Aeration-Basin Heat Loss

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Abstract: Recent developments in wastewater aeration systems have focused on aeration efficiency and minimum energy cost. Many other operating characteristics are ignored. The impact of aeration system alternatives on aeration-basin temperature can be substantial, and design engineers should include potential effects in evaluation of alternatives. To predict aeration-basin temperature and its influence on system design, previous research has been surveyed and a spreadsheet-based computer model has been developed. Calculation has been improved significantly in the areas of heat loss from evaporation due to aeration and atmospheric radiation. The model was verified with 17 literature-data sets, and predicts temperature with a root-mean-squared (RMS) error of 1.24° C for these sets. The model can be used to predict aeration basin temperature for plants at different geographical locations with varying meteorological conditions for surface, subsurface, and high-purity aeration systems. The major heat loss is through evaporation from aeration, accounting for as much as 50%. Heat loss from surface aerators can be twice that of an equivalent subsurface system. Wind speed and ambient humidity are important parameters in determining aeration-basin temperature.

Introduction

The recent emphasis on high-efficiency, low-energy consumption aeration systems has increased the use of fine-bubble, subsurface aeration systems, almost to the exclusion of all other types. Design engineers are choosing this technology over others because of very low energy costs. A factor that is usually not considered is heat loss. Different types of aeration systems can have very different heat losses, which result in different aeration-basin temperatures. In some cases it may be advantageous to avoid aeration-basin cooling, while in others, especially certain industrial wastewater treatment plants, it may be preferable to promote cooling. It is the objective of this manuscript to present a general purpose, comprehensive steady-state temperature prediction model that is broadly applicable to a wide range of meteorological and operational conditions, and is least dependent on empirical constants. Design engineers can use this model to predict equilibrium aeration basin temperature.

To develop the model a literature review was made of all previous efforts to develop quantitative temperature prediction models. The best aspects of each were incorporated into a new spreadsheet-based computer model. Significant improvements were made in the way many calculations are performed and features were added to facilitate the model's use for design.

Background

Previous investigators have attempted to predict heat loss and equilibrium basin temperature for rivers and lakes (Anderson 1954; Harbeck 1962; Meyer

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Note. Discussion open until July 1, 1990. To extend the closing date one month, a written request must be filed with the ASCE Manager of Journals. The manuscript for this paper was submitted for review and possible publication on March 20, 1989. This paper is part of the ***Journal of Environmental Engineering***, Vol. 116, No. 1, February, 1990. ©ASCE, ISSN 0733-9372/90/0001-0070/$1.00 + $.15 per page. Paper No. 24332.

1942; Raphael 1962; Rohwer 1931; Thorne 1951), cooling ponds (Langhaar 1953; Thackston and Parker 1972), aerated lagoons (Barnhart 1968; Friedman and Doesburg 1981), and wastewater treatment plants (Argaman and Adams 1977; Eckenfelder 1966; Ford et al. 1972). Most work has focused on estimating evaporation rates. Eckenfelder (1966) developed an empirical relationship, using only a single parameter, which is widely used today. More recently Ford et al. (1972), Novotny and Krenkel (1973), and Argaman and Adams (1977) have developed more comprehensive models that account for most of the heat loss/gain terms, such as evaporation, solar radiation, conduction, and convective heat losses. Their models provide reasonably accurate, steady-state temperature estimates, but are tedious to perform and require a large amount of site-specific information.

Ford et al. (1972) presented a design approach for predicting temperature for activated sludge aeration basins using mechanical aerators. They used an iteration approach that includes heat loss from the aerator spray, which is calculated from the differential enthalpy of the air flowing through it. Novotny and Krenkel (1973) presented a similar approach but also provided for different evaporation rates of subsurface aeration systems.

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 loss through the basin walls. Their model requires empirical data for determining aerator spray vertical cross-sectional area. Friedman and Doesburg (1981) tested the model of Argaman and Adams using data from eight different industrial bio-treating systems. They concluded that the temperature predicted by their model is accurate to 1–3° C. They conducted a sensitivity analysis to arrive at a general correlation of the heat exchange characteristics of the eight treatment systems.

Model Development and Verification

The equilibrium temperature predicted in this model is obtained from a heat balance around the aeration basin. Various components of heat transfer in the aeration basin are identified, quantified, and arranged for estimating equilibrium aeration basin temperature. The model presented herein applicable to a completely mixed basin under steady-state conditions. The basic assumption of complete mixing implies uniform basin temperature. The model is described more completely by Talati (1988), and only a summary is provided here.

Eq. 1 is the basic heat balance for steady-state conditions.

$$\Delta H = \rho_w c_{pw} Q_w (T_i - T_w) \quad (1)$$

where ΔH = net heat exchange with environment, calories per day; ρ_w = density of water, kilograms per cubic meter; c_{pw} = specific heat of water, calories per kilogram per degree C; Q_w = flow rate of wastewater in aeration basin, cubic meter per day; T_i = influent temperature, degrees C; and T_w = aeration basin temperature, degrees C. Net heat loss, ΔH, represents heat exchange by means of convection, radiation, and evaporation. Various components of the heat exchange with environment are shown in Fig. 1 and the following equation. The positive and negative terms represent heat loss and heat gain, respectively.

$$\Delta H = H_{ar} - H_{sr} + H_{ev} + H_c + H_a - H_p - H_{rx} + H_{tw} \quad (2)$$

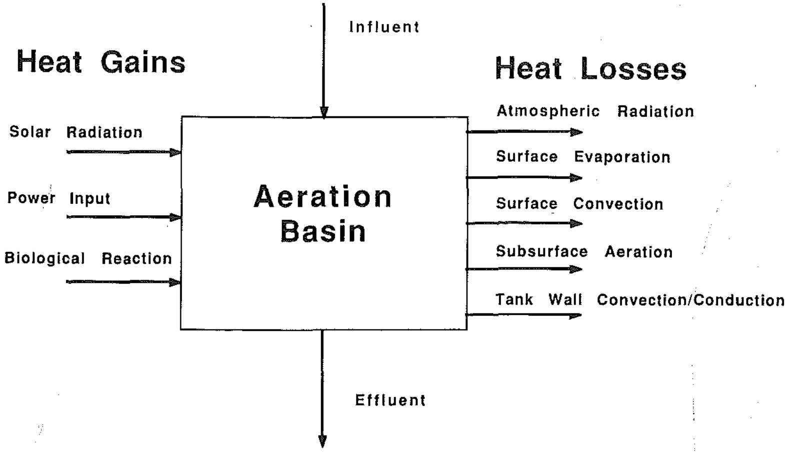


FIG. 1. Aeration-Basin Heat Exchange Components

where H_{ar} = heat loss from solar radiation, calories per day; H_{sr} = heat gained from atmospheric radiation, calories per day; H_{ev} = heat loss from surface evaporation, calories per day; H_c = heat loss from surface convection, calories per day; H_a = heat loss from aeration, calories per day; H_p = heat gained from power input, calories per day; H_{rx} = heat gained from biological reaction, calories per day; and H_{tw} = heat loss through basin walls, calories per day. Each of the terms in Eq. 2 is described in the following sections.

Solar Radiation

Net heat gained from solar radiation is a function of meteorological conditions, site latitude, and the period of the year. The model incorporates Raphael's (1962) correlation:

$$H_{sr} = H_{sr,o}(1 - 0.0071C_c^2)A_s \quad (3)$$

where $H_{sr,o}$ = average daily absorbed solar radiation for clear sky conditions, C_c = cloud cover, tenths; and A_s = basin surface area, m^2.

Absorbed solar radiation for clear skies depends on site latitude, season, and year, and must be estimated, if meteorological data are not available. Thackston and Parker's (1972) correlations can be used, and have been reworked and incorporated into the model as follows:

$$H_{sr,o} = a - b \sin\left(\frac{2\pi d}{366} + c\right) \quad (4a)$$

$$a = 95.1892 - 0.3591k - 8.4537 \times 10^{-3}k^2 \quad (4b)$$

$$b = -6.2484 + 1.6645k - 1.1648 \times 10^{-2}k^2 \quad (4c)$$

$$c = 1.4451 + 1.434 \times 10^{-2}k - 1.745 \times 10^{-4}k^2 \quad (4d)$$

where d = day of the year; and k = latitude of the site, degrees. This correlation is valid between 26° and 46° latitude and is accurate to within ±1% of $H_{sr,o}$.

Atmospheric Radiation

The heat exchange from atmospheric radiation is based on Stefan Boltzman's fourth power radiation law and is expressed as the difference between incoming and back radiation, as follows:

$$H_{ar} = H_{ar,w} - H_{ar,a} \qquad (5)$$

where $H_{ar,w}$ = back radiation from water, calories per day; and $H_{ar,a}$ = net incoming atmospheric radiation, calories per day. The overall equation for heat loss from atmospheric radiation is

$$H_{ar} = [\epsilon\sigma(T_w + 273)^4 A_s] - [(1 - \lambda)\beta\sigma(T_a + 273)^4 A_s] \qquad (6)$$

where ϵ = emissivity of the water surface; σ = Stefan Boltzman constant [1.17×10^{-3} cal/(m^2 day° k^4)]; λ = reflectivity of water; β = atmospheric radiation factor (0.75 − 0.95 for most conditions); and T_a = ambient air temperature, degrees C. Most previous researchers have found that $\epsilon = 0.97$ and $\lambda = 0.03$ are good estimates for the emissivity and reflectivity of water, respectively.

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 the vapor transfer coefficient which is a function of wind velocity. Novotny and Krenkel (1973) suggested that the transfer coefficients for evaporation and convection are the same because the Prandtl numbers in air for both processes are similar. The following are obtained by using their approach.

$$H_c = \rho_a c_{pa} h_v s A_s (T_w - T_a) \qquad (7)$$

$$h_v = 392 A_s^{-0.05} W \qquad (8)$$

where h_v = vapor transfer coefficient, meters per second; ρ_a = density of air, kilograms per cubic meter; c_{pa} = specific heat of air, calories per kilogram per degree C; s = conversion factor, seconds per day; and W = wind velocity at tree top, meters per second.

Evaporative Heat Losses

The heat transfer by surface evaporation depends upon wind velocity, relative humidity, and temperature. Novotny and Krenkel's (1973) method is used here. Their expression shown in Eq. 9 assumes that heat transfer and vapor transfer coefficients are similar.

$$H_{ev} = \left[1.145 \times 10^6\left(1 - \frac{r_h}{100}\right) + 6.86 \times 10^4 (T_w - T_a)\right] e^{0.0604 T_a} W A_s^{0.95} \qquad (9)$$

where r_h = ambient air; percent relative humidity.

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:

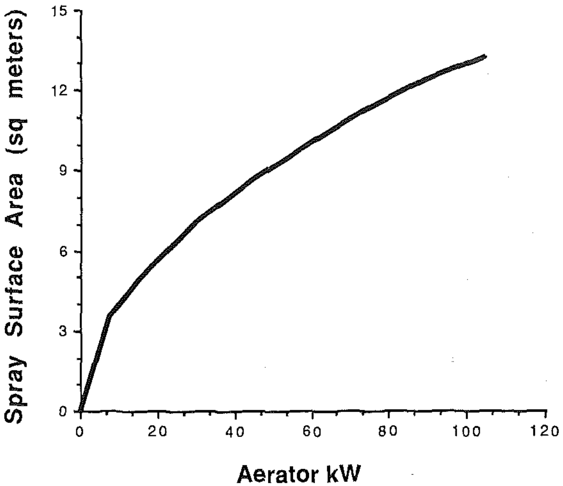


FIG. 2. Spray Area versus Kilowatts for Low-Speed Mechanical Aerators

$$H_a = H_{as} + H_{al} \qquad (10)$$

where H_{as} = sensible heat loss, calories per day; and H_{al} = evaporative heat loss, calories per day. The sensible heat loss for subsurface and surface aerators must be calculated differently, since for subsurface aeration the gas flow rate is known, while the exposure of the spray from a surface aerator to air must be estimated.

For subsurface aeration the sensible heat loss is calculated as follows:

$$H_{as} = Q_a \rho_a c_{pa} s(T_w - T_a) \qquad (13)$$

where h_v = the vapor phase transfer coefficient, calculated as follows:

$$h_v = 392F^{-0.05}W \qquad (12)$$

where F = aerator spray area, square meters. The aerator spray area must be experimentally determined or obtained from manufacturer's data. Fig. 2 shows spray area for one manufacturer's low speed mechanical aerator as a function of aerator kilowatts (Mixing Equipment Company, personal communication, September 1988).

For subsurface aeration the sensible heat loss is calculated as follows:

$$H_{as} = Q_a \rho_a c_{pa} s(T_w - T_a) \qquad (13)$$

where Q_a = air flow rate, cubic meters per second. The heat loss due to evaporation, H_{al}, is calculated as follows:

$$H_{al} = Q_a Ls\left\{\frac{v_w[r_h + h_f(100 - r_h)]}{100} - v_a \frac{r_h}{100}\right\} \qquad (14a)$$

$$H_{al} = \frac{M_w}{R} \frac{Q_a\left\{\frac{v_w[r_h + h_f(100 - r_h)]}{100} - v_a \frac{r_h}{100}\right\}Ls}{(T_a + 273)} \qquad (14b)$$

where h_f = exit-air humidity factor; L = latent heat of vaporization of water, calories per kilogram; v_w = vapor pressure of water at basin temperature, millimeters of mercury; v_a = vapor pressure of water at ambient air temperature, millimeters of mercury; M_w = molecular weight of water; and R = universal gas constant (62.361 mm Hg-l/gmole °K[4]).

For surface aerators the gas flow rate must be estimated from the spray area and wind velocity, as follows:

$$Q_a = NFW \quad (15)$$

where N = number of aerators.

The relative humidity of the gas after contact with water must be known or determined. For subsurface aerators, the relative humidity is virtually 100%, and has been confirmed by offgas measurement field studies conducted by our laboratory. For surface aerators the air is generally less than saturated, and h_f is less than one.

The latent heat of water (L) varies slightly with temperature. Handbook values can be used for specific temperatures, or a regression can be used. Vapor pressure is also a function of temperature. The model uses empirical regressions for these parameters which facilitates its use with spreadsheets or computer programs.

Heat Gain from Power Input

Surface aerators are partially submerged in the aeration basin and are in direct contact with the liquid. 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 the compressor. A portion of the temperature increase during compression is lost as the bubbles expand when they rise through the wastewater. Only the compressor inefficiencies can become heat gain for the aeration basin (e.g., for a 60% efficient compressor, 40% of the brake horsepower at most is translated into heat energy). To calculate this heat gain:

$$H_p = c_{hp}P\left(1 - \frac{\eta}{100}\right) \quad (16)$$

where c_{hp} = constant for conversion from horsepower to calories; calories per horsepower; P = power of aerator/compressor, horsepower; and η = efficiency of compressor, percent.

Biological Reaction

Biological reactions provide heat to aeration basins 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. Argaman and Adams (1977), assuming a net cell yield of 0.25 g of volatile suspends solids (vss)/g of chemical oxygen demand (COD) removed, estimated that the heat released from biological reactions, h_s, as 1,800 cal/g COD removed. Eq. 17 incorporates this into the model:

$$H_{rx} = h_s \Delta S \quad (17)$$

where ΔS = organic removal rate, grams of COD removed per day.

Tank-Wall Heat Loss

Heat is lost from conduction and convection through tank walls. The overall heat transfer equation for basin-wall heat loss is expressed as follows:

$$H_{tw} = UA_w(T_w - T_{ae}) \qquad (18)$$

where U = heat transfer coefficient, cal/day/m^2/°C; A_w = basin wall area, square meters; and T_{ae} = temperature of air/earth, degrees C.

Heat Balance

Each heat transfer term discussed previously is combined with continuity terms to produce the overall heat balance as shown as follows:

$$Q_w \rho_w c_{pw} T_w + \epsilon\sigma T_{kw}^4 A_s + \rho_a c_{pa} h_v A_s S T_w + 6.86 \times 10^4 e^{0.0604T_a} \times W A_s^{0.95} T_w$$

$$+ \frac{M_w}{R} \cdot \frac{Q_a}{T_{ka}} L s v_w \left[\frac{r_h + h_f(100 - r_h)}{100}\right] + UA_w T_w = Q_w \rho_w c_{pw} T_i$$

$$+ (1 - \lambda)\beta\sigma T^{4ka} A_s + H_{sr,o}(1 - 0.0071 C_c^2) A_s + \rho_a c_{pa} h_v A_s s T_a - 1.145$$

$$\times 10^6 e^{0.0604T_a} W A_s^{0.95} \left(1 - \frac{r_h}{100}\right) + 6.86 \times 10^4 e^{0.0604T_a} W A_s^{0.95} T_a$$

$$- H_{as} + \frac{M_w}{R} \cdot \frac{Q_a}{T_{ka}} L s v_a \frac{r_h}{100} + c_{hp} P + h_s \Delta S + UA_w T_{ae} \qquad (19)$$

where v and L = functions of temperature; and $H_{sr,o}$ = a function of latitude. They can be correlated from regression equations, as mentioned earlier, or obtained from standard or site-specific information. A spreadsheet (Lotus 1-2-3) computer program is designed to solve the complex overall equation using iteration for the implicit terms. The required input variables are:

- Site-specific data:
  1. Latitude of plant site, degrees.
  2. Ambient air temperature, degrees C.
  3. Wind speed, meters per second.
  4. Relative humidity, percentage.
  5. Cloud cover, tenths.
  6. Atmospheric radiation factor.
- Process data:
  1. Tank dimensions ($L \times W \times H$), meters.
  2. Wastewater flow rate, cubic meters per day.
  3. Influent temperature, degrees C.
  4. Airflow rate (for diffused aeration), cubic meters per second.
  5. Number of aerators.
  6. Aerator spray area, square meters.
  7. Power input to aerator/compressor, horsepower.
  8. Efficiency of compressor (for diffused aeration), percentage.
  9. Substrate removal rate, kilograms COD removed per day.
  10. Overall heat transfer coefficient for tank walls, cal/m^2/day/°C.
  11. Humidity factor for exit air.

- Physical properties of fluid:
  1. Air density, kilograms per cubic meter.
  2. Water density, kilograms per cubic meter.
  3. Specific heat of air, calories per kilogram per degree C.
  4. Specific heat of water, calories per kilogram per degree C.
  5. Emissivity of water.
  6. Reflectivity of water.

Model Verification

In order to establish validity of this model, the 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 et al. (1972). These data cover a wide range of input variables for both surface and diffused aeration systems.

Since the data presented by Ford et al. (1972) do not include substrate removal rate, it was calculated using their estimated conversion efficiency, assuming that the inlet COD is 270 mg/L. Tables 1 and 2 show the input variables for the 17 data sets used for calibration and verification. To show the improved accuracy of the new model, it is compared to previous models by Eckenfelder (1966), Argaman and Adams (1977), and Langhaar (1953).

The model results are shown in Fig. 3. The temperatures predicted by this model agree well with the measured temperature. The root-mean-square (RMS) error for this model is 1.24, which is an improvement over other models. The results yielded by Argaman and Adam's model closely follow temperature predicted by the model. Eckenfelder's equation follows the pattern of temperature changes but differs in magnitude on an average of +3.8° C. Langhaar's nomogram relies mainly on meteorological factors and shows large deviations from the measured temperature.

TABLE 1. Input Process Data Selected for Verification

Data set (1)	Flow rate (m^3/d) (2)	Influent temperature (°C) (3)	Tank surface area (m^2) (4)	Vertical wall area (m^2) (5)	Average number of aerators (6)	Aerator spray area (m^2) (7)	Airflow rate (m^3/s) (8)	Power input (hp) (9)	Substrate removal rate (10)	Reference (11)
1	22,730	25.8	11,150	13,380	11.5	11.1	—	1,150	27,700	Argarman (1977)
2	22,350	25.1	11,150	13,380	9.1	11.1	—	910	18,600	Argarman (1977)
3	23,110	27.8	11,150	13,380	9.3	11.1	—	930	19,500	Argarman (1977)
4	23,600	28.5	11,150	13,380	9.6	11.1	—	960	20,100	Argarman (1977)
5	25,110	27.5	11,150	13,380	10.7	11.1	—	1,070	21,100	Argarman (1977)
6	25,260	28.0	11,150	13,380	10.8	11.1	—	1,080	21,100	Argarman (1977)
7	26,630	31.0	11,150	13,380	11.8	11.1	—	1,180	31,800	Argarman (1977)
8	27,050	31.3	11,150	13,380	11.8	11.1	—	1,180	30,400	Argarman (1977)
9	28,450	29.0	11,150	13,380	10.5	11.1	—	1,050	38,100	Argarman (1977)
10	25,340	29.5	11,150	13,380	10.0	11.1	—	1,000	36,400	Argarman (1977)
11	22,610	27.5	11,150	13,380	9.8	11.1	—	980	27,900	Argarman (1977)
12	19,730	21.7	11,150	13,380	5.1	11.1	—	510	12,700	Argarman (1977)
13	49,250	37.8	174,630	181,000	—	—	56.6	4,900	11,340	Argarman (1977)
14	7,100	36.7	4,200	5,500	5	5.88	—	100	1,825	Ford et al. (1972)
15	72,300	6.0	9,960	11,150	10	7.43	—	1,000	6,200	Ford et al. (1972)
16	72,300	12.9	9,960	11,150	10	7.43	—	1,000	9,350	Ford et al. (1972)
17	72,300	20.1	9,960	11,150	10	7.43	—	1,000	13,750	Ford et al. (1972)

TABLE 2. Input Meteorological Data Selected for Verification

Data set (1)	Solar radiation (clear sky) ($Kcal/m^2/d$) (2)	Air temperature (°C) (3)	Wind speed (m/s) (4)	Relative humidity (5)	Cloud cover (tenths) (6)
1	2,280	7.4	3.9	82	8.1
2	3,120	5.4	5.0	73	6.1
3	4,360	12.7	4.8	74	7.2
4	5,530	14.8	4.8	66	6.0
5	6,440	21.1	3.9	74	6.5
6	7,090	21.9	4.2	73	5.6
7	7,090	25.6	2.9	74	4.8
8	6,510	24.7	3.1	77	6.2
9	5,270	19.7	3.2	83	6.8
10	3,770	15.2	2.9	69	2.9
11	2,600	10.0	4.0	73	6.1
12	2,280	5.9	4.3	79	7.2
13	2,925	10.5	5.2	70	6.4
14	4,460	31.7	3.6	53	4.0
15	1,670	−5.8	4.0	71	3.0
16	1,670	−5.8	4.0	71	3.0
17	1,670	−5.8	4.0	71	3.0

Sensitivity Analysis and Engineering Significance

It is useful to show the contribution of the various heat loss/gain terms to the total heat balance. In this way model simplifications can be made for specific circumstances. The following analysis is based upon specific data sets shown in Tables 1 and 2.

Fig. 4 and Table 3 show 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 the 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 basin walls and power input, and heat gained from biological reaction are insignificant parameters in both systems. Heat gain from biological reaction may be significant for industrial systems.

Fig. 5 shows the impact of wind speed (0–22 m/s) on both surface and diffused aeration systems. The figure shows that the slope of these curves is highest at low wind speeds, indicating that reduced wind speeds may significantly increase basin temperature. Above a wind speed of 1.7 m/s, heat losses due to evaporation and aeration for surface aeration system are high enough to reduce the basin temperature to less than air temperature of 25.6° C. For diffused aeration, basin temperature approaches air temperature of 10.5° C for wind speed of 7 m/s.

Table 4 shows the basin temperature for two cases when air leaving the aerator spray is at 90% and 100%. It shows a drop in basin temperature of the order of 0.8–1.5° C when air temperature is more than 15° C. The data indicate that when air temperature is high, ΔH is high because saturation

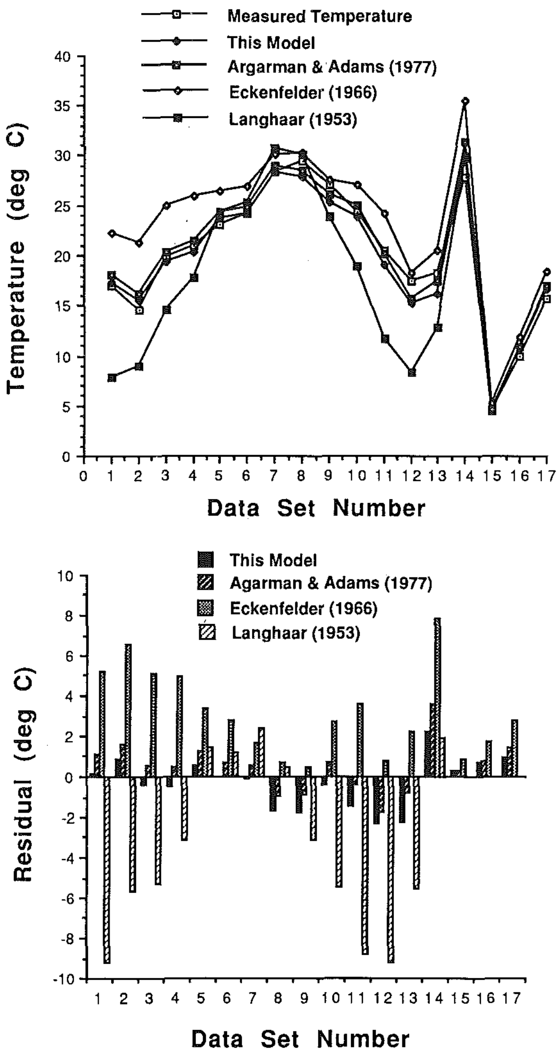


FIG. 3. Predicted Temperature and Measured Temperature for 17 Data Sets

vapor pressure of water increases with temperature.

The model can be used to compare the temperature of an aeration system for open or closed basins. A closed basin is representative of a sludge system activated by high-purity oxygen. For a closed basin, heat losses from solar radiation, atmospheric radiation, surface evaporation, and convection should be set to zero in the computer model. Data set 13 was analyzed in this fashion. The influent temperature and basin temperatures were 37.8° C and 18.3° C, respectively, for this data set. The model predicts that the basin temperature would have been 34.4° C if the basin were closed.

To demonstrate the difference between the cooling characteristics of sur-

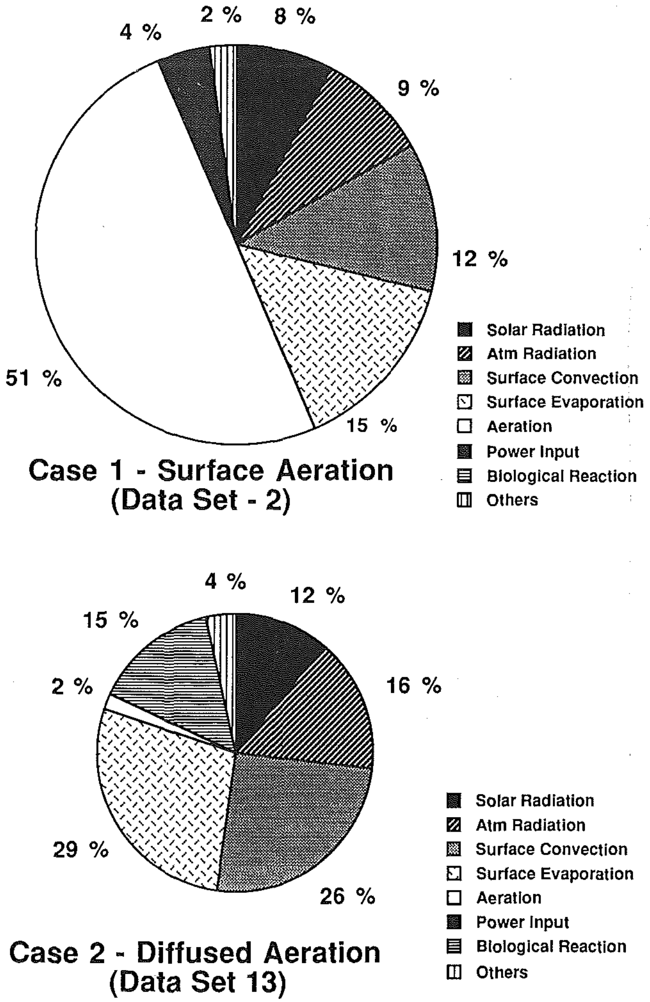


FIG. 4. Comparison of Heat Loss Terms for Surface and Diffused Aeration System

face and diffused aeration systems, a comparison was made for five different plant locations of a hypothetical treatment plant. The same process conditions were assumed for five U.S. cities. Only meteorological data were varied and are shown in Table 5.

The following process data are typical for a 15-mgd wastewater treatment plant which has a primary clarifier five-day biochemical oxygen demand (BOD_5) of 175 mg/l.

- Process data:
  1. Number of aeration basins = 3.
  2. Tank dimensions = 91 × 10 × 4.5 m.

TABLE 3. Comparison of Heat Loss Terms

Term (1)	Heat Loss or Gain (%)	
	Date set 2 (surface) (2)	Date set 13 (diffused) (3)
Solar radiation	8.1	11.6
Atmospheric radiation	9.4	16.3
Surface convection	12.3	26.3
Surface evaporation	14.9	28.8
Aeration	51	2
Power input	4.2	0
Biological reaction	0	14.9
Others	2	3.6

3. Wastewater flow rate = 56,775 m^3/day.
4. Influent wastewater temperature:
   Average = 16° C.
   Summer = 20° C.
   Winter = 12° C.
5. Airflow rate (for diffused aeration) = 3.7 m^3/s.
6. Number of aerators = 19.
7. Aerator spray area = 5.9 m^2.
8. Power input to each aerator = 15 hp.
9. Power input to compressor = 270 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/°C.
13. Humidity factor for exit air = 0.55.

- Fluid properties:

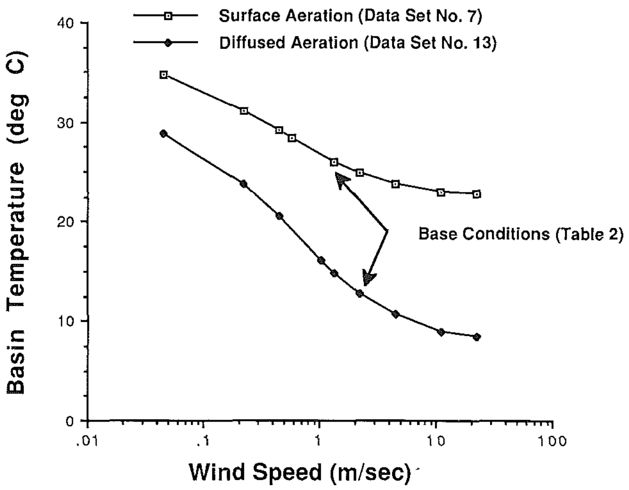


FIG. 5. Effect of Wind Speed on Basin Temperature

TABLE 4. Aeration Basin Temperature for Two Spray Relative Humidities

Data set (1)	Air temperature (°C) (2)	Aeration Basin Temperature (°C) At 90% relative humidity (3)	At 100% relative humidity (4)	ΔT (°C) (5)
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

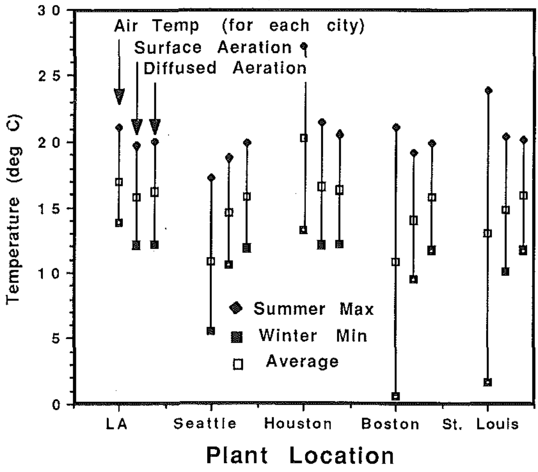
1. Air density = 1.2 kg/m^3.
2. Water density = 1,000 kg/m^3.
3. Specific heat of air = 240 cal/kg/°C.
4. Specific heat of water = 1,000 cal/kg/°C.
5. Emissivity of water = 0.97.
6. Reflectivity of water = 0.03.

The aeration basin hydraulic retention time is five 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 on 28% standard oxygen transfer efficiency (SOTE), and an αSOTE of 7%. The results presented in Fig. 6 confirm lower heat loss for the diffused aeration system. For Boston, Massachusetts, and St. Louis, Missouri, the difference in aeration-basin temperature for the two aeration

TABLE 5. Site-Specific Data for Selected Cities[a]

City (1)	Latitude (degree) (2)	Air Temperature (°C) Average (3)	Summer (4)	Winter (5)	Wind speed (m/s) (6)	Relative humidity (%) (7)	Cloud cover (tenths) (8)
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

[a]Atmospheric radiation factor = 0.75.



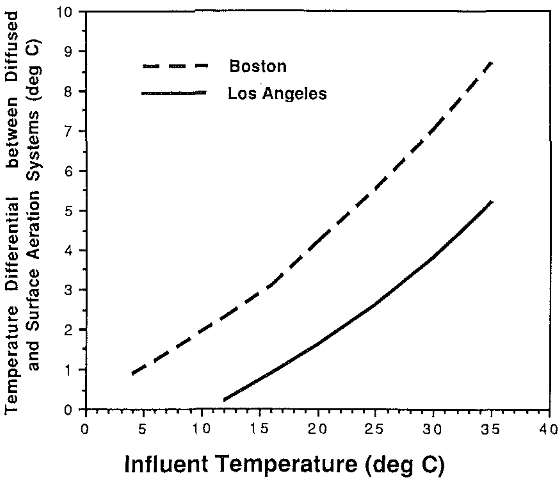


FIG. 6. Comparison of Basin Temperatures of Surface and Diffused Aeration Systems for Different Locations

systems is 3° C, which is large enough to impact rates of biological reactions, such as nitrification. The lower portion of Fig. 6 shows the basin temperature as a function of influent temperature, for winter conditions in Boston, Massachusetts, and Los Angeles, California. The basin temperature difference between surface and diffused aeration increases with rising influent temperature.

Conclusions

The model developed herein is tested for 17 data sets and the predicted temperature is compared with the results of other models. The temperature of the aeration basin predicted by the model fits the data well. The root-mean-squared-error is 1.24° C. Hypothetical cases were created for five different cities to predict basin temperature for yearly average, summer, and winter air temperatures.

Heat loss for subsurface 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. Wind speed and high air humidity are critical factors in determining the temperature of the aeration basin. Low wind speed and high air humidity reduce heat losses. Aeration basin tank temperature is substantially greater for a closed basin subsurface system because of the lesser influence of meteorological conditions and surface evaporation.

Subsurface aeration is preferred in cold climates because of its reduced heat loss. Surface aerators are more useful for warm wastewater in hot climates because increased heat loss may prevent elevated basin temperatures, which might inhibit microbial activity.

Acknowledgment

The senior writer is thankful to CM&E International and the University of California, Los Angeles, for making available all the resources needed for this study.

Appendix I. References

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Appendix II. Notation

The following symbols are used in this paper:

A_s = surface area of aeration basin, m^2;
A_w = basin wall area, m^2;
a = coefficient of equation $H_{sr,o}$;
b = coefficient of equation $H_{sr,o}$;
C_c = cloud cover, tenths;
c = coefficient of equation $H_{sr,o}$;
c_{hp} = conversion factor for horsepower, cal/hp;
c_{pa} = specific heat of air, cal/kg/°C;
c_{pw} = specific heat of water, cal/kg/°C;
d = day of year;
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 through basin walls, cal/day;
ΔH = net heat exchange with environment, cal/day;
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/s;
k = latitude of 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/s;
Q_w = wastewater flow rate, m^3/day;
R = universal gas constant, mm Hg-liters/gmole-°K^4;
r_h = relative humidity of ambient air, percent;
ΔS = substrate removal rate, *g*COD removed/day;
s = conversion factor, s/day;
T_a = temperature of ambient air, °C;
T_{ae} = temperature of air/earth, °C;
T_i = influent temperature, °C;
T_{ka} = temperature of ambient air, °K;
T_{kw} = temperature of aeration basin, °K;
T_w = temperature of aeration basin, °C;
U = overall heat transfer coefficient for basin walls, cal/m^2/day/°C;
V_a = vapor pressure of water at temperature T_a, mm Hg;

V_w = vapor pressure of water at temperature T_w, mm Hg;
W = wind velocity (tree top), m/s;
ρ_a = density of air, kg/m^3;
ρ_w = density of water, kg/m^3;
ϵ = emissivity of water surface;
β = atmospheric radiation factor;
λ = reflectivity of water;
σ = Stefan Boltzman constant, $cal/m^2/day/°K^4$; and
η = efficiency of compressor, percent.

ERRATA

Aeration-Basin Heat Loss[a]

The following corrections should be made to the original paper:

Remove s from equation 7.

Page 74, replace the sentence above equation 13, line 7 with

For surface aeration the sensible heat loss is calculated as:

Page 74, replace equation 13, incorrectly numbered 13, on line 8 with

$$H_{as} = h_v \rho_a c_{pa} \; A_s (T_w - T_a) \tag{11}$$

Page 74, replace equations 14a and 14b with

$$H_{al} = \frac{M_w \; Q_a Ls}{100\,R} \left\{ \frac{v_w [r_h + h_f(100 - r_h)]}{(T_w + 273)} - \frac{v_a \; r_h}{(T_a + 273)} \right\} \tag{14}$$

Page 85, Add to Notation:

F = aerator spray area (m^2)

Page 85, replace the definition of R as follows.

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

[a] February 1990, Vol. 116, No. 1, by S.M. Talati and M.K. Stenstrom (Paper 24332)