, p. 235).
* Threshold shift with negative sign indicates hearing is better after exposure than before.

Effects of Noise on Other Physiological Responses

Aside from damage to the hearing mechanism, noise conditions found in industry are not considered to produce any other physiological impairments. It should be mentioned, however, that intense noise of sudden onset will cause marked physiological changes including a rise in blood pressure, increase in sweating, changes in breathing and sharp contractions of muscles in the body. These changes are generally regarded as an emergency reaction of the body, increasing the effectiveness of any muscular exertion which may be required. While perhaps desirable in emergencies, these changes are not wanted for long periods since they would interfere with other necessary activities or produce undue amounts of fatigue. Fortunately, these physiological reactions subside with repeated presentations of the noise.

It has often been stated that in order for performance on a task to remain unimpaired by noise, man must exert greater effort than necessary under more quiet conditions. Measures of energy expenditure, e.g., oxygen consumption, pulse rate, muscle potential, do show changes in the early stages of work under noise conditions which are indicative of increased effort. With continued exposure, however, these responses return to their normal level (29, 30).

Sounds of tremendously high intensity level (over 140 dB) are capable of causing dizziness or loss of equilibrium since the balancing organs (semi-circular canals) are being stimulated. Such high intensity exposures may also cause alterations with other types of sensory behavior, e.g., the eyeballs may flutter in the noise field, and will definitely cause pain, perhaps even traumatic damage in unprotected ears. Examples of such extreme noise conditions are few; possibly in jet engine test cells would these high levels be reached.

Noise and Speech Interference

The most demonstrable effect of noise on man is that it interferes with his ability to use voice communication. A noise which is not intense enough to cause hearing damage may still disrupt speech communication as well as the hearing of other desired sounds. Obviously, such disruption will affect performance on those jobs which depend upon reliable voice communication. The inability to hear commands or danger signals due to excessive noise may also increase the probability of accidents.

Averaging the readings in decibels for the three octave bands 600-1200, 1200-2400, and 2400-4800 contained in a wide-band noise has

empirically been shown to provide an indication of the ability of that noise to affect the intelligibility of voice communication. The average of these three octave band dB values is called the speech-interference-level (SIL). In noises whose spectra yield an SIL greater than 75 dB, personnel would have to speak in a very loud voice and use a selected and possibly prearranged vocabulary to be understood over a distance of one foot. Telephone use under these noise conditions would be impossible. Noise conditions having an SIL between 65 and 75 dB would permit barely reliable communication over two feet with a raised voice. This span of communication would be extended to four feet by using a loud voice and to eight feet by shouting. Telephone conversation under 65-75 dB SIL conditions would be difficult. In noise having an SIL between 55 and 65 dB, a normal voice level could communicate effectively over a distance of three feet, a raised voice over a distance of six feet, a very loud voice over a distance of twelve feet. Telephone use here would be practically unimpaired. An SIL of 55 dB or less would be permissible in large business or secretarial office areas. An SIL of 45 dB or less would be desirable for private offices or conference rooms. Table III indicates maximum permissible values for different rooms or areas where speech communication is going to be a major function.

Impairments in Performance (Efficiency)

Contrary to popular thinking, there is little evidence to support the notion that noise degrades performance. Laboratory studies of this

TABLE III
Speech-Interference-Level (SIL) Criteria for Different Room Areas

Type of Room Area	Maximum Permissible SIL (Measured While Room is Not in Use)
Small private office	40
Conference room for 20	30
Conference room for 50	25
Movie theatre	30
Theatres for drama (500 seats, no amplification)	25
Sports coliseum (amplification)	50
Concert halls (no amplification)	20
Secretarial offices (typing)	55
Assembly Halls (no amplification)	25
School rooms	25

NOTE: This table is taken from Peterson, A.P.G. and Gross, E.E. Handbook of Noise Measurement, General Radio Company, West Concord, Massachusetts, 1963, p. 4.

problem (summarized in 31, 32) have shown that tasks involving simple, repetitive operations are not affected by noise. While efficiency in more complex tasks may be initially decreased by noise, such effects tend to vanish as exposure time and/or practice on the task increases. There have been reports (32, 33, 34) however, which show noise to cause significant losses on vigilance-type tasks. Such tasks require the subject to keep a constant watch over a number of dials or indicators so as to report changes which may occur on any dial at any time. Noise-related losses in vigilance performance are important because of their implications for automated jobs which involve the monitoring of control panels with many indicators displaying information about an ongoing machine process. Recently completed research on vigilance at this Facility (35) has found such performance to be quite high and unaffected by variations in background noise when a large number of signals are presented for detection on a 10-dial display. These results might suggest that an increased signal rate improves vigilance performance to the extent of overcoming the adverse effects which certain conditions might otherwise have on this type of task.

Deserving of more interest in laboratory studies of noise and performance is the typical marked inter-subject variability revealed in the data. This variability may be due to individual differences in attitudes toward noise, in physiological reactivity to noise, in ability to adjust to noise, or in motivation to overcome the stressful effects of the noise. In exploring some of these factors, the noise-vigilance study described above related performance of the subjects in noise with (a) their noise tolerance as determined by objectionability ratings to a set of laboratory generated noises, and (b) specific personality measures (extroversion-introversion, manifest anxiety, neuroticism) as obtained from a standardized personality questionnaire (Minnesota Multiphasic Personality Inventory). The poorest vigilance performers in noise were found to be less tolerant of noise and showed greater tendencies toward extroversion and neuroticism than the best performers in this test situation. Another study conducted at this Facility (35) found that subjects showing the greatest physiological changes to noise (measured by galvanic skin response) tended to give better performance in noise on a set of learning or practice trials requiring essentially repetitious behavior. This heightened physiological response, however, tended to impair performance when the task was switched. These results would suggest that accurate predictions of the effects of noise on work efficiency would depend upon a fuller appreciation of the physiological

and psychological factors operating in the situation. Giving further support to this contention are the results of field studies concerned with efficiency effects associated with changes in noise conditions. Some investigations have noted that increased output has resulted from noise reduction in work areas (summarized in 31, 32, 36). This improved performance level was maintained, however, with the restoration of the original conditions. The effects on performance in these cases are probably due to morale changes. That is, the workers see that an interest is being taken in them or their working conditions and respond with increased effort, leading to greater output. The fact that field studies cannot control factors such as morale, motivation, worker attitudes toward job or supervisor makes it difficult to obtain valid and reliable data reflecting the effects of manipulation of the occupation noise conditions upon performance (37). For the same reasons, it is difficult to establish cause and effect relationships between industrial noise conditions and accident rate, absenteeism, and employee turnover.

Noise and Annoyance

Perhaps the most widespread reaction to noise is that it is annoying. What constitutes an annoying sound, however, is not an easy question to answer since noise-annoyance judgments depend upon many factors besides the acoustical stimulus. For example, a sound may be judged annoying because it has unpleasant association to an individual. In a poll dealing with the annoyance of aircraft noise in a community near an airport, 80 per cent of those residents complaining of the aircraft noise also reported some fear in connection with airplanes, either fear of the planes crashing into their homes, or else unwillingness to fly themselves (38). A sound may also be considered as annoying on the basis of whether it is believed necessary. In a survey of British homes, 10 per cent of the residents were troubled by the noise of delivery trucks in a neighborhood as compared with 40 per cent who complained over the less intense noises produced by the neighbor's pets (39). A sound may be judged annoying if it is inappropriate to the activity at hand. Complaints to noise in communities impacted by noise are more numerous in the evening, presumably because sleep and relaxation are being interfered with. Conversely, an individual will tolerate certain sounds if there is an advantage associated with them. The comforts derived from air-conditioning apparently outweigh the noise produced by such units. The economic values to the community of nearby factories or airports may partially offset the

noise-nuisance produced by such noise sources. Along with the above factors, there are many differences in individuals with regard to their ability to tolerate noise. Some individuals complain about all kinds of noise, indeed any kind of annoyance. One study has reported, for example, that many people who were greatly affected by aircraft noise were preoccupied with other physical problems in their communities including other kinds of noises, litter, air pollution (summarized in 32). There is some support for the notion that people who have adjustment problems also seem to be more affected by noise than others (40).

Turning to the stimulus itself, there appear to be some basic characteristics of sound which can be considered as more annoying than others. These characteristics are as follows:

1. Loudness—the more intense and consequently louder sounds are more annoying.
2. Pitch—a high pitch sound, i.e., one containing high frequencies is more annoying than a low pitch sound of equal loudness.
3. Intermittency and irregularity—a sound that occurs randomly in time and/or is varying in intensity of frequency is judged more annoying than one which is continuous and unchanging.
4. Localization—a sound which repeatedly tends to change in location or point of origin is less preferred than one which remains stationary.

At the present time, extensive interest is being directed toward identifying which measure or measures of noise best correlates with annoyance reactions.

For office conditions, speech interference level values and loudness level determinations (these values represent the decibel level of a 1000 cps tone judged equal in loudness to the sound or noise in question) correlate well with subjective ratings of annoyance (41).

A new measure called perceived noisiness in decibels (PNdB) (42) has been found to agree well with subjective ratings of the acceptability of flyover aircraft noises. This measure takes into account the octave band intensity levels of the noise in question and adjusts them in terms of data showing equal annoyance judgments for different bands of noise. Some noise criteria for airport operations are specified in terms of PNdB. JFK Airport (formerly Idlewild) for example, has a noise ceiling of 112 PNdB for all aircraft operations as measured under the flight path of outgoing or incoming aircraft at one-fourth mile from the end of the runway.

Besides PNdB computations, still other procedures have been proposed to convert the physical measurements of a noise into numerical expressions of annoyance level. Specifically, conversion to loudness measures in sones or phons as developed by Stevens' (43) or by Zwicker's (44) technique are quite popular for noise annoyance quantification. The assumption in using loudness formulations for rating noise-annoyance is that loudness is the chief determinant in annoyance judgments. Also A-scale sound level values read directly off a conventional sound pressure level meter have been frequently used to provide numerical expressions of noise-annoyance conditions (45). Inherent to the A-scale readings as well as the conversion procedures noted are weighting schemes which reflect, in various ways, established relationships between the physical dimensions of sound (primarily frequency and intensity) and associated auditory reactions, both psychological and physiological. (A discussion of the relationships underlying the various conversion procedures is found in Ref. 44, 46).

At present much research is being done in an attempt to determine which of the various methods just described can best serve to index noise-annoyance for the possible range and variety of community noises encountered. In a recent study at this Facility (46), the annoyance levels of recorded samples of roadway noise, aircraft flyover noise and train noise, as computed by the various proposed procedures, were each correlated with listener's annoyance ratings of the noises. In this investigation, samples of roadway, aircraft and train noise were presented in pairs; 100 listeners having to judge which of the two noises in each pair was more objectionable. Such judgments yielded scaled objectionability ratings[3] for the noise samples which were then correlated with A-scale sound level readings of the noises in dB, and with conversions of their spectral measurements into loudness magnitudes in phon units, as computed by Stevens' and by Zwicker's techniques, and into perceived noise level (PNdB) as determined by Kryter's procedure. Fig. 6 plots the scaled ratings of noise annoyance against the measures derived from the various proposed procedures. A plot of noise ratings vs. the unweighted sound pressure levels (C-

[3] Paired comparison judgments of the noise samples supplied scale values which not only gave the relative rank of each noise in terms of its objectionability, but also indicated the size of the intervals between successive ranks. This latter scale property is meaningful. For example, the size of the interval between the noises ranked first and second in objectionability is liable to be much greater than the intervals between the noises ranked second, third and fourth. This would suggest that the first ranked noise is clearly more objectionable than the next three ranked samples which, in turn, are quite close to one another in objectionability. The scale just described however, has no absolute zero and its graded units are arbitrary. See Guilford (47) for a discussion of paired comparison procedures and scaling.

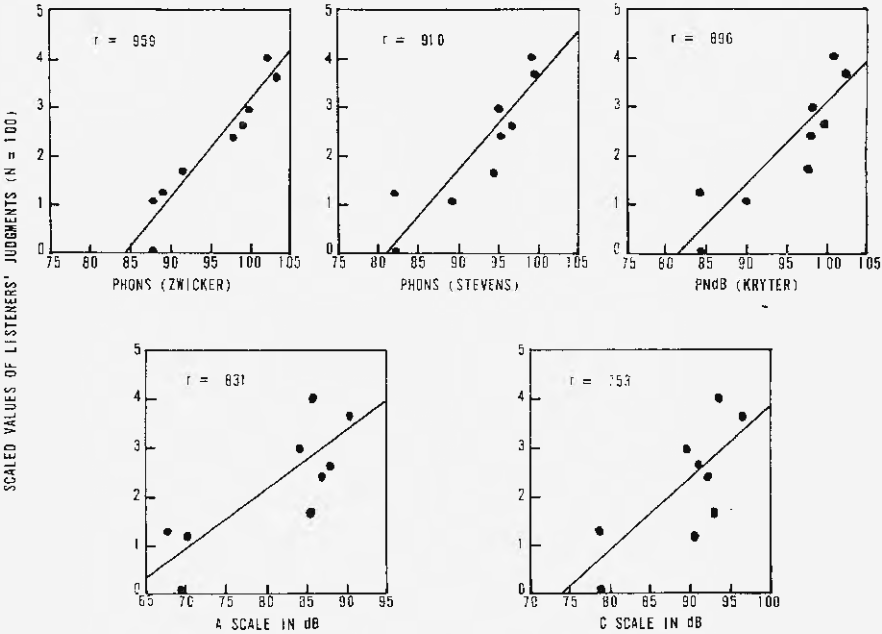

FIG. 6.—SCALED OBJECTIONABILITY RATINGS FOR SAMPLES OF AIRCRAFT FLYOVER, TRAIN, AND ROADWAY NOISES PLOTTED AGAINST MEASURES OF LOUDNESS (ACCORDING TO ZWICKER (44) AND STEVENS (43)), PERCEIVED NOISE LEVEL (ACCORDING TO KRYTER (42)), AND A- AND C-SCALE (OVER-ALL) READINGS FOR THE VARIOUS NOISES.

scale) of the noises is also shown. The relationship of Zwicker's measures of loudness to the scaled objectionability ratings deviated least from a straight line fitted to the data by the least squares method and, accordingly, had the highest observed correlation coefficient ($r = .96$). This indicated that of the different noise annoyance indices under evaluation, Zwicker's loudness values most closely correspond with subjective judgments of noise annoyance. Relationships between the scaled noise ratings and Stevens' loudness and Kryter's perceived noisiness values also showed fairly good agreement, yielding correlation coefficients of .91, and .89 respectively. The A-scale readings showed lesser correspondence with the objectionability ratings of the noise samples used in the study ($r = .83$). As expected, plotted values relating over-all noise levels (C-scale) to the objectionability ratings for the noises showed the most deviation from the least squares fitted line, therein indicating the poorest degree of correspondence and the lowest observed correlation ($r = .75$).

It must be emphasized again that the procedures under evaluation here can give only limited prediction of community noise nuisance because they only consider the physical characteristics of the noise stimulus itself. Other factors—social, personal, economic—must also be taken into account in making such predictions. Several models now exist which consider the physical characteristics of the noise together with known social and psychological factors in estimating the complaint potential of a noise to a community or neighborhood (48, 49). One of these models is described in the Appendix. The accuracy of the predictions made by this and other models has still not been sufficiently determined.

Summary

Adverse effects of noise on man include temporary and permanent hearing loss, speech disruption, loss in performance capacity, and annoyance. Factors believed critical in evaluating a potential noise hazard to hearing are the over-all level, the spectrum of the noise, total exposure duration, time and frequency distribution of short term exposure periods, the susceptibility of an individual's ears to noise-induced hearing loss. Specifications for valid damage risk criteria for noise exposure must take account of these factors. Measures for predicting speech interference of noise are available and have been used as a guide for establishing limiting noise conditions in rooms where effective speech communication is needed. Annoyance reactions to noise are based upon both acoustic and non-acoustic considerations. Models and measures for predicting noise-nuisance are available but require validation.

APPENDIX

The following procedure, developed by Stevens, Rosenblith and Bolt (48), is intended to predict the probable nature of neighborhood reactions to noise taking into account the physical acoustics of the noise as well as other factors of a psychological and sociological nature. The procedural steps are as follows:

1. Develop initial rank for noise spectrum.

a. Obtain octave band analysis for noise condition under evaluation, preferably using numerous measurements at the property lines of the closest residences and deriving an average spectrum for the subject noise.

b. Superimpose this spectrum on the family of curves shown in Fig. 7 which define the level of rank of the curve of the noise spectrum.

The level rank for a given spectrum is that letter (from A (low) to M (high)) corresponding to the highest zone into which any part of the spectrum protrudes.

2. Adjust noise level rank for other influencing factors.

a. Determine for the noise characteristics and neighborhood in question, correction numbers to take account of different conditions as shown in Table IV.

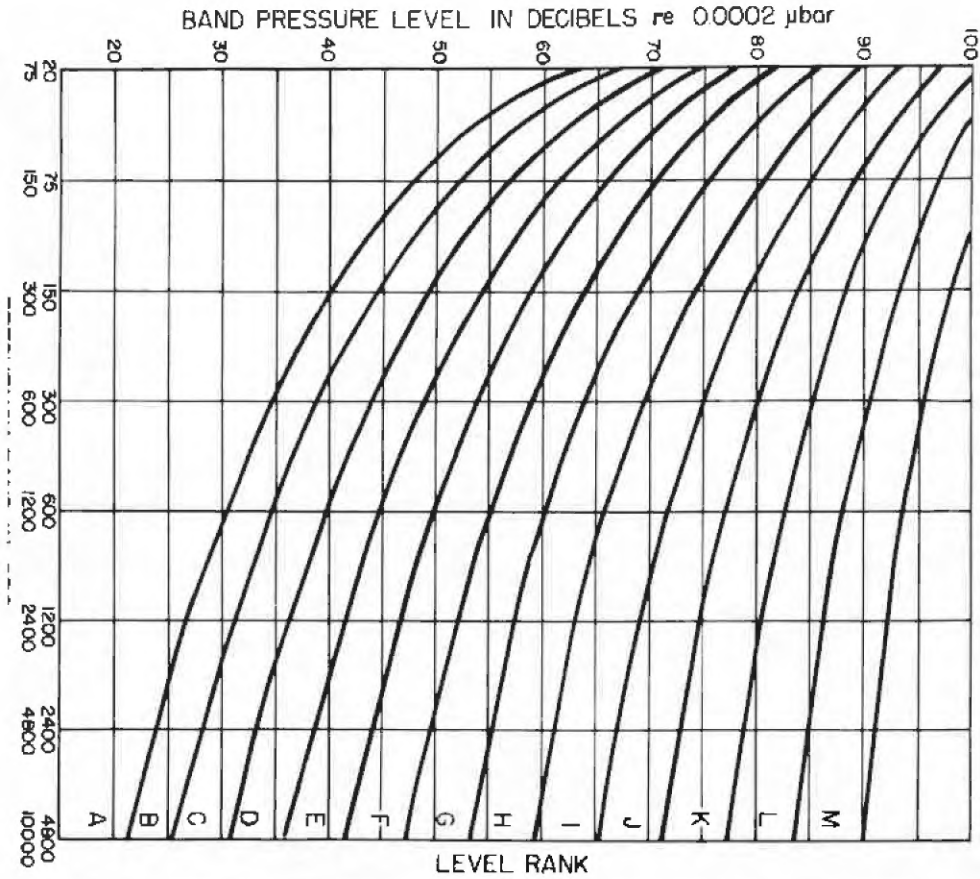

FIG. 7.—SET OF CURVES FOR ASSIGNING A LEVEL RANK TO A COMMUNITY NOISE. THE OCTAVE-BAND LEVELS OF THE NOISE UNDER EVALUATION ARE PLOTTED ON THIS CHART. THE HIGHEST OF THE ALPHABETICALLY LABELED ZONES INTO WHICH ANY OF THE BAND LEVELS PENETRATES IS THE LEVEL OF THE NOISE. CHART TAKEN FROM STEVENS, K. N., ROSENBLITH, W. A., AND BOLT, R. H. (48, p. 66).

b. Depending on such conditions, add or subtract the appropriate correction values, and obtain their algebraic sum to derive a net correction factor.

c. Obtain a corrected noise level rank by adding or subtracting the number of level ranks indicated by correction factor obtained in Step 2b from the original noise level rank defined in Step 1b. Thus, if the original noise level rank was E, and the correction factor was -2, the two ranks would be subtracted from E leaving a level of C. This rank (C) would be the corrected level rank.

3. Determine predicted neighborhood response.

a. Identify the point corresponding to the corrected noise level rank on the horizontal axis of Fig. 8. Follow a vertical line from this point to the shaded

TABLE IV
LIST OF CORRELATION NUMBERS TO BE APPLIED TO LEVEL RANK TO GIVE NOISE RATING

Influencing Factor	Possible Conditions	Correction Number
Noise Spectrum Character	Pure-tone components	+1
	Wide-band noise	0
Peak Factor	Impulsive	+1
	Not Impulsive	0
Repetitive Character (about one-half minute noise duration assumed)	Continuous exposures to one per minute	0
	10-60 exposures per hour	—1
	1-10 exposures per hour	—2
	4-20 exposures per day	—3
	1-4 exposures per day	—4
	1 exposure per day	—5
Background Noise	Very quiet suburban	+1
	Suburban	0
	Residential Urban	—1
	Urban near some industry	—2
	Area of heavy industry	—3
Time of Day	Nighttime	0
	Daytime only	—1
Adjustment to Exposure	No previous conditioning	0
	Considerable previous conditioning	—1
	Extreme conditioning	—2

NOTE: Table taken from Stevens, K. N., Rosenblith, W. A. and Bolt, R. H. (48, pp. 67-68).

area indicated. The vertical range of this shaded area for that rank, when referred to the vertical axis on the left, will indicate the probable expected neighborhood response to the noise under study.

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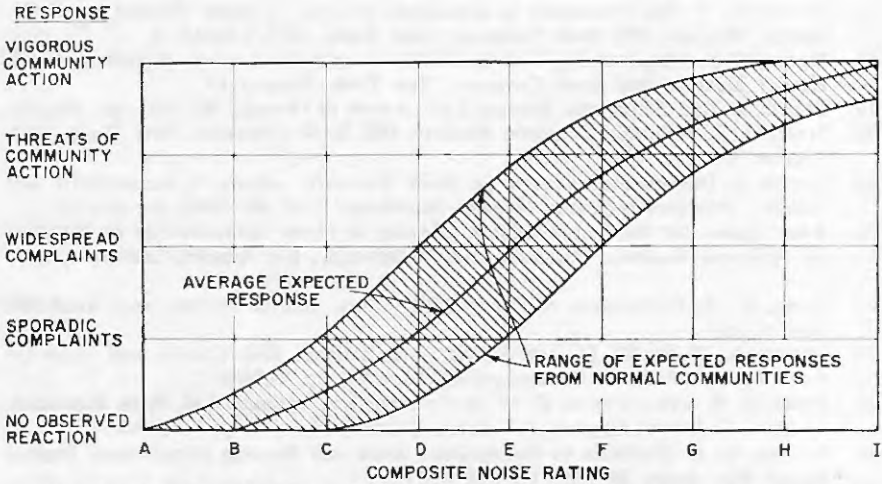

FIG. 8.—RELATION BETWEEN CORRECTED NOISE LEVEL RANK AND EXPECTED RESPONSE FROM THE RESIDENTS EXPOSED TO THE NOISE. GRAPH TAKEN FROM STEVENS, K. N., ROSENBLITH, W. A., AND BOLT, R. H. (48, p. 64).

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OF GENERAL INTEREST

PROCEEDINGS OF THE SOCIETY

MINUTES OF MEETING

Boston Society of Civil Engineers

NOVEMBER 18, 1964:—A Joint Meeting of the Boston Society of Civil Engineers with the Structural Section was held this evening at the Smith House, 500 Memorial Drive, Cambridge, Mass., and was called to order by William A. Henderson, President.

President Henderson stated that the Minutes of the previous meeting October 21, 1964 would be published in a forthcoming issue of the Journal and that the reading of those Minutes would be waived unless there was objection.

President Henderson announced the deaths of the following members:—

Emil A. Gramstorff, elected a Member March 17, 1937, who died September 1, 1964

Edwin A. Taylor, elected a Member February 15, 1893, who died in 1963

Burtis S. Brown, elected a Member June 21, 1911, who died in May, 1964

Clyde M. Durgin, elected a Member January 27, 1915, who died in January, 1964

Arthur L. Ford, elected a Member September 20, 1916, who died in April, 1964

Laurence E. Weeks, elected a Member December 20, 1922, who died July 11, 1964

Thomas A. Wiggin, elected a Member April 11, 1897, who died January 17, 1964

The Secretary announced the names of applicants for membership in the Society and that the following had been elected to membership November 18, 1964:—

Grade of Member.—Benjamin E. Abrams, Roger A. Burke, Bertram Berger, Richard P. Barry, David Mathoff, Bruno Brodfeld, Jun Ho Ham, David P. Sullivan

Grade of Junior Member.—Nils-Frederik Braathen, William P. Breen, Richard F. Desrosiers, Frederick Dewsnap, Jr., Kenneth L. Foster, George D. Jackson, George Johnson, Charles J. Keaney, William P. Kennedy, Paul R. Levine, Arthur E. Miller, Dudley C. Sargent, David H. Stonefield, Frank O. Waterman.

Grade of Student Member.—John E. Kavanagh, 3rd.

President Henderson stated that this was a Joint Meeting with the Structural Section and turned the meeting over to Max D. Sorota, Chairman of that Section to conduct any necessary business at this time.

Chairman Sorota introduced the speaker of the evening, Mr. Robert E. White, Vice President, Spencer, White

and Prentis, who gave an interesting illustrated talk on "Slurry Trench Walls."

A discussion period followed the talk.

Forty nine members and guests attended the dinner preceding the meeting and seventy nine members and guests attended the meeting.

The meeting adjourned at 8:45 P.M.

CHARLES O. BAIRD, JR.,
Secretary

December 16, 1964:—A regular meeting of the Boston Society of Civil Engineers was held this evening at the Society Rooms, 47 Winter Street, Boston, Mass., and was called to order by President William A. Henderson, at 7:00 P.M.

President Henderson stated that the Minutes of the previous meeting November 18, 1964, would be published in a forthcoming issue of the Journal and that the reading of those Minutes would be waived unless there was objection.

President Henderson announced the deaths of the following members:—

Porter W. Dorr, elected a member May 17, 1948, who died December 15, 1964.

Francis S. Wells, elected a member September 15, 1913, who died December 10, 1964.

The Secretary announced that the following had been elected to membership December 16, 1964:—

Grade of Member.—Robert S. Bowes, Joseph F. Canacan, Edward J. Downing, David P. McKittrick, Joseph Mulligan, Isaac L. Newell.

Grade of Junior.—Daniel S. King.

President Henderson requested the Secretary to present recommendation of the Board of Government to the Society for action. The President stated that this matter was before the Society in accordance with provisions of the By-Laws and notice of such action published in the ESNE Journal dated December 7, 1964.

The Secretary presented the following recommendation of the Board of Government to the Society for action to be taken at this meeting.

MOTION "to recommend to the Society that the Board of Government be authorized to transfer an amount not to exceed $6,000 from the Principal of the Permanent Fund to the Current Fund for Current Expenditures."

On motion duly made and seconded it was VOTED "that the Board of Government be authorized to transfer an amount not to exceed $6,000 from the Principal of the Permanent Fund to the Current Fund for current expenditures."

President Henderson stated that final action on this matter would be taken at the January meeting of the Society.

President Henderson introduced the speaker of the evening, Mr. Frank L. Heaney, Associate, Camp, Dresser & McKee, who gave a most interesting talk on "Registration and Professionalism." [The paper is published in this issue of the Journal—Ed.]

A discussion period followed the talk.

Thirty three members and guests attended the meeting.

Meeting adjourned at 8:45 P.M.

CHARLES O. BAIRD, JR.,
Secretary

SANITARY SECTION

DECEMBER 2, 1964:—A meeting of the Sanitary Section was held on this date at the Society Rooms. Speaker was Dr. Alexander Cohen, Research Psycholo-

gist, Department of Health, Education and Welfare, Cincinnati. Dr. Cohen presented an informative paper entitled "Physiological and Psychological Effects of Noise on Man," with illustrations. [The paper is published in this issue of the Journal—Ed.] The meeting was attended by 22 members and guests, and was preceded by dinner at the Smorgasbord.

ROBERT L. MESERVE,
Clerk

STRUCTURAL SECTION

NOVEMBER 18, 1964:—A joint meeting of the Structural Section and the main Society was held on this date at the Smith House in Cambridge, Mass.

The speaker, Robert E. White, Vice President, Spencer, White & Prentis, Inc., presented a paper on "Slurry Trench Walls," illustrated with 35 mm slides.

Attendance was 79 (49 at dinner).

ROBERT L. FULLER,
Clerk

OCTOBER 14, 1964:—A regular meeting of the Structural Section was held on this date, in the Project MAC Conference Room, 545 Technology Square, Cambridge, and was called to order by the Chairman, Max D. Sorota, at 7:20 P.M.

The Chairman introduced the first speaker of the evening, Professor John M. Biggs of M.I.T. who provided the background information on the evening's topic—"Computer Analysis of Structures—On-Line Demonstration" by describing STRESS, a problem-oriented language for structural analysis.

Professor Robert D. Logcher of M.I.T. demonstrated STRESS using the Project MAC Compatible Time Sharing System and teletype for communication with the computer.

After an extensive question and answer period, the meeting was adjourned at 9:00 P.M.

Attendance was 51.

ROBERT L. FULLER,
Clerk

DECEMBER 9, 1964:—A regular meeting of the Structural Section was held on this date, in the Society Rooms and was called to order by the Chairman, Max D. Sorota, at 7:00 P.M.

The Chairman introduced the speaker of the evening, Dr. Ivan Viest, of the Bethlehem Steel Company, who spoke on "Alaskan Earthquake—Effects on Structures."

Dr. Viest described the 1964 Alaskan earthquake and its effect on many types of structures. Many colored slides were used to illustrate the damage which occurred.

The following books on earthquakes have been donated to the Society by Bethlehem Steel Company:

"The Skopje, Yugoslavia Earthquake," by Glen V. Berg

"Anchorage and the Alaska Earthquake of March 27, 1964," by Glen V. Berg and James L. Stratta

After an extensive question and answer period, the meeting was adjourned at 8:15 P.M.

Attendance was 40.

ROBERT L. FULLER,
Clerk

ADDITIONS

Members

Richard P. Barry, ACD-1 Naval Amphibious Base, Coronado, California

Robert S. Bowes, 914 E. 4th St., South Boston, Mass.

Bruno Brodfeld, 50 Follen St., Apt. 316, Cambridge, Mass.

Roger A. Burke, 124 Vine Street, Reading, Mass.

Edward P. Downing, Prospect Hill Road, Harvard, Mass.
Eugene Manomaites, 16 Crockers Lane, W. Roxbury, Mass.
David Mathoff, 48 Clearway St., Boston, Mass.
David P. McKittrick, Country Club Lane, Apt. 14, Woburn, Mass.
Charles E. O'Bannon, 716 E. Hancock, Tempe, Arizona
David P. Sullivan, 287 Huron Ave., Cambridge, Mass.

Juniors

Nils-Frederik Braathen, 26 Parkvale Ave., Allston, Mass.
William P. Breen, 71 Brent St., Dorchester, Mass.
Richard F. Desrosiers, 432 North Main St., Attleboro, Mass.
Kenneth L. Foster, 131 Winstead Rd., Rochester, New York
George D. Jackson, 46 Shea St., Quincy, Mass.
Daniel S. King, 5 Oswald St., Worcester, Mass.
Charles J. Keaney, 6 Dale St., Methuen, Mass.
Paul R. Levine, 131 Washington St., Apt. 26, Brighton, Mass.
Dudley C. Sargent, Bear Hill Rd., Merrimac, Mass.
Mark W. Tenney, 240 N. Harvard St., Allston, Mass.

Students

John E. Kavanagh, 3rd., 29 Kensington St., Newtonville, Mass.

Deaths

Burtis S. Brown, May, 1964
Porter W. Dorr, Dec. 16, 1964
Clyde M. Durgin, Jan. 1964
Arthur L. Ford, 1964
Emil A. Gramstorff, Sept. 1, 1964
Edwin A. Taylor, 1963
Laurence E. Weeks, July 11, 1964
Francis S. Wells, Dec. 10, 1964
Thomas H. Wiggin, Jan. 17, 1964

RALPH W. HORNE FUND

The Ralph W. Horne Fund was recently initiated by the Directors of Fay, Spofford & Thorndike, Inc., with a gift to the Society of $3,000, not $2,000 as erroneously reported in the October, 1964, Journal.

ERRATA:
Subsurface Investigations for Highways and Structures

Note: The following errata and additions apply to the paper by H.A. Mohr which appeared in the October, 1964, issue of the *Journal* and are published at the request of Mr. Mohr.

1. The Editor's note should be changed to read as follows:

Published with the permission of the Board of Commissioners, Commonwealth of Massachusetts, Department of Public Works.

2. The following should be inserted prior to Part I:

FOREWORD

During the past several years, the writer and engineers of the Massachusetts Department of Public Works have been developing procedures and specifications for sub-surface investigations. Adequacy and economy have been the ultimate objectives.

A planning procedure has been developed for highways and their accompanying structures. Its provisions can, with minor changes, be adapted to the planning of sub-surface investigations for other structures. The technical provisions for the types of borings identified in the specifications apply to any soil investigation for foundation purposes. Application of these provisions requires the close attention of the engineer as the work progresses.

The procedures and specifications have been divided as follows:

Part I, entitled "DEFINITIONS AND PROCEDURES TO BE FOLLOWED ON SUB-SURFACE INVESTIGATIONS FOR HIGHWAYS AND STRUCTURES," is for the guidance of the engineer when planning the investigation. It is not issued to the bidders or to the boring contractor, and, therefore, is not included in the contract documents.

Appended to Part I are two examples of actual boring operations:

a) Fig. I shows the locations at which Control* and Complementary* borings were planned at one site, and Fig. 1A, the logs of results obtained from typical borings of each type. From his study of the Control boring results, the Engineer effected a major saving in the footage of Complementary borings at no detriment to an understanding of the foundation problem.

b) Fig. II shows the locations at which borings of both types were to be made at another site, and Fig. IIA, the logs of the Control borings that showed

* See Text for definitions.

the greatest difference. In this instance, because of the inconsistency in results, all Complementary borings were made to refusal with no saving in footage of borehole.

Part II, entitled "SPECIFICATIONS FOR SUB-SURFACE EXPLORATIONS" outlines the provisions governing the boring and sampling operations. These provisions become part of the bid and contract documents. They will be included in the revised Massachusetts Standard Specifications for Highways and Bridges.

This separation into "Definitions . . ." and "Specifications . . ." accounts for some unavoidable repetition.

In the development of these procedures and specifications, the writer acknowledges with thanks the cooperation and help of John F. McGovern, Bridge Engineer, Charles E. Tuck, Robert J. McDonagh, Daniel J. Guilfoyle and Harold B. Frye of the Bridge Division and Carlisle N. Levine of the Highway Division, all of the Massachusetts Department of Public Works, and of the engineers of the United States Bureau of Public Roads.

3. The following provisions were adopted after the original paper was prepared and should be inserted in the proper locations.

a. Page 335, after 2. *Structures* insert:

3. *Ground Water Observation Wellpoint*

Ground water level as reported during a soil-test-boring operation is seldom accurate. When a study of the Pilot or Control borings indicates that an excavation in granular soil must be made below ground-water-level, observation wellpoints shall be installed. Not more than one observation wellpoint shall be installed at a bridge site except with prior approval of the Bridge Engineer. Observation wellpoints shall be located outside footing areas with the bottom of the point approximately 10 feet below the proposed bottom of footing.

District personnel will measure and report water levels monthly to the Bridge Engineer with a copy to the Design Engineer. These data are to be tabulated on the Construction Drawings with the following notation:

"The water levels recorded in the table are those measured on the dates given and do not necessarily represent ground-water level at time of construction."

b. Page 352, Present item "K" change to "L." Insert new item "K" *Ground Water Observation Wellpoint.*

K. *Ground Water Observation Wellpoint*

2½″ casing shall be advanced by loosening the soil at the bottom with a chopping bit and washing the loosened soil out with water. When the bottom of the casing has reached the elevation specified for the tip of the wellpoint it shall be purged to its full depth with clean water.

The wellpoint shall have ample clearance so that it may be lowered freely in the 2½″ casing. The screen shall be 60-mesh. The riser, rigidly fastened to the wellpoint, shall be 1¼″ pipe. A pipe plug or cap shall be furnished to close the top of the riser.

After the wellpoint has been lowered to the specified elevation the annular space between the wellpoint and riser pipe and the 2½″ casing shall be filled with clean, screened, dry sand. This sand shall be retained on a 50-mesh and shall pass a 30-mesh sieve. It shall be poured in slowly to fill the annular space as the casing is pulled.

During the pulling of the casing the wellpoint shall not be raised from its original position.

At completion, the top of the riser pipe shall be closed wrench tight with a pipe plug or cap.

For the above described work complete, including the materials left in place, the Contractor will be paid at the contract unit price per foot measured from the tip of the wellpoint to the top of the riser pipe but not to more than one foot above the ground surface.

c. Page 353, Insert Payment Item.

Ground Water Observation Wellpoint . . . . Lineal Feet.

PROFESSIONAL SERVICES AND ADVERTISEMENTS

The advertising pages of the JOURNAL aim to acquaint readers with Professional and Contracting Services and Sources of Various Supplies and Materials. You would find it of advantage to be represented here.

Volume 52 JANUARY, 1965 Number 1

BOSTON SOCIETY OF CIVIL ENGINEERS

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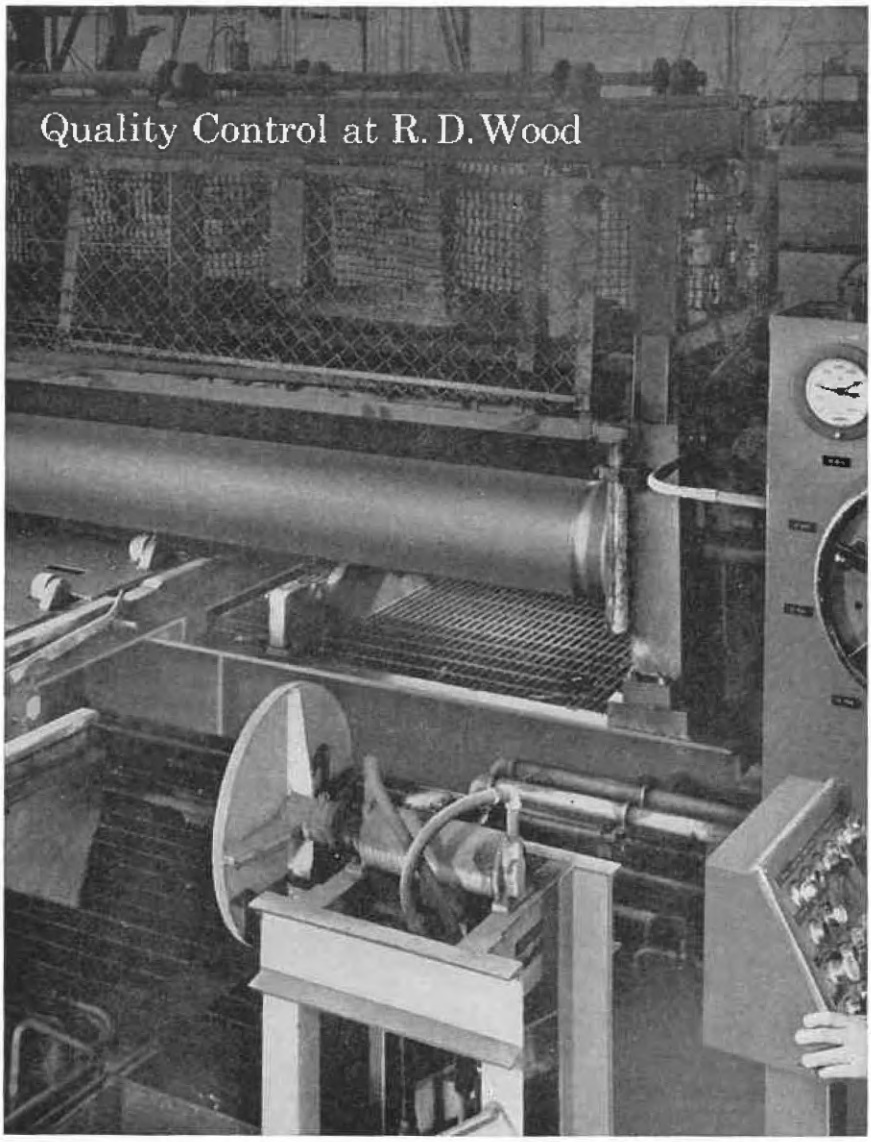
INDEX TO ADVERTISERS



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