Chapter II

Theory and Application

Oxygen Transfer

Lewis and Whitman (1924) developed the two-film concept explaining the phenomenon of the transfer of oxygen from the bubble to the liquid. Considers a thin film of gas and a film of liquid at the gas-liquid interface assuming equilibrium conditions at the interface and liquid film control in oxygen transfer, the transfer rate can be expressed:-

$$N = K_1 A(C_s - C)$$

where: K_1 = Liquid film coefficient

A = Inter facial area

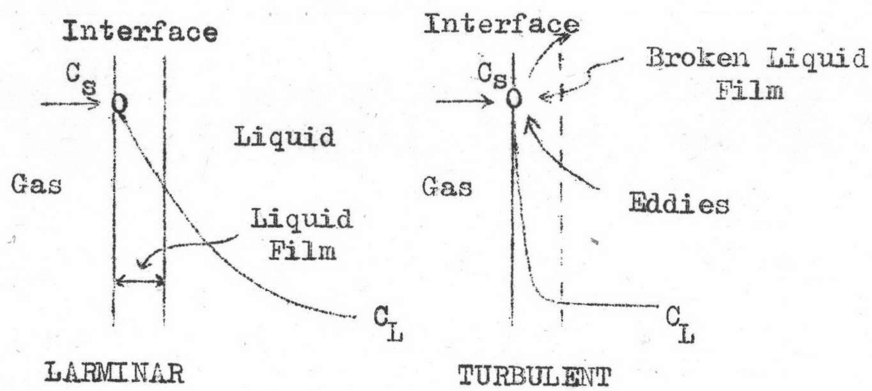
C_s = Saturated concentration of the dissolved substance

C = Its concentration at any time

The principal limitation to this concept is the assumption of steady-state transfer.

Danckwerts (1951), following the work of Higbee (1935) approached the nonsteady-state problem. He defined the liquid film coefficient as the square root of the product of the diffusion coefficient and the rate of surface renewal.

The oxygen transfer process can be considered to occur in three phases. Oxygen molecules from the gas are initially transferred to the liquid surface at a very rapid rate. During the second phase, the oxygen molecules must pass through the liquid interface by molecular diffusion. In the third phase, oxygen is mixed in the body of liquid by diffusion and convection. It is assumed that at low mixing levels or laminar flow conditions the rate of oxygen absorbtion is controlled by the rate of molecular diffusion (phase 2). At increased turbulent levels the surface film is disrupted and the renewal of the film is responsible for the transfer of oxygen to the body of liquid. As shown in the figure below, the turbulent eddies from the liquid bulk move to the interface, undergo a short interval unsteady-state molecular diffusion and are then displaced from the surface by subsequent eddies. (Eckenfelder & Ford,1968)



Ippen and Carver (1952) showed the application of the two film theory for the computation of oxygen absorption rates in water as:-

$$\frac{dc}{dt} = \frac{10^6}{W} KA\,(C_i - C_L)$$

where:

$\frac{dc}{dt}$ = Rate of change of dissolved oxygen concentration in the liquid in ppm/hr.

W = Weight of water in tank, lbs.

A = Interfacial or absorbing area, sq. ft.

K = Coefficient of oxygen absorption in lb. oxygen/(hr. x sq. ft. x unit concentration difference in ppm.

C_i = Equilibrium concentration of oxygen at the phase interface, ppm.

C_L = Concentration of oxygen in the liquid at time t, ppm.

Eckenfelder (1966) re-expressed the oxygen transfer rate based on the two-film theory in concentration units as:-

$$\frac{dc}{dt} = K_L \cdot \frac{A}{V}\,(C_s - C_l) = K_{La}\,(C_s - C_l)$$

Where: K_L = The overall diffusion coefficient based on liquid film resistance, cu. ft. of volume/sec./sq. ft. of area.

A = The total interfacial area, sq. ft.

V = The volume of liquid under consideration, cu. ft.

C_s = The concentration of O_2 in the liquid when saturated at the existing pressure and temporature, mg/l.

C_1 = The oxygen concentration at a point at a time t, mg/l.

K_{La} = Overall coefficient for oxygen transfer.

The use of this equation is based on the following assumption:-

1. The entire liquid content is completely mixed and therefore of uniform composition at any time (t).

2. K_{La} is constant and therefore is independent of time.

Eckenfelder & Ford (1968) explained the determination of mass transfer coefficient under nonsteady state aeration of a deoxygenated water that K_{La} is determined from the slope of a semilogarithmic plot of the concentration deficit (C_s-C_1) versus time of aeration. This is illustrated in Appendix B.

$$K_{La} = \frac{\log (C_s-C_1)_{t1} - \log (C_s-C_1)_{t2}}{t2 - t1} \cdot 2.3 \cdot 60 \text{ hr.}^{-1}$$

Factor Affecting K_L and K_{La}

Eckenfelder and Ford (1968) stated that several factors influence K_L and K_{La}. The most significant of these are temperature and surface active agents present in the liquid. The temperature effect can be defined by the relationship:-

$$K_t = K_{20^\circ C} \cdot \theta^{(T-20)}$$

The temperature effect of K_L, θ, has been reported to vary from 1.016 to 1.037. A study of Eckenfelder and Barnhart (1960) on bubble aeration found a value for θ of 1.028. When considering K_{La} in bubble aeration systems, the effect of temperature on the bubble size and velocity must also be included since this will effect $\frac{A}{V}$. An evaluation of available data showed θ to be 1.02 for K_{La} in bubble aeration systems. The value of θ equal to 1.024 is applied to this study.

Since the evaluated absorbing liquid in this experiment is tap water, the effect due to the surface active agents present is not taken into account.

Flow of an Incompressible Ideal Fluid

Considers the flow of an incompressible ideal fluid in a closed channel shown in the **figure** below, Bernoulli stated the general basic energy equation as:-

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2$$

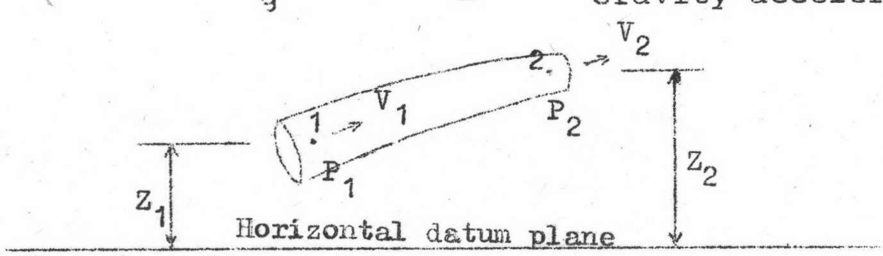
where: P_1, P_2 = The pressure at the section 1 & 2 respectively.

V_1, V_2 = The velocity at the section 1 & 2 respectively.

Z_1, Z_2 = The vertical distance of the section from the assumed horizontal datum plane.

γ = Specific weight of the fluid

g = Gravity acceleration



The terms $\frac{P}{\gamma}$, $\frac{V^2}{2g}$ and Z are considered as energy terms and generally called as pressure head, velocity head and elevation head respectively.

When the Bernoulli Equation is applied to fluid flow in pipes it altered to

$$\frac{P_1}{\gamma} + \alpha_1 \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \alpha_2 \frac{V_2^2}{2g} + Z_2 + h_{L1-2}$$

where: α is dimensionless and represent correction factor to the convectional velocity head $\frac{V^2}{2g}$

h_{L1-2} is the total loss of head from section 1 to section 2.

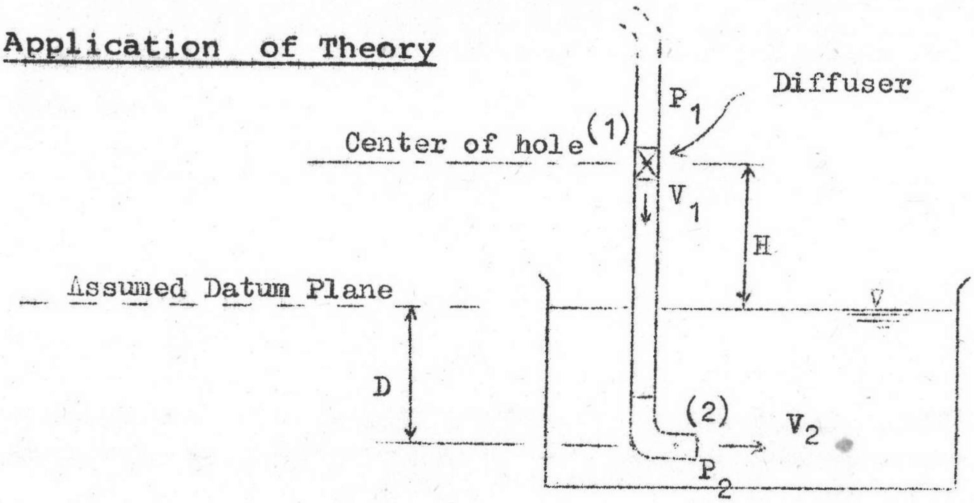
However, in most problems of pipe flow, the α terms may be omitted for several reasons as suggested by Vennard (1961).

The addition of mechanical energy (E_p) to a fluid flow by a pump or its extraction (E_T) by a turbine will alter the basic Bernoulli equation to:-

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 + E_p = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 + E_T + h_{L1-2}$$

which is the most general energy equation applied to fluid flow in pipes. Each term is generally expressed in an energy unit of ft-lb/lb.

Application of Theory



When all holes of the diffuser are closed, the engergy equation between Section (1) and Section (2) of the pipe shown in the above figure is as following:

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 + h_{L1-2}$$

here $Z_1 = H$, $\frac{P_2}{\gamma} = D$ and $Z_2 = -D$

then $$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + H = D + \frac{V_2^2}{2g} - D + h_{L1-2}$$

therefore $$\frac{P_1}{\gamma} = h_{L1-2} + \frac{V_2^2}{2g} - \frac{V_1^2}{2g} - H \quad \text{—— (1)}$$

Total head loss (h_{L1-2}) consists of friction loss, loss due to bend and the exit loss. According to this study, the friction loss is approximately equal to

$f \frac{l}{d} \frac{V_2^2}{2g}$ where f = friction factor, l = length of pipe from (1) to (2) and d_2 = pipe diameter. The loss due to bend of 0.9 $\frac{V_2^2}{2g}$ and the exit loss of $\frac{V_2^2}{2g}$ are recommended by Vennard (1961).

When Equation (1) is applied to the diffuser type I which is an ordinary perforated pipe of same diameter as that of the exit as shown in Figure 2, the velocity V_1 is equal to V_2. Hence Equation (1) is altered to:

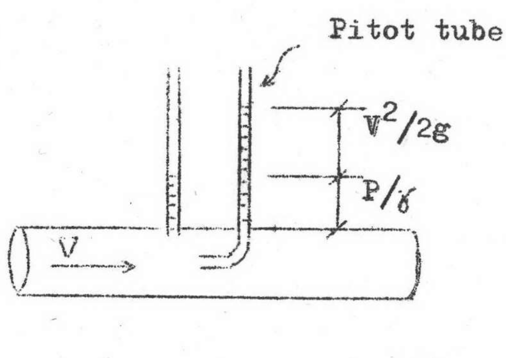
$$\frac{P_1}{\gamma} = h_{L1-2} - H \qquad (2)$$

Therefore if the elevation head, H is greater than h_{L1-2} and all holes are opened, then air will be drawn in, causing a gas-liquid mixture inside the pipes along the down stream direction of flow as long as the velocity of fluid flow is greater than the uplift velocity of air bubbles.

Again, according to the equation 1 if the velocity V_1 is increased by the decrease of cross sectional area of the section 1, the pressure head will tend to be more negative value, thus greater air suction is obtained, so this leads to the use of the Venturi diffuser (Type III Figure 4). The Orifice type can be applied also, but offers more flow losses than the Venturi type. In standard designs of Venturi meter, it consists of converging, parallel and diverging sections, the parallel section is short, ¼ - ½ pipe diameter; the converging section has a taper angle of 15-20° whilst the diverging section is longer, having a taper angle 5-7½°. (Barna, 1969).

Nominal values of the loss of head due to Venturi tube used is recommended to be equal to 0.04 $\frac{V_1^2}{2g}$ in the case of turbulent flow (Vennard, 1961). Though it is not taken in addition to the loss of head h_{L1-2} in the equation 1, it still has an effect on the decrease in rate of discharge. Therefore if the Venturi diffuser is used. the equation 1 must be applied to this aeration system.

Now condider the diffuser type II which can be classified as the Pitot type shown in Figure 3, generally the Pitot tube is made for the determination of flow velocity when it is faced upstream as shown below.



The difference in elevation of fluid in the two tubes represents the velocity head, $V^2/2g$ (Vennard, 1961)

In opposite case, if it is faced downstream, the elevation of fluid in the Pitot tube will drop down causing lower level than that of the straight tube, thus gains the velocity head as an additional suction unit if the loss due to the Pitot tube is negligible. The equation concerned will be:-

$$\frac{P_1}{\gamma} = h_{L1-2} - H - \frac{V_2^{\,2}}{2g} \qquad \text{(3)}$$

and it is applied to the diffuser type II in this study.

Referred to the three equation (1, 2 and 3) mentioned above it is noticed that increase in elevation head or elevation of the diffuser will result in the increase of suction pressure or air supply.