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Heat Loss in Aeration Tanks

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by

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ABSTRACT OF THE THESIS

Heat Loss in Aeration Tanks

by

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Recent developments in wastewater treatment aeration systems have focused on aeration efficiency and minimizing energy cost. Other operating characteristics have often been ignored. The impact of aeration system alternatives on aeration tank temperature can be substantial and design engineers should include these potential effects in the evaluation of alternatives.

To predict aeration tank temperature and its influence on system design, a survey of all previous literature was made and a new computer model on Lotus 123 spreadsheet was developed. The model was based in part upon previous works, but significant improvements were made in the areas of heat loss from aeration and atmospheric radiation. The new model was verified with 17 sets of literature data, and predicts temperature with an RMS error of 1.24 for these 17 sets.

The model was used to determine the relative influence of design, operation and climatic variables. Heat loss from aeration was found to exert a great influence on surface aeration, accounting for 50% of the total heat loss. Heat loss from sub-surface system was 50% of the total heat losses from surface aeration. Wind speed and air humidity are important parameters in determining aeration tank temperature.

1 INTRODUCTION

Biological systems have gained major importance in wastewater treatment in view of stringent regulatory standards for discharging organic wastes. Thus, there is a need to incorporate all possible operating parameters in design stages and then to closely monitor these parameters for design and operation.

The efficiency of a biological system is largely determined by the activity of biomass. Factors affecting biomass activity, such as substrate concentration, oxygen supply rate and oxygen saturation, are significantly influenced by temperature. Moreover, substrate removal rate is related to aeration basin detention time which is the design parameter for calculating basin volume. Biomass activity tends to increase with increasing temperature from 20°C to 40°C which is the normal ambient temperature at which the biological system operates. It leads to the necessity of accurately predicting equilibrium temperature of wastewater in aerated biological system. Inaccurate estimation of temperature can result in failure to comply with effluent discharge standards or cause gross overdesign of the system.

Equilibrium water temperature of the biological system can be predicted by making a heat balance around the system. The overall heat balance employs identification of major heat transfer components. Temperature of the aeration basin is controlled by influent temperature and meteorological conditions. Determination of heat balance and equilibrium water temperature may not be of direct help in controlling wastewater temperature but it could be used to adjust those parameters which compensate for an incremental change in temperature. Therefore, a better understanding of heat transfer factors could help in the control of upset plant conditions or in the design of new systems.

Several researchers proposed heat balance models to determine equilibrium temperature in aerated basins, anaerobic digesters, streams and lakes. From time to time, these models are modified to incorporate all possible parameters in an attempt to make it general purpose. There is still need to further improve these models.

The objective of this study is to present a general purpose, comprehensive temperature prediction model which is applicable to a wide range of meteorological and operational conditions, and is least dependent on empirical constants. A literature review is presented with a comparison of the existing models and a new model is made as a part of this thesis. The equilibrium temperature predicted by various models is compared with literature data to determine the relative accuracy of the models. A sensitivity analysis is conducted to get a better understanding of the impact of design and operating conditions on equilibrium temperature.

This model is designed with the objective of applying it to different plant sites to determine the effect of design and operating parameters at different geographical locations with varying meteorological conditions. The model finds application in temperature prediction for surface mechanical, diffused and pure oxygen aeration systems.

2 LITERATURE REVIEW

One of the major purposes of this study is to examine the literature pertaining to heat transfer from a body of water. Not enough information on heat transfer in wastewater aeration system is available. However, many investigations have been conducted on heat balances in lakes, streams and ponds. Extensive work has been recorded in the area of evaporation of water. Most investigators have derived heat transfer equations for uniform water temperature under steady state conditions.

Many researchers have attempted to evaluate heat transfer in ponds and aerated lagoons. Lima (1936) was one of the first to propose a method, empirically based, to determine cooling effects of ponds. Thorne (1951) suggested a procedure based on a modified energy budget relationships.

The simplest general formula for heat transfer in aerated lagoons was proposed by Eckenfelder (1966). He defined a single heat transfer coefficient to account for the effects of wind, humidity and aeration equipment. Commenting on this coefficient, Mancini and Barnhart (1968) indicated that it varies with geographical location and day of the year. Barnhart (1968) derived a heat balance relationships for aerated lagoons by introducing an empirical constant in the evaporation term. He proposed to modify tank surface area to account for aeration equipment, and presented the graphical relationship between apparent surface area and power input to account for the effects of mixing intensity.

Rohwer (1931), Meyer (1942) and Anderson (1954) collected voluminous data from their experiments and derived equations for evaporation from a water surface. Their results are widely accepted and utilized by fellow researchers such as Thorne (1951), Langhaar (1953), Raphael (1962), Velz and Gannon (1960).

In an attempt to determine Cooling Pond area, Langhaar (1953) developed nomograms for predicting equilibrium temperature of ponds for various conditions of air temperature, relative humidity, wind speed and solar radiation. However, Langhaar cautioned that his method was not verified experimentally and might not accurately predict experimental results. Langhaar's approach was modified by Velz and Gannon (1960) in their attempt to evaluate waste heat problems. After careful data analysis of meteorological and hydrological variables reported by the U.S. Weather Bureau and U.S. Geological Survey, they suggested that the greatest probability of occurrence of the variable could be obtained by developing normal distributions from the recorded data.

Raphael (1962) correlated empirical relationships for conduction, convection, radiation and evaporation while neglecting heat loss due to chemical and biological processes, heat loss through tank walls and the transformation of kinetic energy into thermal energy. Much of his methodology was derived from the Lake Hefner studies by Anderson (1954). Thackston (1972) followed Raphael's procedure to prepare contour maps of equilibrium temperature, heat exchange coefficients and the amount of cooling effected by given pond sizes at various loacations under different meteorological conditions. His results show that the latitude (which controls solar radiation), wind speed and air temperature have a strong influence on equilibrium temperature.

Ford, Shih and Sebesta (1972) presented and verified a design approach for predicting a temperature profile across activated sludge basins using mechanical aerators. Their model is based on a trial and error technique and includes the heat loss attributable to the aerator spray cloud which is calculated from the differential enthalpy of the air flowing through it. Novotny and Krenkel (1973) gave the most comprehensive overview of evaporation and heat balances in aerated basins by accounting for aeration heat loss. Their results agree quiet well with the model developed by Ford, Shih and Sebesta (1972).

Argaman and Adams (1977) extended Novotny and Krenkel's model by including the terms for heat gained from mechanical energy input and biological reactions, and heat lost through the tank walls. Novotny and Krenkel neglected these heat losses in view of their small magnitude. They attempted to present the complete temperature prediction model for both surface and diffused aeration systems. However, their model relies on empirical data for determining aerator spray vertical cross sectional area.

Friedman and Doesburg (1981) tested the model of Argaman and Adams by applying data from eight different industrial biotreating systems. They concluded that the temperature predicted by their model is accurate to 1 to 3°C. They conducted a sensitivity analysis to arrive at a general correlation of the heat exchange characteristics of the eight treatment systems.

The remaining portion of this Chapter discusses each term in an overall heat balance. Each of the previously cited researcher's approach is discussed. The best technique from each are selected for further use and development.

2.1 SOLAR RADIATION

Energy from Sun in the form of short wave radiation is an important factor for the heat balance of open surfaces. Short wave radiation contains most of its energy in the 0.1 to 4.0 μm wavelength range. The radiation from Sun is absorbed, scattered and diffused by moisture, gases and dust particles before it reaches the surface of earth. Net absorbed solar radiation on a given surface may be expressed as:

$$Net\ solar\ radiation = Incident\ radiation - Reflected\ radiation \tag{1}$$

Incident energy received on the earth's surface depends on the angle of incidence of Sun's rays which is strongly influenced by the latitude of the surface and time of the year. Reflected radiation is a function of solar altitude, type and amount of cloud, turbidity of the atmosphere and wind velocity. Several tables and graphs of average daily incident radiation as a function of latitude and time of the year for clear sky conditions are available. These are primarily based on the analysis of weather bureau records for a number of stations throughout the country.

An accurate determination of solar radiation is not possible because it depends on the complex and unpredictable meteorological parameters which are further influenced by local environmental conditions such as cloud cover and air pollution. These parameters vary throughout the day and year. Attempts have been made by many researchers to develop accurate expressions for solar radiation.

Langhaar (1953) presented a table which shows maximum expected solar radiation for *daily average* and *Noon* at Northern Hemisphere latitudes for different months of the year. His values are conservative and express incident solar radiation for limited latitudes. The factors affecting solar radiation are analyzed at length in the Lake Hefner studies by Anderson (1954). He concluded that the reflectivity of a natural water surface is primarily a function of sun altitude. Wind velocity, turbidity of the atmosphere and the type of clouds have insignificant impact on reflectivity.

Raphael (1962) furnished data of total absorbed solar radiation on a horizontal surface for a given time as a function of solar altitude, taken from the tables prepared by Moon (1940). He used Anderson's result to develop the following empirical equation for the net rate of solar radiation on an exposed basin surface. It is a function of cloud cover, basin surface area and absorbed solar radiation rate under clear sky conditions.

$$H_{sr} = H_{sr,o}\,(1-0.0071C_c^2) \qquad (2)$$

where :

H_{sr} = Net absorbed solar radiation, Btu/hr/ft²;

$H_{sr,o}$ = Absorbed solar radiation under clear sky conditions, Btu/hr/ft²;

C_c = Cloud cover, tenths.

Thackston and Parker (1971) developed the following expressions for predicting net absorbed solar radiation by using the non-linear least square method as earlier outlined by Thackston (1967). These expressions yield instantaneous radiation which is of little use for practical purposes.

$$H_{sr,o} = 2.044\alpha + 0.1296\alpha^2 - 0.0019\alpha^3 + 0.0000076\alpha^4 \qquad (3)$$

$$\sin\alpha = \sin\phi\sin\delta + \cos\phi\cos\delta\cos h$$

$$\delta = -23.28\cos[(2\pi x/365)+0.164]$$

where :

α = Solar altitude, degrees;

ϕ = Latitude of the site, degrees;

δ = declination of the Sun, degrees;

h = Hour angle of the Sun (+ve: before noon, -ve: after noon);

x = Day of the year.

Thackston and Parker (1972) modified Raphael's approach to obtain average daily absorbed solar radiation rate by using the technique of numerical integration. They developed equations for each latitude by using non-linear least square procedures. These equations are summarized in *Appendix I.*

Argaman and Adams (1977) applied Raphael's (1962) equation in their model. Friedman and Doesburg (1981) while testing Argaman and Adams's model made indirect use of Raphael's equation.

Equation (2) is used for this study because it is based on extensive data analysis. The value of net absorbed solar radiation ($H_{sr,o}$) is obtained from Thackston and Parker data.

2.2 ATMOSPHERIC RADIATION

Water surfaces receive long wave radiation from the atmosphere, reflecting a portion of it and absorbing the rest. This type of radiation occurs in the 4 to 100 μm wavelength range. The atmospheric radiation depends upon the temperature of water, temperature of air, water emissivity and air constituents. The net radiation is the difference between long wave radiation absorbed by water and the radiation leaving a body of water. It may be expressed as:

$$\textit{Net atmospheric radiation} = \textit{Absorbed radiation - Back radiation from water} \qquad (4)$$

Langhaar (1953), and Velz and Gannon (1960) applied the following simple equation to determine net radiation at ordinary temperatures based on emissivity of water equal to 1.0 BTU/hr/ft^2/°F. They assumed that the objects around water are at ordinary temperature.

$$H_{ar} = 1.0(T_w - T_a) \tag{5}$$

where :

H_{ar} = Net absorbed atmospheric radiation, Btu/hr/ft²;

T_w = Temperature of water, °F;

T_a = Temperature of ambient air, °F.

Anderson (1954) observed that the atmospheric radiation is a function of many variables. Some of them such as moisture, CO_2, O_3 are not normally measured and therefore difficult to include into a model. He proposed that such uncertain factors may be combined together to form an empirical constant called an atmospheric radiation factor, β. Anderson concluded that the effect of vapor pressure decreases as the cloud cover increases. He also noticed that the radiation is an inverse function of the height of clouds for a given cloud amount. As shown below, Anderson developed correlations utilizing Stefan-Boltzman radiation law.

$$\begin{aligned} H_{ar} &= \beta\,\sigma\,T_a^4 \\ \beta &= a + b e_a; \\ a &= 0.740 + 0.025C_\epsilon^{-0.0584h}; \\ b &= 0.0049 - 0.00054C_\epsilon^{-0.06h} \end{aligned} \tag{6}$$

where :

β = Atmospheric radiation factor;

σ = Stefan-Boltzman Constant (= 0.173×10^{-8} Btu/hr/ft^2/°R^4);

e_a = Vapor pressure, millibars;

C = Cloud cover, tenths;

h = Cloud height, feet;

ϵ = the Naperian base.

Raphael (1962) combined Burt's (1958) method and Anderson's graph to construct plots of the atmospheric radiation factor, β as a function of cloud cover and vapor pressure of air as shown in Figure 1. His equation given below is widely used by fellow investigators. This equation is based on Dunkle's (1949) observation that the reflectivity of a water surface for normal air temperatures is 0.03 . The emissivity of water is assumed to be 0.97 . The equation becomes:

$$H_{ar} = 0.97\, A_s \sigma [T_w^4 - \beta T_a^4] \qquad (7)$$

where :

A_s = Surface area of tank, ft^2.

Similar work is reported by Thackston and Parker (1972). They correlated linear equations to calculate atmospheric radiation factor, β for each line of cloud cover from the graph produced by Raphael. The equation is given below. Table 1 shows values of the constants a and b for the various cloud covers.

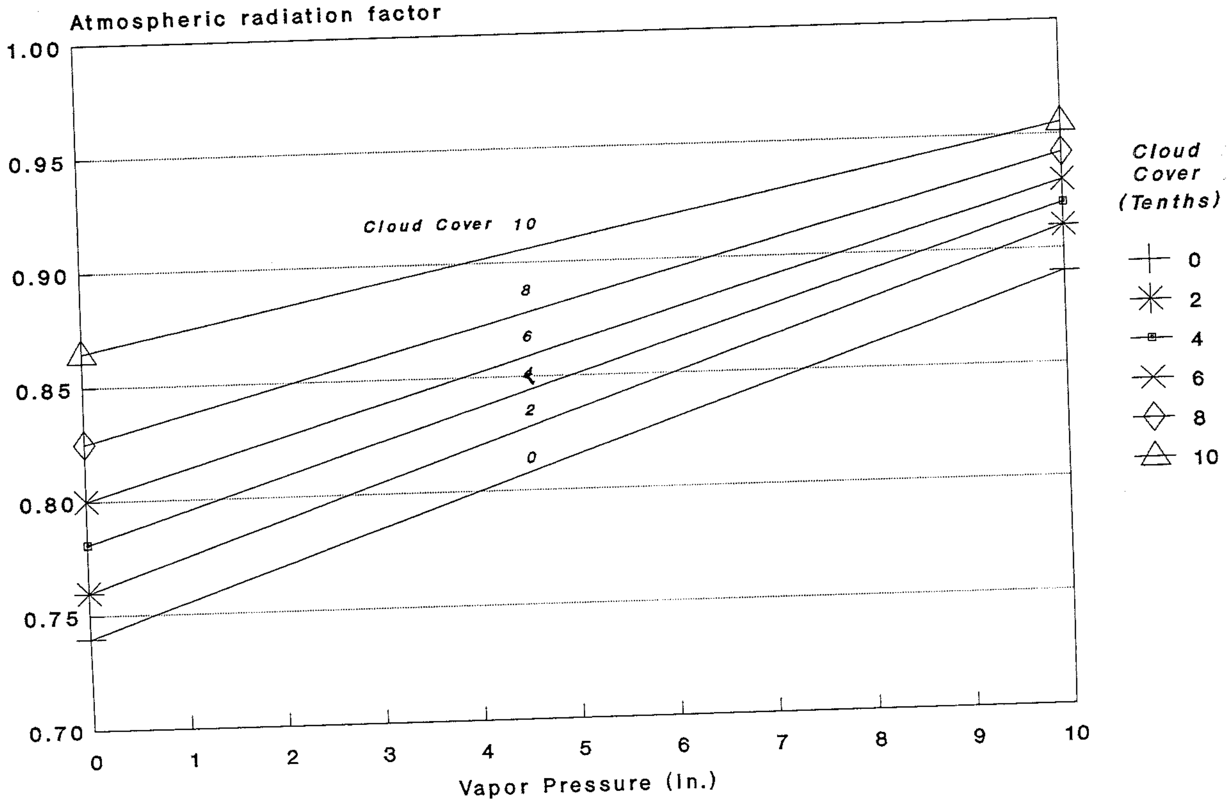

Figure 1 Relationship between Atmospheric Radiation Factor and Vapor Pressure for Various Cloud Covers (After Raphael (1962))

$$\beta = a + be_a \tag{8}$$

Thackston and Parker suggested that the vapor pressure of ambient air is equal to saturated vapor pressure at the wet bulb temperature, T_{wb}. The expressions for T_{wb} and υ_s are as follows:

$$T_{wb} = (0.655 + 0.36r_h)\,T_a \tag{9}$$

$$\upsilon_s = \mathrm{Exp}\left[17.62 - \frac{9,501}{(T_{wb} + 460)}\right] \tag{10}$$

where :

T_{wb} = Wet bulb temperature of air, °F;

υ_s = Vapor pressure of water, in. Hg;

r_h = Relative humidity, % .

Novotny and Krenkel (1973) linearized Raphael's equation because it was based on the fourth power of temperature. They argued that the linearization is possible because the difference between air and water temperature is small compared to the absolute values of the temperatures. The relationships developed by Novotny and Krenkel are given below. Argaman and Adams (1977) and Friedman and Doesburg (1981) modified these correlations for prediction of equilibrium temperature of aeration tank.

$$H_{ar} = A_a + B_a(T_a - T_w) + C_a T_a \tag{11}$$

$$A_a = 695.04(\beta - 0.874);$$

$$B_a = 11.42;$$

$$C_a = 10.19(\beta - 1.123)$$

where:

T_a = Temperature of air, °C;

T_w = Temperature of water, °C.

The linearized form of Raphael's equation shown in Equation (11) is used in this study for the calculation of atmospheric radiation. It utilizes classical Stefan-Boltzman radiation law.

Table 1

Values of the Constants *a* and *b* for Various Cloud Covers (after Thackston and Parker (1972))

Cloud cover (tenths)	a	b
0	0.740	0.150
1	0.750	0.150
2	0.760	0.150
3	0.770	0.143
4	0.783	0.138
5	0.793	0.137
6	0.800	0.135
7	0.810	0.130
8	0.825	0.120
9	0.845	0.105
10	0.866	0.090

2.3 SURFACE CONVECTION

Heat transfer due to convection occurs because of the difference in temperature of air and water, and is magnified by incoming wind. Previous investigators have assumed that wind velocity at the water surface is approximately half of tree top wind velocity. The expression for this heat exchange is given by:

$$H_c = C \rho_a c_{pa} W (T_w - T_a) \tag{12}$$

where :

H_c = Heat transfer by surface conduction;

C = Empirical constant;

ρ_a = Density of water;

c_{pa} = Specific heat of air;

W = Wind speed.

Langhaar (1953) derived equation (13) on the basis of previous data of convection coefficient and its effect on wind. He assumed convection coefficient to be 0.8 for a flat surface in still air and 0.12 for every *additional mph* of wind velocity. Velz and Gannon (1960) and Barnhart (1968) modified Langhaar's equation by modifying convection coefficient to 0.16 so as to include higher effect of wind velocity. Their correlation is shown below by equation 14.

$$H_c = (0.8+0.12W)\,(T_w-T_a) \tag{13}$$

$$H_c = (0.8+0.16W)\,(T_w-T_a) \tag{14}$$

where :

H_c = Heat lost by surface convection, Btu/hr/ft^2;

W = Wind velocity (tree top), mph

T_w = Temperature of water, °F;

T_a = Temperature of ambient air, °F.

Raphael (1962) suggested the following equation to calculate convective heat losses. His equation is based on Bowen's (1962) ratio of heat transfer by conduction to evaporation. Thackston and Parker (1972) derived a similar equation by applying the same parameters.

$$H_c = 0.00407\,W\,P(T_a-T_w) \tag{15}$$

where :

W = Wind velocity, knots;

P = Atmospheric pressure, in. Hg.

Novotny and Krenkel (1973) concluded that the values of heat transfer coefficient and vapor coefficient would be same on the basis of assumption that the Prandtl numbers for both evaporation and conduction in air are similar. They applied Harbeck's (1962) formula for calculation of the transfer coefficients.

$$H_c = h \rho_a c_{pa} (T_w - T_a) \tag{16}$$

$$h = 39200 A_s^{-0.05} (W - W_w)$$

where :

h = Vapor transfer coefficient, cm/day;

W = Wind velocity, m/sec;

W_w = Water surface velocity in wind direction, m/sec;

T_a = Temperature of ambient air, °C;

T_w = Temperature of water, °C.

Argaman and Adams (1977) essentially used Novotny and Krenkel's equation for the computation of convective heat losses. Their equation may be expressed as:

$$H_c = 11.8 \times 10^4 \; W A_s^{0.95} (T_w - T_a) \tag{17}$$

where :

H_c = Heat loss by convection, cal/day;

A_s = Surface area, m^2;

W = Wind velocity, m/sec;

T_a = Temperature of ambient air, °C;

T_w = Temperature of water, °C.

Equation (16) is the general form of equation based on the assumptions described earlier. It is used in this model to calculate convective heat losses.

2.4 EVAPORATIVE HEAT LOSSES

Evaporation of water from surfaces such as rivers, streams and ponds has been the subject of major concern in the fields of hydrology, geology and irrigation where water conservation is of prime importance. Evaporation of water into air occurs when the vapor pressure of air is less than its saturated vapor pressure at the water temperature. This process of mass transfer consumes considerable amount of energy and results in a decrease in water temperature.

The calculation of heat loss due to evaporation involves determination of the amount of water evaporated and the latent heat of vaporization. The quantity of water evaporated is a function of vapor pressure, air temperature, water temperature, wind velocity and relative humidity. Latent heat of vaporization may be obtained from Steam tables or by using a regression expression. The general type of evaporation formula may be expressed as:

$$H_{ev} = \rho_v \, (a + bw) \, (\upsilon_s - \upsilon_a) \, L \tag{18}$$

where :

H_{ev} = Evaporative heat loss;

a , b = Empirical constants;

W = Wind speed;

ρ_v = Density of evaporated water;

υ_a = Air vapor pressure at T_a ;

υ_s = Saturated water vapor pressure at T_W ;

L = Latent heat of vaporization.

The correlation developed by Meyer (1942) has gained wide acceptance in engineering practice and is based on routine weather bureau observation. It holds good for the temperature of natural water. The constant C in the equation ranges from 10 to 15, and depends upon depth and exposure of water, and frequency of meteorological measurements. For large deep lakes, C tends to the lower side of the range.

$$E = C\,(1+0.1W)\,(\upsilon_w-\upsilon_a) \qquad (19)$$

where,

C = Empirical constant;

E = Evaporation per month, in. ;

W = Wind velocity at tree top, mph;

υ_w = Vapor pressure of water at water temperature, in. Hg;

υ_a = Monthly average morning and evening vapor pressures, in. Hg.

Thackston and Parker (1972) used an empirical constant which is based on size, shape and exposure of the water body. Langhaar (1953) applied extension to equation (19) in order to calculate heat loss from cooling ponds.

$$H_{ev} = 75\,(1+0.1W)\,(\upsilon_w-\upsilon_a) \qquad (20)$$

where :

H_{ev} = Heat loss from evaporation, Btu/hr/ft^2

Rohwer (1931) conducted a series of experiments to predict evaporation rate under different laboratory conditions. He observed that rate of evaporation increases as the pressure of air decreases. He observed that the relationship between altitude and evaporation is difficult to determine since both the temperature and vapor pressure of air are functions of pressure. Experimentally, he found that evaporation rate increased with the altitude. The empirical equation showing rate of evaporation for reservoirs is given by:

$$E = 0.771\ (1.465 - 0.0186B)\ (0.44 + 0.118W)\ (\upsilon_s - \upsilon_d) \qquad (21)$$

where,

E = Evaporation per day, in.;

B = Mean Barometer reading, in. Hg at 32°C;

W = Mean wind velocity at water surface, mph;

υ_s = Mean vapor pressure of saturated vapor at water temp., in. Hg;

υ_d = Mean vapor pressure of saturated air at dew point temp., in. Hg.

Raphael made a detailed comparison of Anderson's (1954) evaporation data and suggested that the equation (22) given below best conformed with the results of experiment conducted by Anderson. He derived equation (23) from equation (22) by assuming latent heat of vaporization equal to 1060 Btu/lb. Parker and Krenkel (1970) also applied

Anderson's result in their study as follows:

$$E = 0.00177\ W\ (\upsilon_w - \upsilon_a) \tag{22}$$

$$H_{ev} = 12\ W\ (\upsilon_w - \upsilon_a) \tag{23}$$

where,

E = Evaporation per day, in.;

W = Mean wind velocity at water surface, mph;

υ_w = Mean vapor pressure of saturated vapor at water temp., millibar;

υ_a = Mean vapor pressure of saturated air at dew point, millibar.

In the study of aerated lagoons, Barnhart (1968) used an empirical approach shown in equation (24). Velz and Gannon (1960) used the same equation in their study.

$$H_{ev} = 0.00722\ LC(1-0.1W)\left(\upsilon_w - \upsilon_a \frac{r_h}{100}\right) \tag{24}$$

where,

L = Latent heat of vaporization;

C = Lagoon constant;

W = Mean wind velocity at water surface, mph;

υ_w = Vapor pressure at water temperature, in. Hg;

υ_a = Vapor pressure in atmosphere, in. Hg;

r_h = Relative humidity, %.

Andrews and Kambhu (1971) developed following equation for prediction of evaporation heat loss in thermophilic aerobic digestion.

$$
\begin{aligned}
H_{ev} &= 1020\, Q_a \rho_a a_h \\
a_h &= \left(\frac{\rho_{wv}}{\rho_a}\right)\left(\frac{p_{wv}}{p - p_{wv}}\right) \\
\log p_{wv} &= -2238/T + 8.896
\end{aligned}
\tag{25}
$$

where:

Q_a = Air flow rate;

a_h = Absolute humidity of air, lb water vapor /lb dry air;

P = Total pressure of air, mm.Hg;

p_{wv} = Partial pressure of water vapor, mm.Hg;

T = Absolute temperature, °K;

ρ_a = Density of air;

ρ_{wv} = Density of water vapor.

Novotny and Krenkel (1973) used Harbeck's (1962) expression of transfer coefficient and simplified their equation as given below, by replacing vapor pressure term with temperature.

$$
\begin{aligned}
H_{ev} &= A_e + K_e(T_w - T_a) \\
A_e &= 2.92 \times 10^{-3} \exp\left[0.0604 T_a \left(1 - \frac{r_h}{100}\right) h_v\right] \\
K_e &= 1.75 \times 10^{-4} \exp[0.0604\, T_a h_v] \\
h_v &= 39200\, A_s^{-0.05}\, (W - W_w)
\end{aligned}
\tag{26}
$$

where :

H_{ev} = Heat loss from evaporation, cal/cm² day;

W = Wind velocity, m/sec;

W_w = Water surface velocity in wind direction, m/sec.

r_h = Relative humidity, % ;

Argaman and Adams (1977) and Friedman and Doesburg (1981) used the following equation which is based on the correlation developed by Novotny and Krenkel.

$$H_{ev} = \left[1.145\times 10^6 \left(1-\frac{r_h}{100}\right) + 6.86\times 10^4 (T_w - T_a)\right] \times \exp[0.0604 T_a W A^{0.96}] \tag{27}$$

where :

H_{ev} = Heat loss from evaporation, cal/day;

W = Wind velocity, m/sec;

T_a = Temperature of air, °C;

T_w = Temperature of water, °C.

r_h = Relative humidity, % ;

The heat loss from evaporation in this study is based on the equation (27) which was originally developed by Novotny and Krenkel. This equation is directly correlated with temperature.

2.5 AERATION HEAT LOSS

Aerobic biological wastewater systems are usually supplied with additional amount of air in order to maintain sufficient dissolved oxygen. In the course of contact between air and water, air gets enriched with water vapor because of evaporation of water from the tank. The amount of water evaporated depends on air flow rate, tank temperature, ambient air temperature and relative humidity. At higher air temperatures, the water vapor pressure increases allowing the air to contain more water.

Evaporation of water due to aeration causes removal of sensible and latent heat from the wastewater. The driving force for sensible and latent heat loss is the difference in temperature of air and water. These heat losses may be expressed as follows:

$$H_a = Q_a \rho_a c_{pa} (T_w - T_a) + Q_a \rho_a L_a (\upsilon_s - \upsilon_a) \qquad (28)$$

where :

υ_s = Absolute humidity at tank temperature, lb water vapor/lb air;

υ_a = Absolute humidity at air temperature, lb water vapor/lb air.

Heat loss from aeration depends on the type of aeration method used. Surface and diffused aeration systems are most commonly used in a wastewater treatment plant. The surface aeration is usually accompanied by higher heat losses because of greater exposure of water to the ambient air.

Ford, Shih and Sebesta (1972) predicted temperature of activated sludge basins using surface aerators. Their approach was to calculate heat losses from the unaerated water surface and from the aerator spray. They used Langhaar's (1953) expression to calculate heat loss from water surface. The spray heat loss was based on the differential enthalpy of the air flow through the spray cloud.

Novotny and Krenkel (1973) correlated vapor pressure with temperature and used Harbeck's (1962) equation to calculate the vapor transfer coefficient. They developed following equation using the same approach for calculating heat loss from surface evaporation and convection. Their equation was used by Argaman and Adams (1977) and Friedman and Doesburg (1980) to predict temperature in an aerated biological system.

$$\begin{aligned} H_a &= Q_a \rho_a C_{pa} (T_w - T_a) + 1.76 \times 10^{-4} Q_a e^{0.0604 T_a} (T_w - T_a) \\ &\quad + 2.92 \times 10^{-3} e^{0.0604 T_a} \left(1 - \frac{r_h}{100}\right) \end{aligned} \tag{29}$$

where :

H_a = Aeration heat loss, cal/cm²/day;

T_w = Temperature of water, °C;

T_a = Temperature of air, °C

2.6 HEAT FROM POWER INPUT

The purpose of using aerators and compressors in an aeration tank is two fold: to supply oxygen and to provide proper mixing. A portion of the power supplied to run this equipment is transformed into mechanical energy; a significant fraction of this energy is converted to heat energy, which results in an increase in water temperature. Hence power input to aeration equipment can have an important role in the overall heat balance of the aeration tank.

Heat input to wastewater through a power source can be estimated from the horsepower supplied to the aeration device. Sparks (1984) applied following correlation to calculate mixing heat input for predicting temperature in aerobic digestion. For coarse bubble diffused aeration systems, he suggested that all the mixing energy supplied to aeration equipment may not transfer to the wastewater.

$$H_p = 2546 V h \tag{30}$$

where :

H_p = Power input, Btu/hr;

V = Reactor volume, ft^3;

h = Horsepower input per unit volume, HP/ft^3.

Ford, Shih and Sebesta (1972) ignored aeration power input in their calculations of heat loss in four aeration basins, though they identified its contribution in general equation. Novotny and Krenkel (1973) also neglected power input in their heat balances in aerated basins.

Argaman and Adams (1977) suggested that with the diffused air systems, some of the mechanical energy would be lost in friction in the compressor. They assumed that this heat going to the surrounding air would be recovered from increased temperature of the air entering the compressor. They used a simple expression shown below which is based on conversion of units from horsepower to calories. Friedman and Doesburg (1981) used the same approach to evaluate temperature effects in aerated biotreaters.

$$H_p = 15.2 \times 10^6 \, P \tag{31}$$

where :

H_p = Heat input to aeration tank, cal/day;

P = Power input to aeration equipment, HP

2.7 BIOLOGICAL REACTION

Biological reactions are exothermic and therefore need to be incorporated in calculation of heat losses from aeration tank. Kambhu and

Andrews (1969) applied following equation in simulating aerobic thermophilic digestor for unsteady state conditions.

$$H_{rx} = \frac{\rho_w \, C_{pw} \, \Delta T}{k \; t \; (BVS)} \tag{32}$$

where :

H_{rx} = Heat of biological reaction, Btu/lb;

ρ_w = Density of water, lb/ft^3;

C_{pw} = Specific heat of water, Btu/lb/°F;

k = Reaction rate constant, day^{-1};

t = Time, days;

ΔT = Temperature difference between air and water, °F

BVS = Concentration of biodegradable volatile solids in effluent, lb/ft^3.

Spoehr and Milner (1949) applied the following empirical equation in calculating heat produced from combustion of organic materials.

$$H_{rx} = 127R + 400 \tag{33}$$

$$R = 100 \, \frac{[2.66(\%C) + 7.94(\%H) - (\%O)]}{398.9}$$

where :

H_{rx} = Heat of combustion of organic mixture, Cal/gm;

R = Degree of reduction of organics, % (ash free).

Andrews and Kambhu (1971) used following equation in computing heat loss from digestion of organic solid wastes.

$$H_{rx} = 6,100R_o + 11,000 f_N k V (BVS) \qquad (34)$$

$$= 6,100(oe) k V (BVS) + 11,000 f_N k V (BVS)$$

where :

H_{rx} = Heat of reaction, Btu/hr;

oe = Oxygen equivalent of BVS for NH_3 as the product, lb O_2/lb BVS;

k = Rate constant, hr^{-1}

f_N = Fraction of nitrogen in sludge, lb N/lb BVS;

V = Volume of the digestor, ft^3.

Argaman and Adams (1977) suggested that heat produced from biodegradation of organic wastes is very sensitive to the net cellular yield. They predicted that net energy release varies from 370 to 3300 cal /g COD depending upon the production of volatile suspended solids. They concluded that a low sludge age system would release less energy as shown by following equation.

$$H_{rx} = 1.8 \times 10^6 \ S_r \qquad (35)$$

where :

H_{rx} = Heat of biological reaction, cal/day;

S_r = Organic removal rate, Kg COD removed/day.

Equation (35) developed by Argaman and Adams, is included in this study because it is applicable to any aerobic biological system.

2.8 TANK WALL HEAT LOSS

Heat losses from aeration basin tank walls depend upon the material of construction, the wall area and its thickness. The material of construction most widely used is concrete followed by steel. The classical heat transfer equation is:

$$H_{tw} = U A_w (T_w - T_a) \qquad (36)$$

$$U = \frac{1}{\frac{1}{h_i} + \sum_{1}^{n} \frac{x_i}{k_i} + \frac{1}{h_o}}$$

where :

H_{tw} = Heat loss through tank walls;

U = Overall heat transfer coefficient;

A_w = Area of material normal to the direction of heat flow;

h_i = heat transfer coefficient inside the tank;

h_o = heat transfer coefficient outside the tank;

x_i = Thickness of the material;

k_i = Thermal conductivity of the material;

n = Number of materials.

The overall heat transfer coefficient, U, depends on tank material and its thickness, conductivity of the material and temperature difference between tank contents and surroundings. The bottom of the basin generally derives its insulation from the supporting earth while the sidewalls may have earth backfill for insulation. Unless the ground-water level is known, the earth at sides of the tank is taken as being dry, while the earth at the bottom is taken as being wet.

For an accurate determination of heat transfer, separate values of U are found for the side and the bottom. For underground tanks, the calculation of U for the wall surrounded by earth is based on the compound wall of tank material and earth.

The *Sewage Treatment Design Manual* (1959) suggests the following values for individual transfer coefficients of ordinary materials.

h_i = 1.6 Btu/hr/ft^2/°F (in still air);

h_o = 6.0 Btu/hr/ft^2/°F (with wind velocity of 15 mph);

= 1.0 Btu/hr/ft^2/°F for tank bottom.

Normally, heat transfer through tank walls is only a small fraction of the total heat loss from aeration tank. Thus, in general practice, a single heat transfer coefficient for sides and bottom of the tank can be selected. Table 2 A gives the values for thermal conductivity of the materials. Common values for overall heat transfer coefficients are presented in Table 2 B.

Table 2 A

Thermal Conductivity of commonly used Materials

Material	U (Btu/hr/ft^2)
Brick Masonry	5.0
Cement mortar	12.0
Cinder concrete	5.4
Concrete (typical)	12.0
Concrete blocks	0.8 - 1.0
Dry earth	8.0
Wet earth	16.0

Table 2 B

Overall Heat Transfer Coefficient for Various Portions of Aeration Tank

Sections of Tank	U (Btu/hr/ft^2)
12" concrete walls with air space	0.35
12" concrete walls, wet earth covered	0.25
12" concrete walls, dry earth covered	0.18
Floor	0.12

Source: Sewage Treatment Plant Design (1959)

2.9 SUMMARY

In this chapter each component of heat loss was identified and the work of previous investigators described. Most of the researchers have not included all the elements of heat loss in their overall heat balance equation. Argaman and Adam's (1977) model is most complete since they combined all the heat loss components in their overall equation.

Eckenfelder presented a simple and generally accepted equation (37) for calculating temperature of aerated lagoons. This equation is not based on the calculation of each heat loss components. It incorporates a factor f which accounts for the effects of wind, humidity and aeration equipment. Thus it also depends on the plant location.

$$T_i - T_w = \frac{(T_w - T_a)\ f\ A_s}{Q} \tag{37}$$

where:

T_i = Influent water temperature, °F;

T_w = Lagoon temperature, °F;

T_a = Air temperature, °F;

A_s = Surface area of lagoon, ft^2;

Q = Water flow rate, mgd;

f = Empirical factor ($= 12 \times 10^{-6}$ for central U.S.A.)

3 MODEL DEVELOPMENT & VERIFICATION

This Chapter on model development and verification describes a procedure for prediction of equilibrium temperature of aeration tank. A final form of equation for temperature prediction was established by running a heat balance around the aeration tank. In order to establish the validity of this model, it is compared with previous models and literature data for actual measured temperatures.

Development of the proposed model employs three basic strategies: First, various components of heat transfer in the aeration tank were identified to get an overview of the overall heat balance. Secondly, with the aid of extensive literature survey, these components were compiled from various sources to put in the form of equations. These equations were critically analysed and modified, if necessary, before incorporating them into the model. Some of the correlations are experimentally derived by previous investigators. A general form of such correlations is not possible because of their inherent empirical nature. Finally, all the heat transfer terms are combined and rearranged to obtain final equation for estimating equilibrium temperature of an aeration tank.

3.1 MODEL DEVELOPMENT

The model presented herein is applicable to a completely mixed tank under steady state conditions. The basic assumption of completely mixed implies uniform tank temperature and instantaneous dispersion of particles entering the tank. Such behavior is favored by high detention time, sufficient tank depth and the use of a mixing device. The other extreme hydraulic condition, plug flow, signifies no longitudinal mixing at all. Plug flow is difficult to approach since longitudinal mixing can not be entirely eliminated.

Equilibrium temperature is attained in the aeration tank by assuming steady state conditions. An aeration tank reaches its equilibrium temperature when it is exposed to constant meteorological conditions for a sufficiently long period. Since the temperature of a large body of water does not vary much during the day because of high heat content of water, a close approach to equilibrium temperature may be assumed. A body of water not at equilibrium temperature will tend to approach equilibrium asymptotically.

The general form of heat balance in the aeration tank is shown in Figure 2 and may be written as:

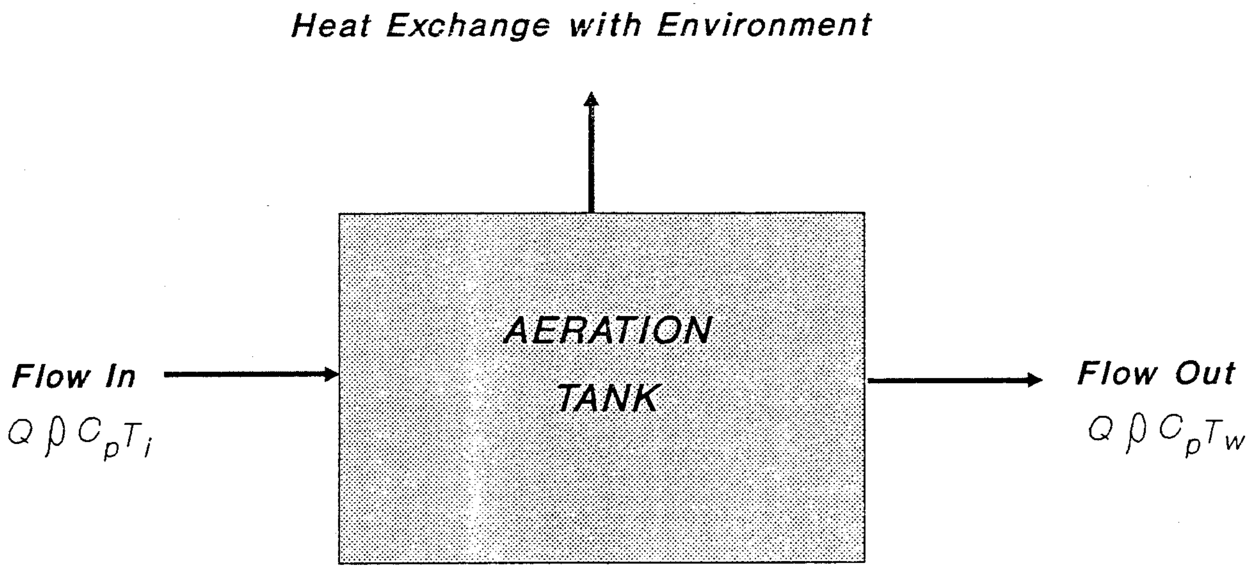

Figure 2 Components of Overall Heat Balance in Aeration Tank

$$\textit{Change of heat content} = \textit{Heat In (flow) - Heat Out (flow)} + \textit{Heat exchange with the environment} \tag{38}$$

Equations (39) and (40) given below represent heat balances for unsteady and steady state conditions respectively.

$$\rho_w c_{pw} V \frac{dT_w}{dt} = \Delta H + \rho_w c_{pw} Q_w (T_w - T_i) \tag{39}$$

$$\Delta H = \rho_w c_{pw} Q_w (T_i - T_w) \tag{40}$$

where :

ΔH = Net heat exchange with the environment, cal/day;

ρ_w = Density of water, Kg/m^3;

c_{pw} = Specific heat of water, cal/kg/°C;

Q_w = Flow rate of wastewater in aeration tank, m^3/day;

V = Volume of aeration tank, m^3;

T_i = Temperature of influent, °C;

T_w = Temperature of wastewater in aeration tank, °C;

t = Time period, days.

Net heat loss, ΔH represents heat exchange with the atmosphere by means of convection, radiation and evaporation. Various components involved in the heat exchange with environment are shown in Figure 3 and are represented as:

Mechanisms of Heat Exchange with Environment

Heat Loss (if $T_W > T_a$)	Heat Gain
Atmospheric Radiation	Solar Radiation
Surface Convection	Power Input
Surface Evaporation	Biological Reaction
Aeration	
Tank Wall	

These elements of heat balance are expressed below in the form of equation. The positive and negative terms represent heat loss and heat gain respectively.

$$\Delta H = H_{ar} - H_{sr} + H_{ev} + H_{c} + H_{a} - H_{p} - H_{rx} + H_{tw} \qquad (41)$$

where :

H_{ar} = Heat loss from atmospheric radiation, cal/day;

H_{sr} = Heat gained from atmospheric radiation, cal/day;

H_{ev} = Heat loss from surface evaporation, cal/day;

H_{c} = Heat loss from surface convection, cal/day;

H_{a} = Heat loss due to aeration, cal/day;

H_{p} = Heat gained from power input, cal/day;

H_{rx} = Heat gained from biological reaction, cal/day;

H_{tw} = Heat loss through tank walls, cal/day

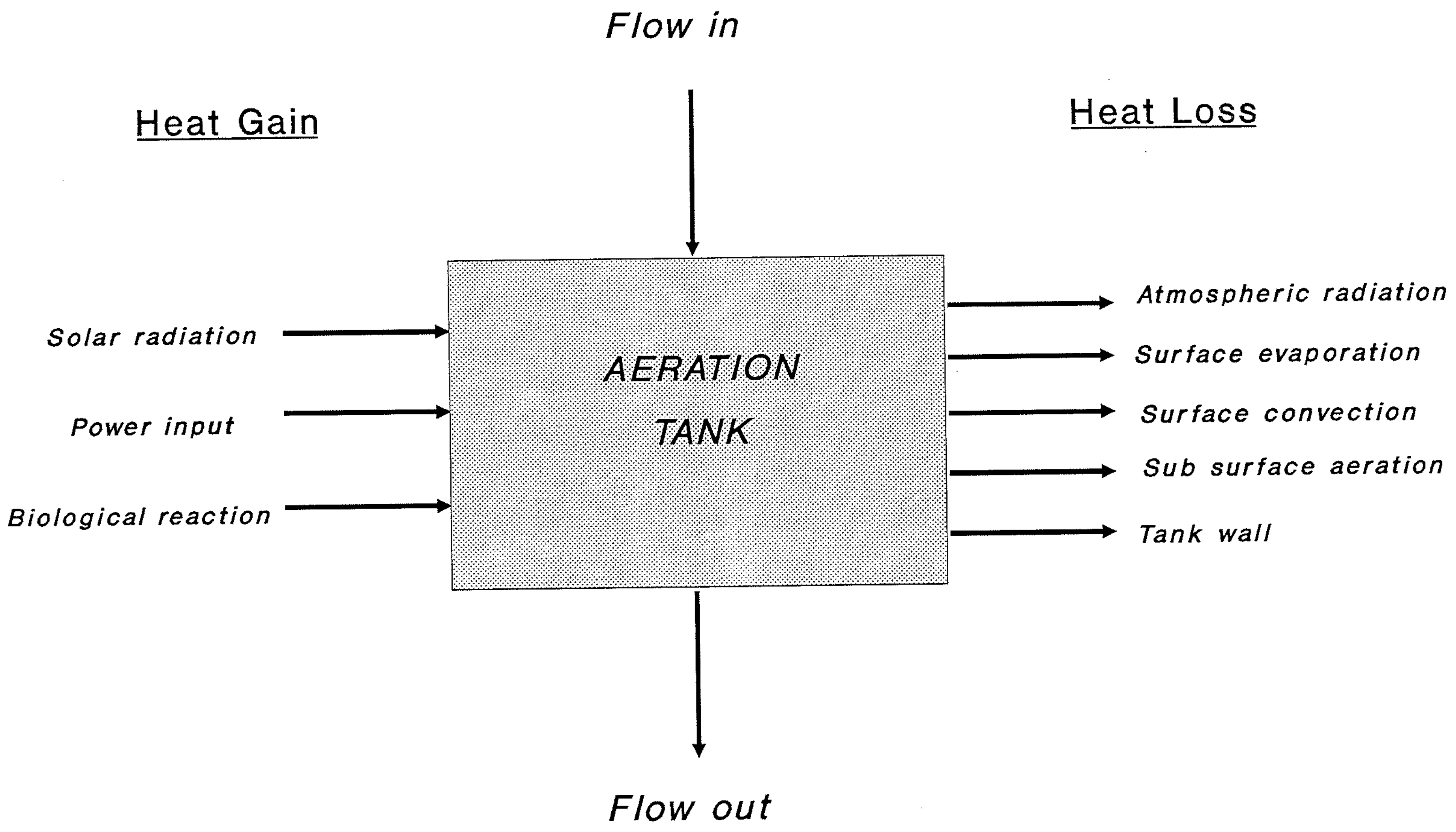

Figure 3 Components of Heat Exchange with Environment

3.1.1 SOLAR RADIATION

Net heat gained from solar radiation is a function of meteorological factors, latitude of the site and period of the year. The model incorporates Raphael's (1962) equation as given below because it is based on a critical analysis of experimental data of other researchers.

$$H_{sr} = H_{sr,o} (1 - 0.0071 C_c^2) A_s \tag{42}$$

where :

$H_{sr,o}$ = Average daily absorbed solar radiation for clear sky conditions;

C_c = Cloud cover, tenths;

A_s = Surface area of tank, m^2.

Absorbed solar radiation for clear skies depends on the latitude of site and day of the year, and is determined from the equations presented by Thackston and Parker (1972). They provided a separate equation for each latitude from 26° to 46°. These equations are regressed in this model to arrive at a single equation as shown below. Table 3 shows comparison of regression equation with those developed by Thackston and Parker.

$$H_{sr,o} = a - b\sin\left[\frac{2\pi d}{366} + c\right] \tag{43}$$

$$a = 95.1892 - 0.3591k - 8.4537\times10^{-3}k^2;$$

$$b = -6.2484 + 1.6645k - 1.1648\times10^{-2}k^2;$$

$$c = 1.4451 + 1.434\times10^{-2}k - 1.745\times10^{-4}k^2$$

where :

d = Day of the year;

k = Latitude of the site, degrees.

3.1.2 ATMOSPHERIC RADIATION

The heat loss from atmospheric radiation is based on the Stefan Boltzman's fourth power radiation law and is expressed as:

$$H_{ar} = H_{ar,w} - H_{ar,a} \tag{44}$$

where :

$H_{ar,w}$ = Back radiation from water, cal/day;

$H_{ar,a}$ = Net incoming atmospheric radiation, cal/day.

Back radiation from water, $H_{ar,w}$ is computed as shown below by applying a correction for emissivity of water because surface of water does not behave as a perfect black body.

Table 3

Comparison of average daily absorbed solar radiation, $H_{sr,o}$
(Thackston's Equations vs Model Regression)

Latitude (deg)	Day of the year											
	1			100			200			300		
	Thack-ston	Model	% diff.	Thack-ston	Model	% diff.	Thack-ston	Model	% diff.	Thack-ston	Model	% diff.
26	51.177	51.295	0.23%	87.498	88.058	0.64%	107.056	106.746	0.29%	64.990	64.480	0.78%
27	49.518	49.475	0.09%	87.963	87.683	0.32%	106.803	106.832	0.03%	62.802	62.981	0.29%
28	47.728	47.661	0.14%	87.347	87.284	0.07%	106.929	106.879	0.05%	61.537	61.479	0.09%
29	46.016	45.854	0.35%	87.567	86.860	0.81%	106.393	106.887	0.46%	59.269	59.974	1.19%
30	44.002	44.053	0.12%	86.552	86.410	0.16%	106.524	106.859	0.31%	58.072	58.467	0.68%
31	42.244	42.258	0.03%	85.116	85.932	0.96%	107.276	106.793	0.45%	57.882	56.960	1.59%
32	40.456	40.469	0.03%	85.867	85.425	0.52%	106.626	106.692	0.06%	55.073	55.454	0.69%
33	38.691	38.685	0.02%	84.979	84.887	0.11%	106.327	106.555	0.21%	53.725	53.950	0.42%
34	36.879	36.906	0.07%	84.121	84.319	0.24%	106.359	106.384	0.02%	52.547	52.448	0.19%
35	35.188	35.132	0.16%	83.860	83.719	0.17%	106.044	106.177	0.13%	50.807	50.949	0.28%
36	33.331	33.362	0.09%	82.653	83.086	0.52%	106.278	105.938	0.32%	49.981	49.455	1.05%
37	31.592	31.596	0.01%	82.234	82.418	0.22%	105.865	105.665	0.19%	48.239	47.965	0.57%
38	29.865	29.835	0.10%	81.844	81.715	0.16%	105.324	105.359	0.03%	46.395	46.482	0.19%
39	28.082	28.077	0.02%	81.025	80.977	0.06%	105.083	105.020	0.06%	45.021	45.005	0.04%
40	26.313	26.323	0.04%	80.286	80.202	0.10%	104.595	104.650	0.05%	43.424	43.535	0.26%
41	24.502	24.572	0.28%	79.233	79.390	0.20%	104.326	104.248	0.07%	42.172	42.073	0.24%
42	22.816	22.824	0.04%	78.564	78.539	0.03%	103.822	103.815	0.01%	40.598	40.619	0.05%
43	21.075	21.080	0.02%	77.643	77.650	0.01%	103.349	103.351	0.00%	39.173	39.175	0.00%
44	19.301	19.338	0.19%	76.751	76.721	0.04%	102.858	102.857	0.00%	37.681	37.740	0.16%
45	17.671	17.600	0.40%	75.864	75.752	0.15%	102.230	102.332	0.10%	36.233	36.316	0.23%
46	15.892	15.864	0.17%	74.710	74.742	0.04%	101.757	101.776	0.02%	34.938	34.902	0.10%

$$H_{ar,w} = \epsilon \sigma \, (T_w + 273)^4 A_s \tag{45}$$

where :

ϵ = Emissivity of water surface;

σ = Stefan Boltzman Constant (= 1.17 x 10^{-3} cal/m^2/day/°K^4);

A_s = Surface area of aeration tank, m^2.

As shown in equation (46) below, net absorbed longwave radiation takes into account atmospheric radiation factor, β and reflectivity of water, λ. Dunkle (1949) determined that reflectivity of water is normally equal to 0.03 .

$$H_{ar,a} = (1 - \lambda) \beta \sigma \, (T_a + 273)^4 A_s \tag{46}$$

where :

β = Atmospheric radiation factor (0.75 - 0.95 for most atmospheric conditions);

λ = Reflectivity of water.

Equation (47) shows the overall equation for heat loss from atmospheric radiation.

$$H_{ar} = [\epsilon \sigma (T_w + 273)^4 \; A_s \,] - [(1 - \lambda) \beta \sigma (T_a + 273)^4 A_s] \tag{47}$$

$$H_{ar} = [695 \times 10^4 (1 - \beta) + 10.18 \times 10^4 (T_w - T_a) + 10.18 \times 10^4 (1 - \beta) \; T_a] \; A_s \tag{48}$$

Argaman and Adams (1977) linearized above equation for simplicity in calculation. While developing this model, their equation (48) was compared with equation (47) for several hypothetical values and the result is shown in Figure 4.

3.1.3 SURFACE CONVECTION

The driving force for heat loss by surface convection is the temperature difference between air and the water surface. The rate of convective heat loss is influenced by vapor transfer coefficient which is a function of wind velocity. Novotny and Krenkel (1973) suggested that the transfer coefficients for evaporation and convection are same because the Prandtl numbers in air for both processes are similar. Following equation is obtained by using their approach.

$$H_c = \rho_a c_{pa} h_v s A_s (T_w - T_a) \tag{49}$$

$$h_v = 392\ A_s^{-0.05} W$$

where :

h_v = Vapor transfer coefficient, m/sec;

ρ_a = Density of air, kg/m^3;

c_{pa} = Specific heat of air, cal/kg/°C;

s = Conversion factor, sec/day.

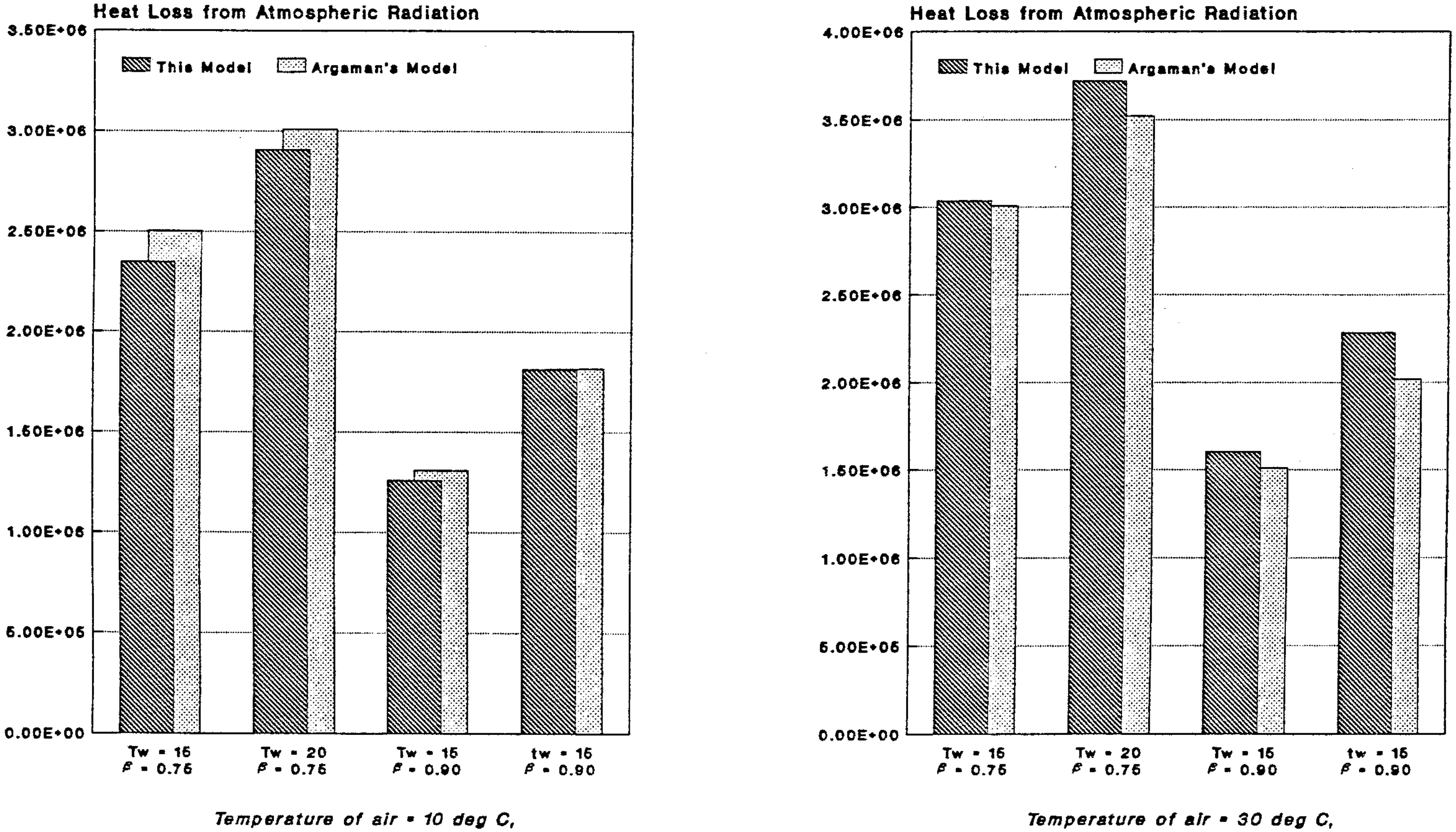

Figure 4 Comparison of Heat Loss from Radiation
Classical Equation (this Model) vs Argaman's Model

3.1.4 EVAPORATIVE HEAT LOSSES

The heat transfer by surface evaporation depends upon wind velocity and vapor pressure of water at tank and air temperatures. Novotny and Krenkel's (1973) method is used here. His expression given below assumes that heat transfer and vapor transfer coefficients are similar.

$$H_{ev} = \left[1.145\times10^{6}\left(1-\frac{r_h}{100}\right) + 6.86\times10^{4}(T_w - T_a)\right] \times e^{0.0604T_a}\, W\, A_s^{0.95} \qquad (50)$$

where :

r_h = Relative humidity, %;

W = Wind velocity at tree top, m/sec.

3.1.5 AERATION HEAT LOSS

Heat loss due to aeration consists of two components: sensible and evaporative heat losses. Heat loss from aeration depends to a large extent on the type of aeration equipment employed. The general form of heat loss equation used in this model is expressed as:

$$H_a = H_{as} + H_{al} \tag{51}$$

$$H_{as} = h_v \rho_a c_{pa} s A_s (T_w - T_a) \quad (for\ \ surface\ \ aeration);$$

$$H_{as} = Q_a \rho_a c_{pa} s (T_w - T_a) \quad (for\ \ diffused\ \ aeration);$$

$$H_{al} = Q_a\ L\ s \left[\frac{\upsilon_w (r_h + h_f (100 - r_h))}{100} - \upsilon_a \cdot \frac{r_h}{100} \right];$$

$$= \frac{M_w}{R} \frac{Q_a \left[\frac{\upsilon_w (r_h + h_f (100 - r_h))}{100} - \upsilon_a \frac{r_h}{100} \right] L s}{(T_a + 273)};$$

$$h_v = 392\ F^{-0.05}\ W;$$

$$Q_a = N\ F\ W \quad (for\ \ surface\ \ aeration)$$

where :

H_{as} = Sensible heat loss due to aeration, cal/day;

H_{al} = Evaporation heat loss due to aeration, cal/day;

Q_a = Air flow rate, m^3/sec;

N = Number of aerators;

F = Aerator spray area, m^2;

h_f = Exit air humidity factor, (0-1, =1 for diffused aeration);

L = Latent heat of vaporization of water, cal/kg;

υ_w = Vapor pressure of water at tank temperature, mm. Hg;

υ_a = Vapor pressure of water at ambient air temperature, mm. Hg;

M_w = Molecular Weight of water;

R = Universal Gas Constant, (= 62.361 mm.Hg - liters / gmole $°K^4$)

Latent heat of vaporization of water used in the above expression varies with temperature. The following expression is used to calculate latent heat at different temperatures.

$$L = \exp[a_0 + a_2 \ln T + a_3 T + a_4 T^2] \tag{52}$$

$a_0 = 7.0492;$

$a_2 = -1.7539987 \times 10^{-2};$

$a_3 = -1.46508 \times 10^{-4};$

$a_4 = -1.024186 \times 10^{-6}$

where :

L = Latent heat of vaporization of water, Btu/lb;

T = Temperature of wastewater in aeration tank, °F

Similarly a regression equation shown below is used to determine vapor pressure of water.

$$\upsilon = 4.5101 + 0.39240122T + 0.0014456T^2 + 6.6553 \times 10^{-4} T^3 - 4.59143 \times 10^{-6} T^4 \tag{53}$$

In diffused aeration systems, the temperature of the ambient air entering the blowers is increased by compression. It is assumed that the compressed air expands in the aeration tank and reaches ambient temperature. Since air travels through the surface of water, the model assumes that outgoing air attains the temperature of wastewater and is fully saturated with respect to water vapor.

Surface aerators spray water across the liquid surface and bring the water in contact with a large volume of air resulting in high evaporation rates. Since the spray droplets are exposed to ambient air for a short period, the chances of air becoming saturated are less. Therefore, in equation (51), a heat transfer coefficient is introduced for determining convective heat losses and exit air humidity factor for the evaporative losses. The exit air humidity factor depends upon wind speed and ambient air humidity. These calculations are based on the empirical data of vertical cross sectional area of aerator spray.

3.1.6 HEAT LOSS FROM POWER INPUT

Surface aerators are partially submerged in the aeration tank and are therefore in direct contact with wastewater. Hence, all the power supplied to the impellers in such aerators is available in the form of heat energy to wastewater.

As opposed to surface aerators, heat input in diffused aeration systems depends upon the efficiency of compressor. Since air is adiabatically compressed, the *Pressure - Temperature* relationship shows an increase in air temperature at higher pressure. It was determined that for a given air flow rate and outlet pressure, only a fraction of the horsepower supplied to compressor would be converted in the form of sensible heat pick up by the compressed air. This fraction of heat transferred to wastewater is equivalent to (1 - efficiency /100) of the compressor. Therefore for a compressor of 70% efficiency, 30% of the brake horsepower is translated into heat energy.

Equations (55) and (56) are used to calculate heat gained from power input in surface and diffused aeration systems respectively.

$$H_p = c_{hp} P \tag{55}$$

$$H_p = c_{hp} P \left(1 - \frac{\eta}{100} \right) \tag{56}$$

where :

c_{hp} = Constant for conversion from HP to Cal, cal/HP;

P = Break horsepower of the aeration blower, HP;

η = Efficiency of compressor, %

3.1.7 BIOLOGICAL REACTION

Biological Reactions contribute in heat addition to aeration tank because such reactions are exothermic in nature. Heat released from a biological process depends upon composition of wastewater, mass of organics removed and cellular yield.

This model calculates heat gained from a biological reaction by using following correlation which was suggested by Argaman and Adams (1977).

$$H_{rx} = h_s \Delta S \qquad (57)$$

where:

h_s = Heat released from biodegradation of organics, cal/g COD;

ΔS = Organic removal rate, Kg COD removed/day

Argaman and Adams estimated that the heat released from biodegradation of organics is 1,800 cal/g COD on the basis of assumptions given below.

- Net cellular yield is 0.25 g VSS/g COD removed.
- Free energy of oxidation of organics is -3,300 cal/g COD oxidized.
- Free energy of conversion of substrate to pyruvate is -100 cal/g COD.
- Free energy of conversion of pyruvate to cellular matter is +930 cal/g COD.
- COD of cellular matter is 1.42 g COD /g VSS.

3.1.8 TANK WALL HEAT LOSS

This model incorporates an option of computing heat loss for each section of the aeration tank. The sides and bottom of the aeration tank would have different heat transfer coefficient because of their exposure to varied surroundings, typically air or earth. On identification of such sections, appropriate values of heat transfer coefficient and the wall area are provided for each of them.

Heat loss through tank floor depends upon the temperature of ground and the type of surrounding earth, wet or dry. The overall heat transfer equation is expressed as follows:

$$H_{tw} = \sum U A_w (T_w - T_{ae}) \tag{58}$$

where :

U = Heat transfer coefficient, cal/day/m^2/°C;

A_w = Tank wall area, m^2;

T_{ae} = Temperature of air/earth, °C

3.2 MODEL VERIFICATION

Each heat transfer term discussed previously is combined with continuity terms to produce the overall heat balance equation (59).

$$\begin{aligned}
& Q_w \rho_w c_{pw} T_w + \epsilon \sigma T_{kw}^4 A_s + \rho_a c_{pa} h_v A_s s T_w + 6.86 \times 10^4 e^{0.0604 T_a} \\
& \times W A_s^{0.95} T_w + \frac{M_w}{R} \cdot \frac{Q_a}{T_{ka}} L s \upsilon_w \left[\frac{r_h + h_f (100 - r_h)}{100} \right] + U A_w T_w \\
& \quad = Q_w \rho_w c_{pw} T_i + (1 - \lambda) \beta \sigma T_{ka}^4 A_s + H_{sr,o} (1 - 0.0071 C_c^2) A_s \\
& \quad + \rho_a c_{pa} h_v A_s s T_a - 1.145 \times 10^6 e^{0.0604 T_a} W A_s^{0.95} \left(1 - \frac{r_h}{100} \right) \\
& \quad + 6.86 \times 10^4 e^{0.0604 T_a} W A_s^{0.95} T_a - H_{as} \\
& \quad + \frac{M_w}{R} \cdot \frac{Q_a}{T_{ka}} L s \upsilon_a \frac{r_h}{100} + c_{hp} P + h_s \Delta S + U A_w T_{ae}
\end{aligned} \tag{59}$$

Terms υ and L are functions of temperature. $H_{sr,o}$ is a function of latitude. They can be correlated from regression equations (54), (53) and (44) respectively, or, obtained from standard or site-specific information. A spreadsheet (Lotus 123) computer program is designed to solve the complex overall equation using iteration for the implicit terms. *Appendix II* shows the input variables used in the computer program.

In order to establish validity of this model, predicted temperature is compared with measured temperature and the temperature estimated by previous models. The model is tested on plant data collected by Argaman and Adams (1977) and Ford, Shih and Sebesta (1972). These data cover wide range of input variables for both surface and diffused aeration systems.

Since the data presented by Ford, Shih and Sebesta do not include substrate removal rate, it was calculated using their estimated conversion efficiency assuming that the inlet COD is 270 mg/L. Tables 4A & 4B shows the input variables for 17 data sets that were entered into the computer program. The model selected for comparison are: Argaman and Adam's model, Eckenfelder's equation and Langhaar's nomogram.

The output is shown in Table 5 and is represented graphically in Figure 5. The temperatures predicted by this model agree well with the measured temperature. The RMS error for this model is 1.24 which is less than the other models, as shown in the last line of Table 5. The results yielded by Argaman and Adam's model closely follow temperature predicted by the model. Eckenfelder's equation follows pattern of temperature changes but differ in magnitude on an average by +3.8 °C. Langhaar's nomogram relies mainly on meteorological factors and shows large deviations from the measured temperature.

Table 4 A

Input Process Data Selected for Verification

Data Set	Flow rate (m^3/d)	Influent temp. (°C)	Tank surface area (m^2)	Vertical wall area (m^2)	Average no. of aerators	Aerator spray area (m^2)	Air flow rate (m^3/s)	Power input (HP)	Substrate removal rate	Reference*
1	22,730	25.8	11,150	13,380	11.5	11.1	-	1,150	27,700	1
2	22,350	25.1	11,150	13,380	9.1	11.1	-	910	18,600	1
3	23,110	27.8	11,150	13,380	9.3	11.1	-	930	19,500	1
4	23,600	28.5	11,150	13,380	9.6	11.1	-	960	20,100	1
5	25,110	27.5	11,150	13,380	10.7	11.1	-	1,070	21,100	1
6	25,260	28.0	11,150	13,380	10.8	11.1	-	1,080	21,100	1
7	26,630	31.0	11,150	13,380	11.8	11.1	-	1,180	31,800	1
8	27,050	31.3	11,150	13,380	11.8	11.1	-	1,180	30,400	1
9	28,450	29.0	11,150	13,380	10.5	11.1	-	1,050	38,100	1
10	25,340	29.5	11,150	13,380	10.0	11.1	-	1,000	36,400	1
11	22,610	27.5	11,150	13,380	9.8	11.1	-	980	27,900	1
12	19,730	21.7	11,150	13,380	5.1	11.1	-	510	12,700	1
13	49,250	37.8	174,630	181,000	-	-	56.6	4,900	11,340	1
14	7,100	36.7	4,200	5,500	5	5.88	-	100	1,825	2
15	72,300	-5.8	9,960	11,150	10	7.43	-	1,000	6,200	2
16	72,300	-5.8	9,960	11,150	10	7.43	-	1,000	9,350	2
17	72,300	-5.8	9,960	11,150	10	7.43	-	1,000	13,750	2

* 1: Argaman and Adams (1977)

2: Ford, Shih and Sebesta (1972)

Table 4 B

Input Meteorological Data Selected for Verification

Data Set	Solar radiation (Clear sky) Kcal/m²/day	Wind speed (m/sec)	Relative humidity (%)	Cloud cover
1	2,280	3.9	82	8.1
2	3,120	5.0	73	6.1
3	4,360	4.8	74	7.2
4	5,530	4.8	66	6.0
5	6,440	3.9	74	6.5
6	7,090	4.2	73	5.6
7	7,090	2.9	74	4.8
8	6,510	3.1	77	6.2
9	5,270	3.2	83	6.8
10	3,770	2.9	69	2.9
11	2,600	4.0	73	6.1
12	2,280	4.3	79	7.2
13	2,925	5.2	70	6.4
14	4,460	3.6	53	4.0
15	1,670	4.0	71	3.0
16	1,670	4.0	71	3.0
17	1,670	4.0	71	3.0

Table 5

Comparison of Tank Temperature for Various Models

Data Set	Temperature (°C)				
	Measured	Model	Argaman	Eckenfelder	Langhaar
1	17.0	17.2	18.1	22.2	7.8
2	14.6	15.5	16.2	21.2	8.9
3	19.8	19.4	20.3	24.9	14.5
4	20.9	20.4	21.4	25.9	17.8
5	23.0	23.6	24.3	26.4	24.4
6	24.1	24.1	24.8	27.0	25.3
7	28.4	28.3	29.0	30.1	30.8
8	29.5	27.8	28.5	30.2	30.0
9	27.0	25.2	26.1	27.5	23.9
10	24.3	23.9	25.0	27.0	18.9
11	20.5	19.1	20.1	24.1	11.7
12	17.5	15.2	15.6	18.3	8.3
13	18.3	16.1	17.5	20.5	12.8
14	27.8	29.9	31.4	35.6	29.7
15	4.4*	4.7	4.6	5.3	-
16	10.0*	10.7	10.8	11.7	-
17	15.6*	16.6	17.0	18.4	-
Root mean square error		1.24	1.31	3.79	5.85

* Indicates temperatures predicted by Ford, Shih and Sebesta (1972)

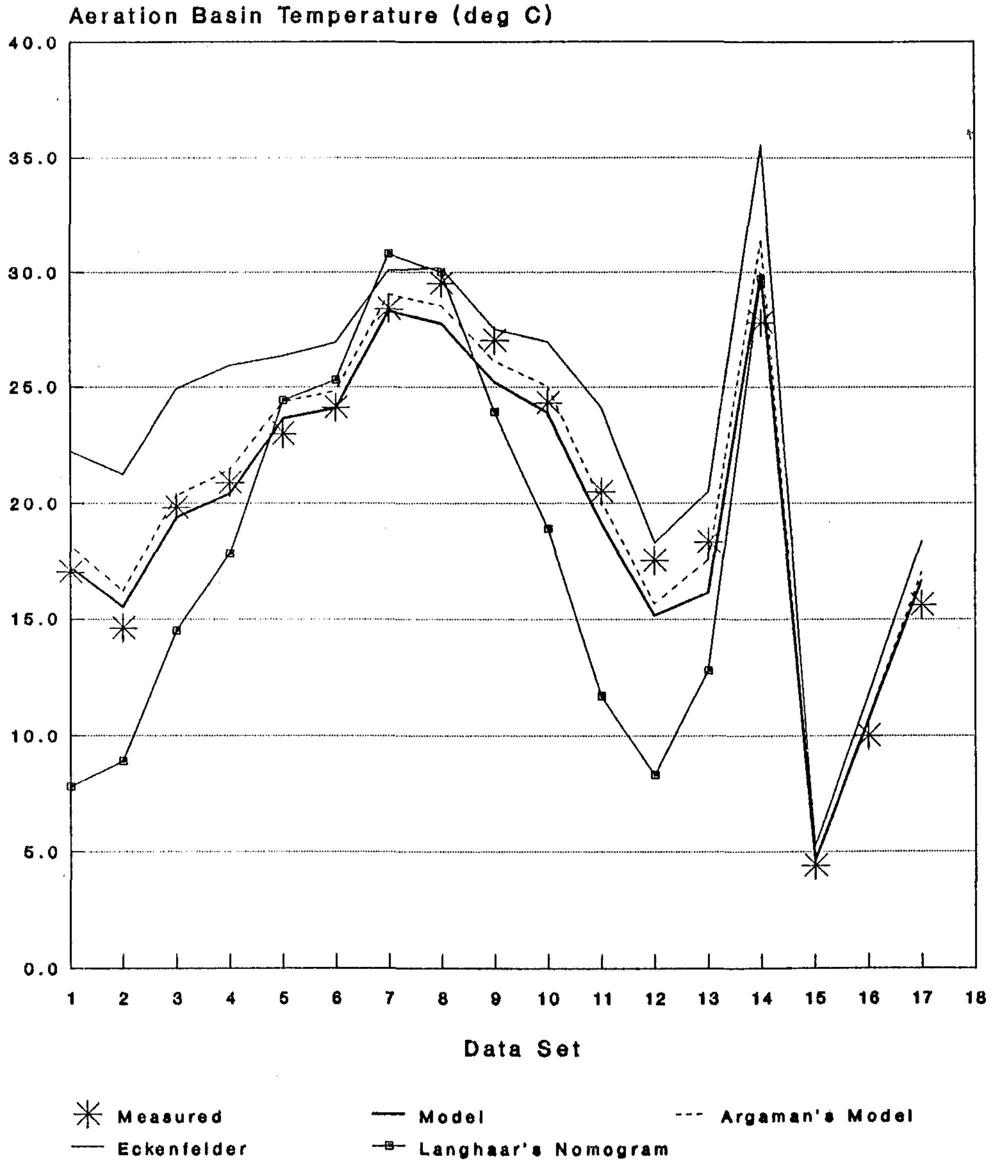

Figure 5 Comparison of Aeration Tank Temperatures Between This Model and Other Models

Figure 6 compares calculated and measured heat loss for this model and Argaman and Adam's model for four data sets. These results confirm accuracy of the model for determination of heat loss and temperature of the aeration tank.

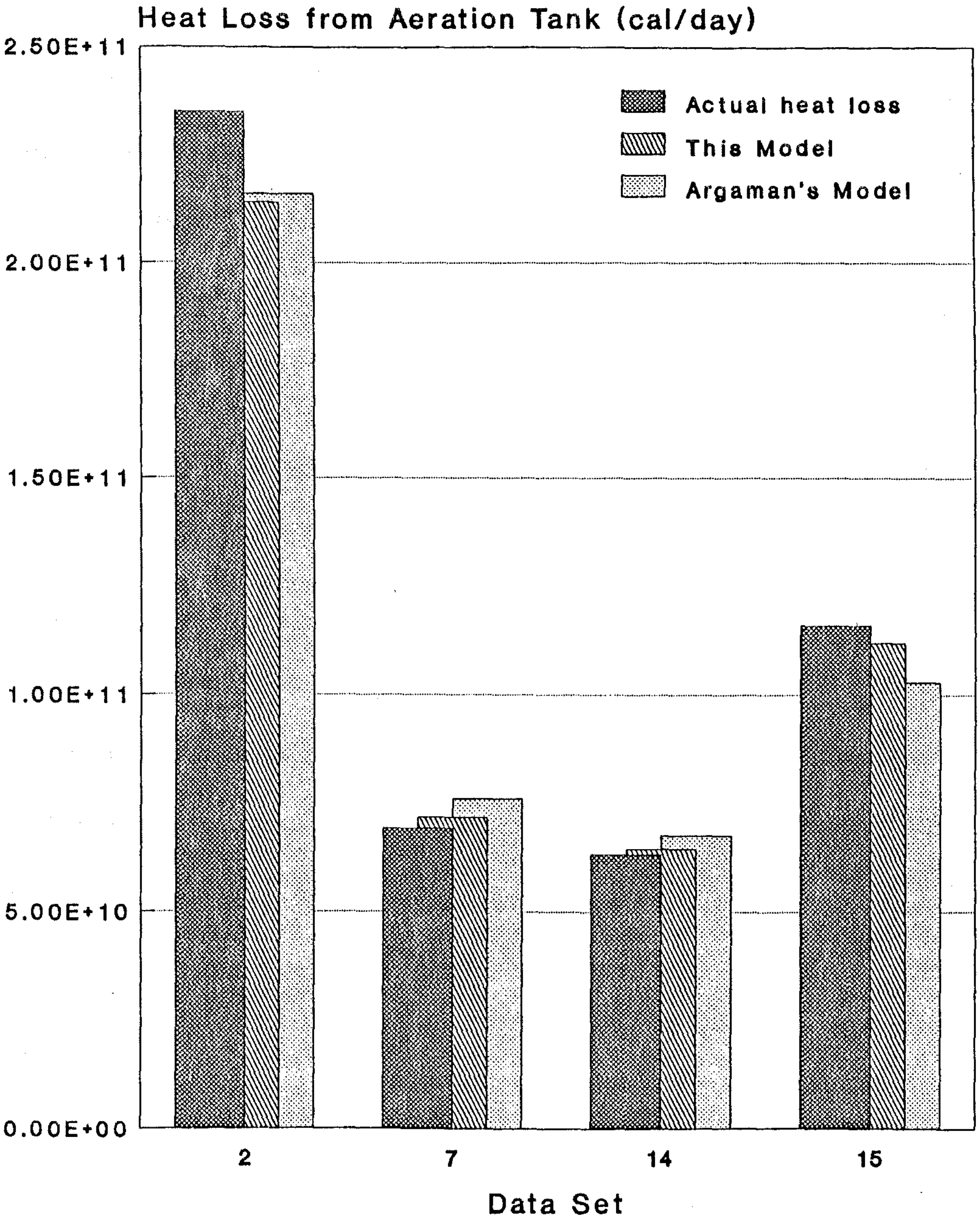

Figure 6 Comparison of Heat Loss from Aeration Tank

4 SENSITIVITY ANALYSIS & ENGINEERING SIGNIFICANCE

This chapter examines the sensitivity of tank temperature to various input variables. The relative importance of these variables is analysed for both surface and diffused aeration systems. Contribution of heat loss components to the overall heat balance is determined and those making a significant impact are identified.

Figure 7 shows a relative contribution of various factors to the overall heat balance for surface and diffused aeration systems. These data indicate that heat loss due to aeration is a single most important factor accounting for 50% of the total heat losses from surface aeration. This results because of contact of a large volume of air with the aerator spray. Surface evaporation and radiation appear to be important elements of heat loss for diffused aeration. Heat loss from tank walls and power input, and heat gained from biological reaction are insignificant parameters in both systems.

A comparison of tank temperature with ambient air and influent temperature is made in Figure 8. For air and influent temperature in the range of 0 - 30°C, tank temperature is nearly an average of these two temperatures. The figure shows that the tank temperature approaches influent temperature when air temperature is less than 0°C.

Figure 9 shows the impact of wind speed (0 - 50 mph) on both surface and diffused aeration systems. The figure shows that the slope of these curves is highest at low wind speeds indicating that low wind speeds may significantly increase tank temperature. Above wind speed of 4 mph, heat losses due to evaporation and aeration for surface aeration system are high enough to reduce the tank temperature to less than air temperature of 25.6°C. For diffused aeration, tank temperature approaches air temperature of 10.5°C for wind speed of 15 mph.

Table 6 shows the tank temperature when air leaving aerator spray is at 90% humidity and is saturated. It shows a drop in tank temperature of the order of 0.8 to 1.5°C when air temperature is more than 15°C. The data indicate that when air temperature is high, ΔT is high because saturation vapor pressure of water increases with temperature.

The sensitivity of tank temperature to fluctuations in wastewater flow rate for surface and diffused aeration is shown in Figures 10 and 11. These figures reveal that tank temperature increases with wastewater flow rate. Also the rate of change of temperature is small at low flow rates.

Figure 12 compares the temperature of diffused aeration system for open and closed tanks. A closed tank is representative of a high purity oxygen activated sludge system. For the closed tank, heat losses from solar radiation, atmospheric radiation, surface evaporation and convection were set to zero in the computer model. Influence of these factors is evident from the results that the temperature of closed tank is 34.4°C which is more than twice the temperature of the open tank. The data also indicate that the temperature of closed tank advances to influent temperature of 37.8°C.

A comparison of tank temperature for surface and diffused aeration systems at different plant locations is made for a hypothetical treatment plant. The same process conditions were applied to treatment plants in 5 U.S. cities which are shown in Figure 13. Meteorological data of these cities are given in Table 7A.

The process data shown in Table 7B are typical for a 15 MGD wastewater treatment plant which receives influent BOD of 175 mg/L with a hydraulic retention time of 5 hours. Influent temperatures are assumed to be 12, 16 and 20°C for winter, yearly average and summer conditions respectively. The calculation for fine bubble diffused system is based rates.

Figure 12 compares the temperature of diffused aeration system for open and closed tanks. A closed tank is representative of a high purity oxygen activated sludge system. For the closed tank, heat losses from solar radiation, atmospheric radiation, surface evaporation and convection were set to zero in the computer model. Influence of these factors is evident from the results that the temperature of closed tank is 34.4°C which is more than twice the temperature of the open tank. The data also indicate that the temperature of closed tank advances to influent temperature of 37.8°C.

A comparison of tank temperature for surface and diffused aeration systems at different plant locations is made for a hypothetical treatment plant. The same process conditions were applied to treatment plants in 5 U.S. cities which are shown in Figure 13. Meteorological data of these cities are given in Table 7A.

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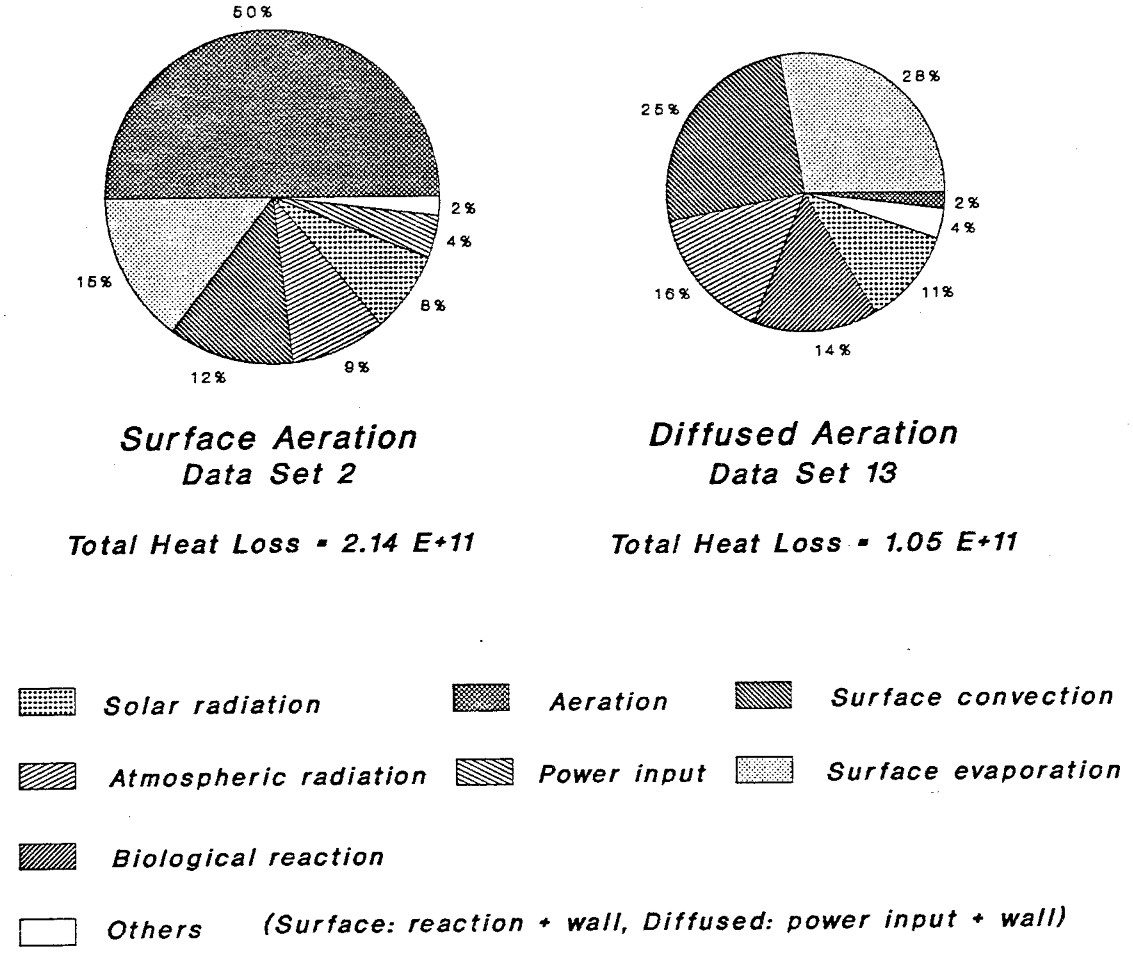
Figure 7 Comparison of Heat Loss from Aeration Tank
(Surface aeration vs Diffused aeration)



Surface Aeration
Data Set 2

Total Heat Loss = 2.14 E+11

Diffused Aeration
Data Set 13

Total Heat Loss = 1.05 E+11

(Surface: reaction + wall, Diffused: power input + wall)

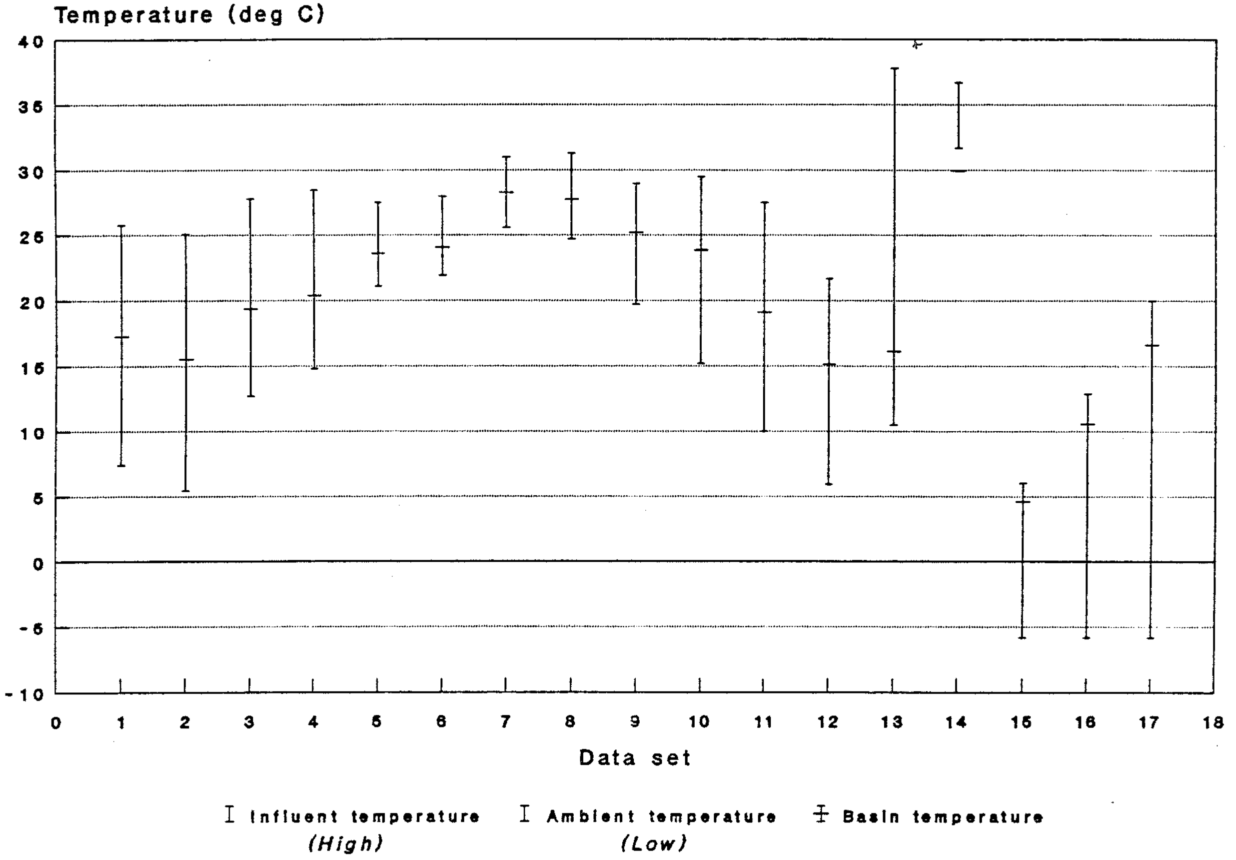
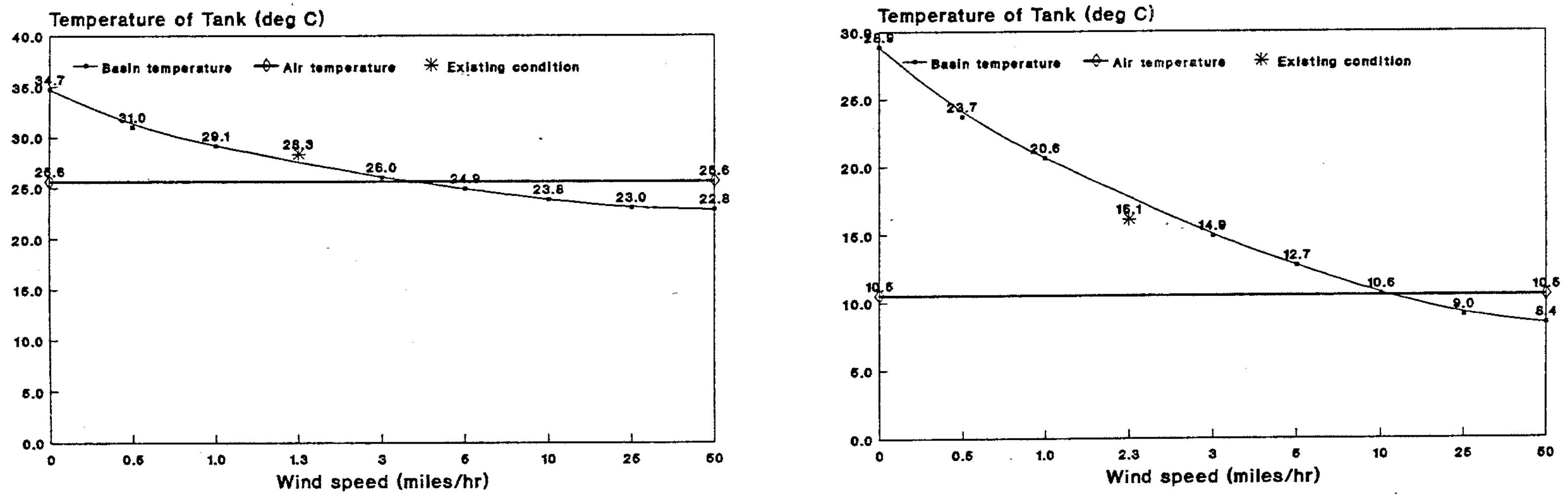


Figure 8 Comparison of Air, Influent and Tank Temperatures



Surface Aeration
Data Set 7

Diffused Aeration
Data Set 13

Figure 9 Influence of Wind Speed on Aeration Tank Temperate

Table 6

Influence of Exit Air Humidity (leaving aerator spray) on Tank Temperature

Data Set	Temperature of air (°C)	Temperature of basin (°C) @ 90% R.H.	Temperature of basin (°C) @ 90% R.H.	ΔT (°C)
1	7.4	17.2	16.7	0.5
2	5.4	15.5	14.7	0.8
3	12.7	19.4	18.5	0.9
4	14.8	20.4	19.3	1.1
5	21.1	23.6	22.7	0.9
6	21.9	24.1	23.1	1.0
7	25.6	28.3	27.4	0.9
8	24.7	27.8	26.9	0.9
9	19.7	25.2	24.7	0.5
10	15.2	23.9	22.9	1.0
11	10.0	19.1	18.3	0.8
12	5.9	15.2	14.8	0.4
13*	10.5	16.1	16.1	0.0
14	31.7	29.9	28.4	1.5
15	-5.8	4.6	4.5	0.1
16	-5.8	10.7	10.5	0.2
17	-5.8	16.6	16.3	0.3

* Diffused aeration

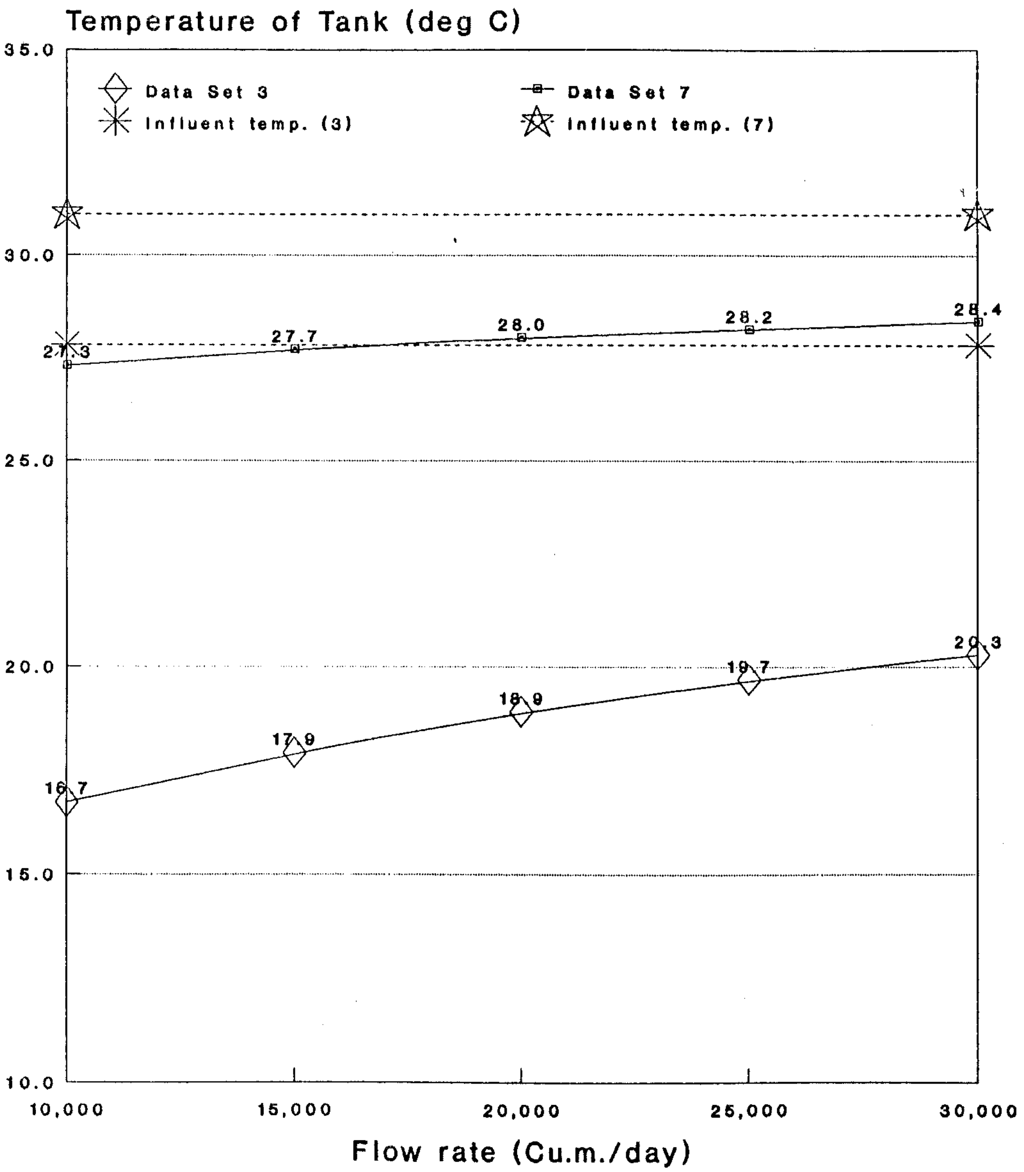


Figure 10 Effect of Wastewater Flow Rate on Tank Temperature (Surface Aeration)

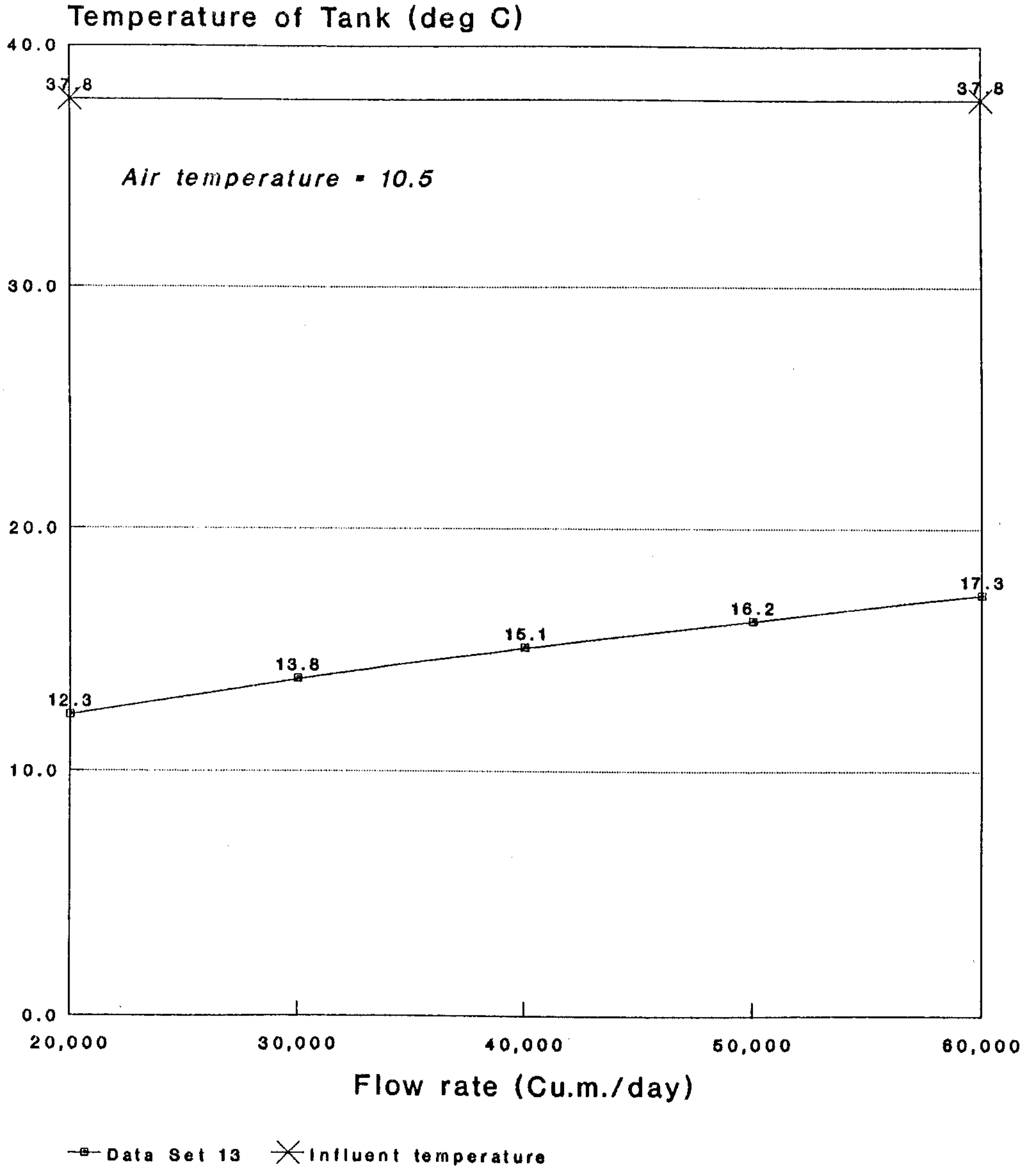


Figure 11 Effect of Wastewater Flow Rate on Tank Temperature (Diffused Aeration)

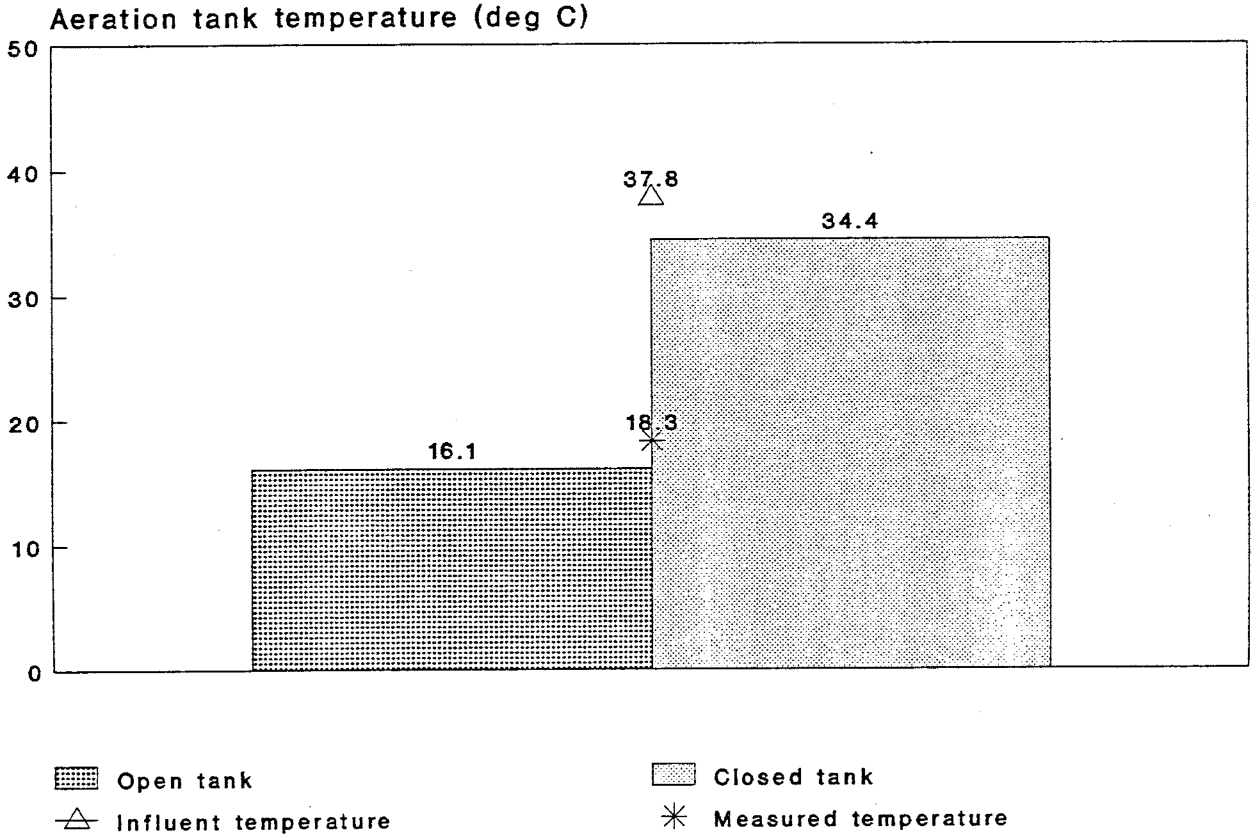


Figure 12 Comparison of Aeration Tank Temperature for Diffused Aeration (Open Tank vs Closed Tank)

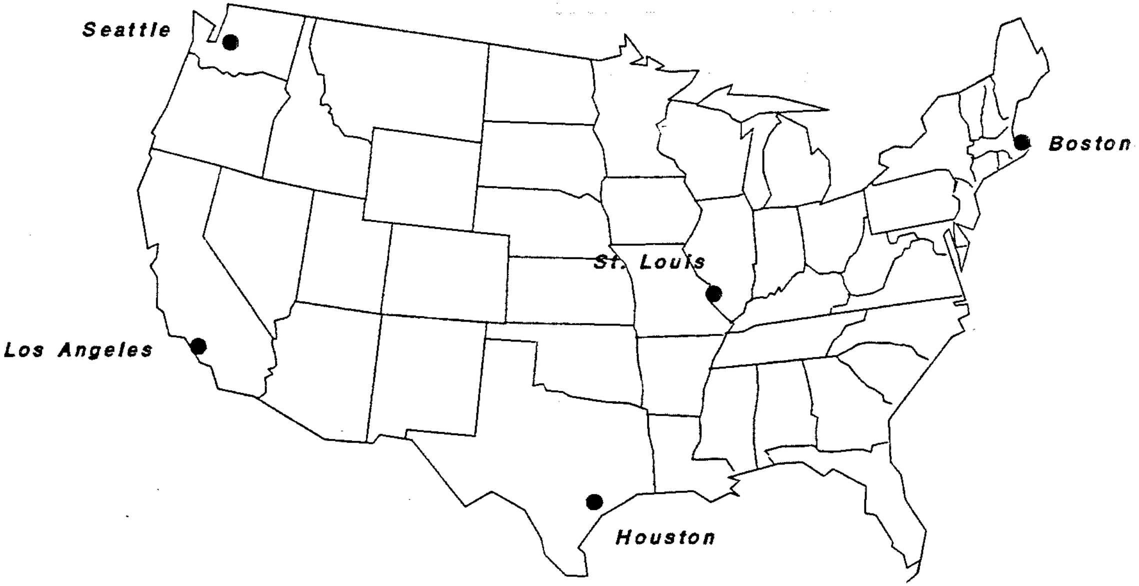


Figure 13 Cities Selected to study the effects of Climatic Conditions on Aeration Tank Temperature

Table 7A

Site - Specific Data for Selected Cities*

City	Latitude (deg)	Air temperature (°C)			Wind speed (m/sec)	Relative humidity (%)	Cloud cover (tenths)
		Average	Summer	Winter			
Los Angeles	34.0	17.0	21.1	13.9	3.3	71	4.7
Seattle	47.5	10.8	17.2	5.6	4.1	73	7.4
Houston	30.0	20.2	27.2	13.3	3.5	76	6.0
Boston	42.4	10.8	21.1	0.6	5.6	66	6.1
St. Louis	38.8	13.0	23.9	1.7	4.3	71	6.0

* Atmospheric radiation factor = 0.75.

Table 7B

Process Data for Comparison of Tank Temperature for Selected Cities

A. Process Data Data

1. No. of aeration basins = 3
2. Tank dimensions = 91 x 10 x 4.5 m.
3. Wastewater flow rate = 56,775 m^3/day
4. Influent wastewater temperature:
   - Average = 16°C
   - Summer = 20°C
   - Winter = 12°C
5. Air flow rate (for diffused aeration) = 3.7 m^3/sec
6. Number of aerators = 19
7. Aerator spray area = 5.9 m^2
8. Power input to each aerator = 20 HP
9. Power input to compressor = 360 HP
10. Efficiency of compressor = 60 %
11. Substrate removal rate = 13,800 kg COD removed /day
12. Overall heat transfer coefficient (tank walls) = 2×10^4 cal/m^2/day/°C
13. Humidity factor for exit air = 0.55

B. Physical Properties of fluid

1. Air density = 1.2 kg/m^3
2. Water density = 1,000 kg/m^3
3. Specific heat of air = 240 cal/kg/°C
4. Specific heat of water = 1,000 cal/kg/°C
5. Emissivity of water = 0.97
6. Reflectivity of water = 0.03

Table 8

Comparison of Tank Temperature for Selected Cities

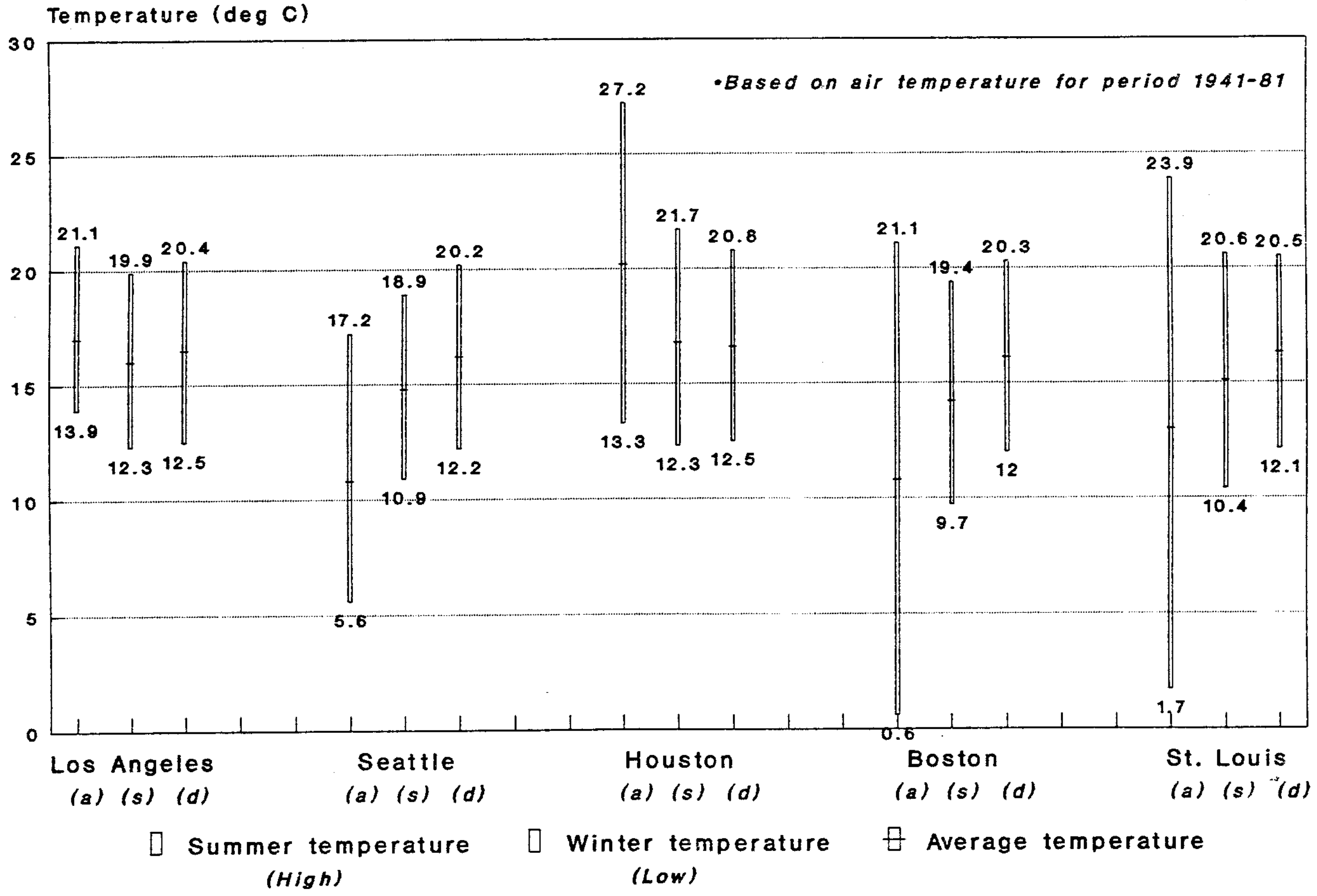
City	Air temperature (°C)			Tank Temperature (Surface Aeration) (°C)			Tank Temperature (Diffused Aeration) (°C)		
	Average	Summer	Winter	Average	Summer	Winter	Average	Summer	Winter
Los Angeles	17.0	21.1	13.9	16.0	19.9	12.3	16.5	20.4	12.5
Seattle	10.8	17.2	5.6	14.8	18.9	10.9	16.2	20.2	12.2
Houston	20.2	27.2	13.3	16.8	21.7	12.3	16.6	20.8	12.5
Boston	10.8	21.1	0.6	14.2	19.4	9.7	16.1	20.3	12.0
St. Louis	13.0	23.9	1.7	15.1	20.6	10.4	16.3	20.5	12.1

Note: Average tank temperature based on influent temperature of 16°C

Summer tank temperature based on influent temperature of 20°C

Winter tank temperature based on influent temperature of 12°C

Figure 14 Effect of Climatic Conditions on Aeration
Tank Temperature for different Cities
(Surface and Diffused Aeration)



(a) Ambient air (s) Tank temperature - Surface aeration (d) Tank temperature - Diffused aeration

5 CONCLUSIONS

This thesis has reviewed environmental engineering literature for heat loss from a water surface. Over ten previous studies have been identified. The best features of each previous study have been incorporated into this study. The general form of equations are introduced for predicting heat losses from aeration and atmospheric radiation. The overall heat balance equation is complex and iterative, and therefore, a computer model using Lotus 123 spreadsheet was designed to solve it.

This model is tested for 17 data sets and the predicted temperature is compared with the results of other models. The temperature of the aeration tank predicted by the model fits well the actual data. The root mean square error is observed to be 1.24 . Hypothetical cases were created for five different cities to predict tank temperature for yearly average, summer and winter air temperatures.

The following results emerge from the sensitivity analysis and the hypothetical case study of 5 cities.

(a) Heat loss for sub-surface aeration system is 50% of the heat losses from surface aeration. The major portion of heat loss in surface aeration is due to evaporation and comprises approximately 50% of the total heat loss.

(b) Wind speed and air humidity are the critical factors in determining temperature of aeration tank. Low wind speeds and high air humidity reduce heat losses and tend to bring tank temperature closer to the influent temperature.

(c) Tank temperature increases with the wastewater flow rate. The rate of change of temperature is observed to be small at low flow rates.

(d) Tank temperature increases substantially for a closed tank sub-surface system because of the lesser influence of meteorological conditions.

Since heat loss from sub-surface aeration is small, its use in cold climates is recommended to avoid the adverse effects of low temperatures on biological reactions. Surface aerators are more useful for warm climates because higher heat losses prevent elevated tank temperatures, which might otherwise inhibit microbial activity.

6 NOMENCLATURE

Equations presented in this study use numerous symbols representing different units. Following is the list of symbols and their units which are used in the Chapter 3 of Model Development and Verification. The symbols do not necessarily apply to the various models reviewed in the literature.

a = Coefficient of equation $H_{sr,o}$

A_s = Surface area of aeration tank, m^2

A_w = Vertical area of tank walls, m^2

b = Coefficient of equation $H_{sr,o}$

c = Coefficient of equation $H_{sr,o}$

C_c = Cloud cover, tenths

c_{hp} = Conversion factor for Horsepower, cal/m^2/HP

c_{pa} = Specific heat of air, cal/kg/°C

c_{pw} = Specific heat of water, cal/kg/°C

d = Day of the year

h_f = Exit air humidity factor, (0-1, =1 for diffused aeration)

h_s = Heat produced from biodegradation of organics, cal/kg COD

h_v = Vapor transfer coefficient, m/day

H_a = Heat loss due to aeration, cal/day

H_{ar} = Heat loss from atmospheric radiation, cal/day

H_c = Heat loss from surface convection, cal/day

H_{ev} = Heat loss from surface evaporation, cal/day

H_p = Heat gained from power input to aerator/compressor, cal/day

H_{rx} = Heat gained from biological reaction, cal/day

$H_{sr,o}$ = Absorbed solar radiation for clear sky conditions, $cal/m^2/day$

H_{sr} = Heat gained from solar radiation, cal/day

H_{tw} = Heat loss from tank walls, cal/day

k = Latitude of the site, degrees

L = Latent heat of vaporization of water, cal/kg

M_w = Molecular weight of water, g/gmole

P = Power of aerator /compressor, HP

Q_a = Air flow rate, m^3/sec

Q_w = Wastewater flow rate, m^3/day

r_h = Relative humidity of ambient air, %

R = Universal gas constant, mm. Hg - liters/ gmole - °K

s = Conversion factor, sec/day

t = Time period, days

T = Temperature of wastewater in aeration tank, °F

T_a = Temperature of ambient air, °C

T_{ae} = Temperature of air/earth, °C

T_{ka} = Temperature of ambient air, °K

T_i = Influent temperature, °C

T_w = Temperature of aeration tank, °C

T_{kw} = Temperature of aeration tank, °K

U = Overall heat transfer coefficient for tank walls, $cal/m^2/day/°C$

V = Volume of aeration tank, m^3

W = Wind velocity (tree top), m/sec

ρ_a = Density of air, kg/m^3

ρ_w = Density of water, kg/m^3

ϵ = Emissivity of water
β = Atmospheric radiation factor
λ = Reflectivity of water
σ = Stefan Boltzman constant, cal/m^2/day/°K^4
υ_a = Saturation vapor pressure of water at temperature T_a, mm. Hg
υ_w = Saturation vapor pressure of water at temperature T_w, mm. Hg
η = Efficiency of compressor, %
ΔH = Net heat exchange with environment, cal/day
ΔS = Substrate removal rate, kg COD removed /day

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8 APPENDIX I

Equations for Average Daily Absorbed Solar Radiation, $H_{sr,o}$ for Clear Sky Conditions

(after Thackston and Parker (1972))

Latitude (deg)	Equations	Standard Error
26	$80.155 - 29.207 \sin[2 \times 3.14159 \times day/366 + 1.679]$	2.76
27	$79.371 - 30.236 \sin[2 \times 3.14159 \times day/366 + 1.713]$	2.40
28	$78.566 - 31.219 \sin[2 \times 3.14159 \times day/366 + 1.710]$	2.33
29	$77.604 - 32.145 \sin[2 \times 3.14159 \times day/366 + 1.740]$	2.10
30	$76.655 - 33.156 \sin[2 \times 3.14159 \times day/366 + 1.728]$	2.02
31	$76.041 - 34.133 \sin[2 \times 3.14159 \times day/366 + 1.694]$	2.21
32	$75.060 - 35.194 \sin[2 \times 3.14159 \times day/366 + 1.737]$	1.85
33	$74.046 - 35.938 \sin[2 \times 3.14159 \times day/366 + 1.734]$	1.61
34	$73.161 - 36.834 \sin[2 \times 3.14159 \times day/366 + 1.727]$	1.52
35	$72.248 - 37.699 \sin[2 \times 3.14159 \times day/366 + 1.738]$	1.32
36	$71.390 - 38.598 \sin[2 \times 3.14159 \times day/366 + 1.721]$	1.32
37	$70.394 - 39.413 \sin[2 \times 3.14159 \times day/366 + 1.730]$	1.07
38	$69.350 - 40.188 \sin[2 \times 3.14159 \times day/366 + 1.741]$	0.86
39	$68.362 - 40.982 \sin[2 \times 3.14159 \times day/366 + 1.739]$	0.73
40	$67.281 - 41.706 \sin[2 \times 3.14159 \times day/366 + 1.742]$	0.61
41	$66.240 - 42.442 \sin[2 \times 3.14159 \times day/366 + 1.736]$	0.58
42	$65.197 - 43.128 \sin[2 \times 3.14159 \times day/366 + 1.740]$	0.54
43	$64.113 - 43.788 \sin[2 \times 3.14159 \times day/366 + 1.739]$	0.63
44	$63.010 - 44.471 \sin[2 \times 3.14159 \times day/366 + 1.739]$	0.76
45	$61.911 - 45.020 \sin[2 \times 3.14159 \times day/366 + 1.740]$	0.93
46	$60.782 - 45.639 \sin[2 \times 3.14159 \times day/366 + 1.735]$	1.16

9 APPENDIX II

MODEL INPUT VARIABLES

A. Site - Specific Data

1. Latitude of plant site, degrees
2. Ambient air temperature, °C
3. Wind speed, m/sec
4. Relative humidity, %
5. Cloud cover, tenths
6. Atmospheric radiation factor

B. Process Data

1. Tank dimensions (L x W x H), m.
2. Wastewater flow rate, m^3/day
3. Influent temperature, °C
4. Air flow rate (for diffused aeration), m^3/sec
5. Number of aerators
6. Aerator spray area, m^2
7. Power input to aerator/compressor, HP
8. Efficiency of compressor (for diffused aeration), %
9. Substrate removal rate, kg COD removed /day
10. Overall heat transfer coefficient for tank walls, cal/m^2/day/°C
11. Humidity factor for exit air

C. Physical Properties of fluid

1. Air density, kg/m^3
2. Water density, kg/m^3
3. Specific heat of air, cal/kg/°C
4. Specific heat of water, cal/kg/°C
5. Emissivity of water
6. Reflectivity of water