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Temperature modelling and prediction for activated sludge systems

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ABSTRACT

Temperature is an important factor affecting biomass activity, which is critical to maintain efficient biological wastewater treatment, and also physiochemical properties of mixed liquor as dissolved oxygen saturation and settling velocity. Controlling temperature is not normally possible for treatment systems but incorporating factors impacting temperature in the design process, such as aeration system, surface to volume ratio, and tank geometry can reduce the range of temperature extremes and improve the overall process performance. Determining how much these design or up-grade options affect the tank temperature requires a temperature model that can be used with existing design methodologies. This paper presents a new steady state temperature model developed by incorporating the best aspects of previously published models, introducing new functions for selected heat exchange paths and improving the method for predicting the effects of covering aeration tanks. Numerical improvements with embedded reference data provide simpler formulation, faster execution, easier sensitivity analyses, using an ordinary spreadsheet. The paper presents several cases to validate the model.

Key words | activated sludge systems, steady state modelling, temperature, wastewater treatment

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NOMENCLATURE

ρ_a'	density of air under the cover ($kg\,m^{-3}$)
ρ_w	density of water ($kg\,m^{-3}$),
μ	dynamic viscosity ($N\,s\,m^{-2}$),
v	kinematic viscosity ($m^2\,s^{-1}$),
A_s	aeration tank surface area (m^2),
c_{pw}	specific heat of water ($J\,kg^{-1}{}^\circ C^{-1}$),
c_{pa}	specific heat of air ($J\,kg^{-1}{}^\circ C^{-1}$),
CASP	Conventional Activated Sludge Plant,
D_{AB}	binary mass diffusion coefficient (A = water vapour, B = air, $m^2\,s^{-1}$),
h	heat transfer coefficient ($W\,K^{-1}\,m^{-2}$),
h_m	mass transfer coefficient ($m\,s^{-1}$),
k	thermal conductivity ($W\,m^{-1}{}^\circ C^{-1}$),
N–DN	nitrification–denitrification,
Nu	Nusselt number (dimensionless),
Pr	Prandtl number (dimensionless),
q_a	air flow rate ($m^3\,s^{-1}$),
q_{ev}	external ventilation rate under the cover ($m^3\,s^{-1}$),
q_v	total ventilation flow rate under the cover ($m^3\,s^{-1}$),
q_w	volumetric flow rate ($m^3 day^{-1}$),
Q_c	surface convection heat (W),
Q_{cover}	cover conduction heat exchange (W),
Q_e	surface evaporation heat (W),
Q_t	net heat exchange (W),
Re	Reynolds number (dimensionless),
Sh	Sherwood number (dimensionless),
Sc	Schmidt number (dimensionless),

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T_a ambient air temperature (°C),
T_a' temperature of the air under the cover (°C),
T_i temperature of influent stream (°C),
T_w tank equilibrium temperature (°C),

INTRODUCTION

The influence of temperature on wastewater treatment efficiency is universally known. Temperature is an important factor affecting biomass activity, which is critical to maintain efficient biological wastewater treatment. It also affects other physiochemical properties such as dissolved oxygen saturation concentration, viscosity, and settling velocity (Sedory & Stenstrom 1995).

Controlling aeration tank temperature is not normally feasible, since heating or cooling is cost-prohibitive; however, incorporating factors affecting aeration tank heat transfer in the design can reduce the range of temperature extremes, improving overall process performance and reducing process variability. In most cases it will be advantageous to avoid aeration tank heat loss, while in others, especially certain industrial wastewater treatment plants, it may be preferable to promote cooling. When covering tanks for air quality purposes, temperature impacts should be considered, and when designing or replacing aeration systems the cooling effects of different systems (surface aeration > coarse bubble > fine bubble; Talati & Stenstrom 1990) should also be considered. Determining how much these design or upgrade options affect the tank temperature requires a temperature model that can be used with existing design methodologies and is suitably flexible to evaluate alternatives and facilitate sensitivity studies.

Previous published models for steady state (Argaman & Adams 1977; Talati & Stenstrom 1990) and dynamic temperature prediction (Sedory & Stenstrom 1995; Scherfig *et al.* 1996), present different data requirements and provide results with varying accuracy and precision, ranging from $\pm 0.5-1$°C to $\pm 1-3$°C. The earliest temperature models included only an empirical relationship to estimate the evaporation rate (Eckenfelder 1966). Later improvements by Ford *et al.* (1972) and Novotny & Krenkel (1973) included more heat exchange terms in an expanded heat balance. Argaman & Adams (1977) extended Novotny & Krenkel (1973) model by including heat exchange terms for mechanical power input, biological reactions, and basin wall conduction. Talati & Stenstrom (1990) simplified and expanded the model introducing new functions for the aeration-system heat-exchange terms. Brown & Enzminger (1991) presented a temperature model with similar features to the Talati & Stenstrom (1990) model, and applied it to equalization tanks and trickling filters. Afterwards Sedory & Stenstrom (1995) and Scherfig *et al.* (1996) expanded the steady state equations for dynamic simulation of treatment plants, incorporating daily changes in wastewater temperature and weather conditions. Recently, Gillot & Vanrolleghem (2003) compared the Talati and Stenstrom model to a simplified model (Van der Graaf 1976) and showed that for specific cases (i.e. aerated lagoons with surface aerators) the predictions were equally valid, even if Van der Graaf (1976) model didn't account for different evaporation heat losses related with different aeration systems. In 2005 Makinia *et al.* modified the Sedory & Stenstrom (1995) model adopting a biological-processes-heat exchange term from la Cour Jansen *et al.* (1992) based on Gibb's free energy terms; the previous equation, based only on organic-substrate removal, was expanded to account the effect of nitrification and denitrification in advanced wastewater-treatment systems with nitrogen removal.

This paper presents a new steady state temperature model developed improving the Sedory & Stenstrom (1995) model, incorporating the biological-processes-heat exchange term of Makinia *et al.* (2005), introducing new functions for surface convection and evaporation heat exchange terms, and improving the approach to predict the effects of covering aeration tanks. Additional numerical improvements allow faster execution, simpler formulation and sensitivity analysis in an ordinary spread sheet. Several scenarios studies are presented to validate the model and demonstrate its capabilities.

METHODOLOGY

This paper describes only the major improvements over previously published models. The detailed model is available for download at www.epl.eng.uci.edu, as well as the methods for applying the model to covered tanks.

Model development

For a generic aeration tank, the overall energy balance in steady conditions is given by:

$$Q_t = \rho_w q_w c_{pw}(T_w - T_i) \quad (1)$$

where

Q_t = net heat exchange (W)
ρ_w = density of water (kg m^{-3}),
c_{pw} = specific heat of water (J kg^{-1}°C^{-1}),
q_w = wastewater flow rate (m^3 day^{-1}),
T_w = tank equilibrium temperature (°C),
T_i = temperature of influent stream (°C).

The net heat exchange is the sum of the heat exchange phenomena as follows:

$$Q_t = Q_{sr} + Q_p + Q_b + Q_{lr} + Q_e + Q_c + Q_a + Q_{tw} \quad (2)$$

Solar thermal radiation (Q_{sr}), biological reaction (Q_b) and power input (Q_p) terms are always positive and represent heat gains, while long wave thermal radiation (Q_{lw}), convection (Q_c), evaporation (Q_e) and tank wall conduction (Q_{tw}) terms can be gains or losses, with appropriate signs, depending on site-specific conditions. The equilibrium temperature of the tank T_w is a state variable that can be calculated after the heat exchange terms are determined. However, many heat exchange terms are dependent on the equilibrium temperature T_w, and an iterative calculation method is needed. Newton-Raphson was used here to solve the iterative equations, since it rapidly converges after a plausible initial guess for equilibrium water temperature. Figure 1 shows the logical steps embedded in the temperature modelling algorithm.

For covered tanks, air temperature under the cover (T'_a) is considered as a second state-variable; assuming an ideal gas and a non-adiabatic heat transfer, the steady-conditions energy balance of the gaseous system under the cover is given by:

$$Q'_{ta} = \rho'_a q_v c_{pa}(T'_a - T_o) \quad (3)$$

where

$q_v = q_a + q_{ev}$ = overall ventilation rate under the cover (m^3 h^{-1})

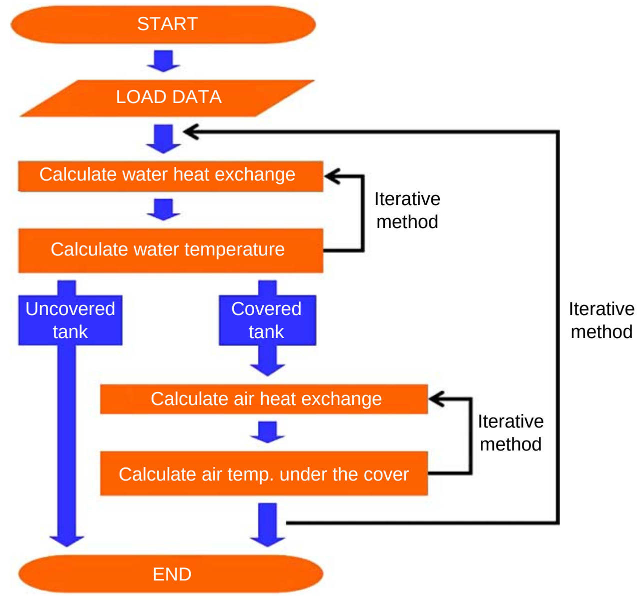

Figure 1 | Temperature modelling algorithm.

with

q_a = air flow rate (for surface aerators $q_a = 0$, m^3 h^{-1})
q_{ev} = external ventilation rate (m^3 h^{-1})
T_o = temperature of the air entering the cover, or

$$T_o = \begin{cases} T_a & \text{for surface aerators} \\ \frac{T_a \cdot q_{ev} + T_w \cdot q_a}{q_{ev} + q_a} & \text{for diffused aeration} \end{cases}$$

ρ'_a = density of air under the cover (kg m^{-3})
c_{pa} = specific heat of air (J kg^{-1}°C^{-1})
T'_a = air temperature under the cover (°C).

The net heat gain or loss Q'_{ta} is the sum of the heat exchanges phenomena that occur under the cover, as follows:

$$Q'_{ta} = Q_{cover} - Q_e - Q_c - Q_a \quad (4)$$

where Q_{cover} = cover conduction heat exchange (W).

Conduction heat exchange terms are described in the detailed model presentation. The model result for the covered tank scenario requires two sequential iterative calculations (Figure 1): first, the water temperature (T_w) is calculated; secondly, the air temperature (T'_a) is calculated. The water temperature is re-calculated with the new air temperature value, and the process repeats until convergence is reached. This model can also be used to calculate heat fluxes in primary clarifiers and equalization tanks.

Heat exchange terms

Surface convection

The method introduced in this paper is based on the dimensionless analysis of convection between a horizontal air stream and a plane water surface:

$$Q_c = h(T_a - T_w)A_s \tag{5}$$

where

$h = Nu\,k\,a^{-1/2}b^{-1/2}$ Heat transfer coefficient ($W\,K^{-1}\,m^{-2}$)
$Nu = (0.037\,Re^{4/5} - 871)Pr^{1/3}$, Nusselt number (dimensionless)
$Re = W\,a^{1/2}b^{1/2}\,v^{-1}$, Reynolds number (dimensionless)
$Pr = c_{pv}\mu k^{-1}$, Prandtl number (dimensionless)

with

k = thermal conductivity ($W\,m^{-1}{}^\circ C^{-1}$)
$a^{-1/2}b^{-1/2}$ = specific length of the tank (m)
μ = dynamic viscosity ($N\,s\,m^{-2}$)
v = kinematic viscosity ($m^2\,s^{-1}$)
c_{pv} = specific heat of saturated water vapour ($J\,kg^{-1}\,{}^\circ C^{-1}$)
As = aeration tank surface area (m^{-2})

Surface evaporation

The method introduced in this paper is based on the dimensionless analysis of water evaporation from a plane water surface, as given by:

$$Q_e = h_m A_s h_{fg}(\rho_{v,Ta} - \rho_{v,Tw}) \tag{6}$$

where

$h_m = Sh D_{AB} a^{-1/2} b^{-1/2}$, Mass transfer coefficient ($m\,s^{-1}$)
$Sh = (0.037\ Re^{4/5} - 871)\ Sc^{1/3}$, Sherwood number (dimensionless)
$Sc = vD_{AB}^{-1}$, Schmidt number (dimensionless)

with

D_{AB} = binary mass diffusion coefficient (A = water vapour, B = air, $m^2\,s^{-1}$)
h_{fg} = latent heat of vaporization ($J\,kg^{-1}$)
$\rho_{v,Ta}$ = water vapour density in air ($kg\,m^{-3}$)
$\rho_{v,Tw}$ = water vapour out stream density ($kg\,m^{-3}$)

Water vapour density and latent heat of vaporization are calculated with empirical functions based on literature data and process parameters.

RESULTS AND DISCUSSION

Model validation

For model validation data sets from our previous publications and other published sources were used (Table 1). Missing data are calculated with the same assumption of previous publications. Measured values are referred to monthly or daily average temperatures. In most cases, data recording accuracy is ±0.6°C considering the precision of Fahrenheit scale field thermometers. The residuals between estimated and measured temperatures are showed in Figure 2. Using the new model the precision is in the range ±0.44°C. Using the steady-state equivalent of Sedory & Stenstrom (1995) the standard deviation is 0.47°C. Both ranges remain below the accuracy level of the measuring equipment.

Design analysis

Figure 3a shows the design analysis performed to evaluate the impact of changing the aeration system or covering the tank on equilibrium temperature and heat fluxes of wwtp #5. Scenario #0 and measured temperatures refer to

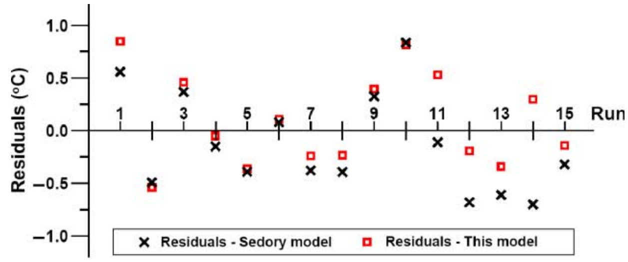

Figure 2 | Comparison of residuals between this model and the model by Sedory & Stenstrom (1995).

Table 1 | Datasets used for model validation

WWTP	Run	Data format	Reference	Treatment	Aeration type	Influent Temp. (°C)	Air Temp. (°C)	Relative humidity (%)	Basin Temp. (°C)	Plant latitude (degrees)	Notes
1	1	Monthly Ave. (January)	Sedory & Stenstrom (1995)	CASP	Fine bubble diffusers	13.51	−2.16	90.5	**12.46**	42	Municipal Uncovered
	2	Monthly Ave. (July)				18.07	23.59	66.9	**19.28**		
2	3	Monthly Ave. (January)	Sedory & Stenstrom (1995)	CASP	Surface aerators	19.80	13.68	57.9	**18.83**	34	Municipal Uncovered
	4	Monthly Ave. (July)				27.40	18.91	78.7	**25.40**		
3	5	Monthly Ave. (January)	Sedory & Stenstrom (1995)	CASP	HPO, mech. mixers	19.25	5.99	10.0	**19.74**	38.5	Industrial Covered
	6	Monthly Ave. (July)				26.42	23.34	10.0	**26.47**		
4	7	Monthly Ave. (January)	Sedory & Stenstrom (1995)	CASP	Fine bubble diffusers	21.50	13.82	58.2	**21.93**	33.5	Municipal Uncovered
	8	Monthly Ave. (July)				27	18.91	78.5	**27.37**		
5	9	Monthly Ave. (January)	Sedory & Stenstrom (1995)	CASP	Fine bubble diffusers	35.43	−4.79	65.14	**32.18**	44.5	Industrial Uncovered
	10	Monthly Ave. (July)				40.41	21.32	64.35	**37.63**		
6	11	Daily Average	Gillot & Vanrolleghem (2003)	Aerated lagoon	Surface aerators	23.30	8.20	79	**10.10**	50	Industrial Uncovered
7	12	Monthly Ave. (January)	Argaman & Adams (1977)	CASP	Surface aerators	25.80	7.40	82	**17.10**	37	Industrial Uncovered
	13	Monthly Ave. (July)				31.00	25.60	74	**28.40**		
8	14	–	Argaman & Adams (1977)	Aerated lagoon	Diffusers	37.80	10.50	70.0	**18.30**	–	Uncovered
9	15	Daily Average	Scherfig *et al.* (1996)	CASP	Diffusers	8.95	6.83	83.3	**8.91**	56	Uncovered

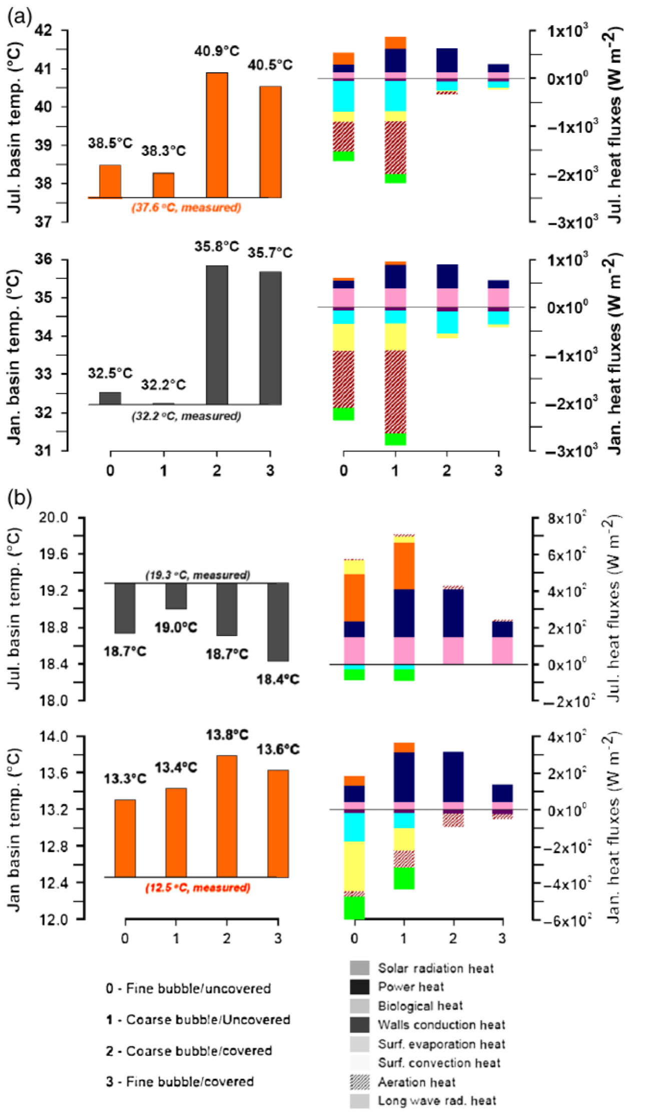

Figure 3 | Design analysis: equilibrium temperature and heat fluxes variation adopting fine or bubble diffusers into covered or uncovered tanks. (a) wwtp #5 January and July; (b) wwtp #1 January and July. Measured temperatures are referred to the actual plants design (scenario #0, uncovered tank with fine bubble diffusers).

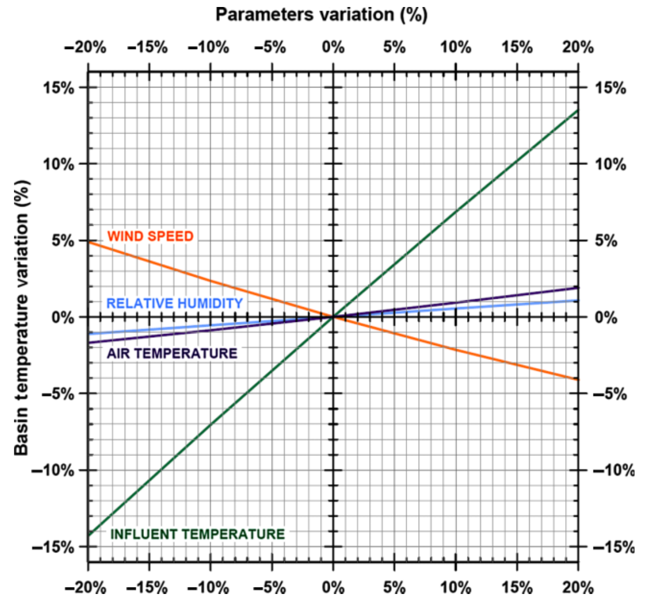

Figure 4 | Dimensionless effect of environmental parameters variation (wind speed, relative humidity, ambient air temperature and influent wastewater temperature) on basin temperature variation for warm climate. WWTP#7, uncovered tank with surface aerators, summer data set (run#13).

real plant layout. Main design assumptions are summarized in Table 2. Covering the tank (Figure 3a, scenario #3) reduces the heat losses in both climate conditions, with a temperature increasing of +3°C in January and +2°C in July. Using coarse bubble diffusers increases the temperature into covered tanks (Figure 3a, scenario #2) while decreases the temperature in uncovered tank layout (Figure 3a, scenario #1)

Analyses conducted on wwtp #1 are reported in Figure 3b. Coarse bubble diffusers increase the equilibrium temperature by +0.2°C; cooling is not promoted because the increased power input is more influential on the overall heat balance than evaporation due to higher air flow rate. Covering the tanks increases the cold climate

Table 2 | Design analysis assumptions for selected aeration tank layouts

Design scenario	Surface layout	Aeration type	Air flow rate	Power input	Off-gas humidity factor
0	Uncovered	Fine bubble diffusers	1x	1x	~1
1	Uncovered	Coarse bubble diffusers	3x	3x	0.9
2	Covered	Coarse bubble diffusers	3x	3x	0.9
3	Covered	Fine bubble diffusers	1x	1x	~1

temperature and decreases the warm climate temperature with a reduction of temperature extremes range.

Sensitivity analysis

The tank equilibrium temperature sensitivity analysis is performed with a single equation that requires climate, process or design parameters percent variations as inputs. Analyses conducted on a plant with surface aeration treating industrial wastewater, which is part of the data sets presented above (run #13), are reported in Figure 4. The nomograph shows the impact of changing several input parameters (wind speed, relative humidity, ambient air temperature and influent wastewater temperature) on the basin equilibrium temperature. In this specific case, increasing wind speed by 20% ($+0.84\,m\,s^{-1}$) reduces the equilibrium temperature by 4% (-0.8°C).

CONCLUSIONS

The presented model refines previous models and includes a second control volume and a second state variable to predict the effect of covering the aeration tanks. For selected heat exchange terms, new equations based on dimensionless analysis are presented. The model validation agrees well with all data sets simulation and the standard deviation of residuals remains below the accuracy level of measurement equipment. A sensitivity analysis and nomograph are presented that allows the impact of small variations in parameters on basin temperature to be estimated.

Using default data provided and specifying input data by the user, this model provides designers with key information on expected temperature extremes in different design scenarios and can support the overall aeration tanks design process. For this reason, and considering the significant numerical improvements, this temperature model should be embedded into modelling-and-design commercial softwares.

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