

DEVELOPMENT OF A MATHEMATICAL MODEL FOR WATERTUBE BOILER HEAT TRANSFER CALCULATIONS

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Abstract

The scientific basis behind the development of a mathematical model for heat transfer calculations of a typical watertube boiler are discussed. The formulae that can be used to calculate gas properties, heat transfer for gas flow across tubes and inside tubes, steam properties, etc. are given. All data are presented in a format that can be readily used for computerising heat transfer calculations. The information is based on an existing proven computerised mathematical model. Some techniques to facilitate the computerisation of boiler heat transfer calculations are described.

Keywords: watertube boiler, heat transfer, heat transfer model

Introduction

During the 1980s, John Thompson Africa developed a computer program to model the thermal design of industrial boiler plant. Apart from the heat transfer theory, the formulae used in the model are also based on several technical papers and have been adapted to reflect the data received from actual site readings. It is not possible to cover the entire mathematical model in this paper. However, the most important data and formulae are reviewed and discussed, allowing engineers to computerise the essential heat transfer relations encountered in modern industrial boiler designs.

Gas temperature at furnace exit

The model uses the combustion heat balance equation whereby the total net heat input into the furnace equates the sum of the heat absorbed by the furnace walls and the heat carried by the gases leaving the furnace. The model uses the ambient air temperature as its reference temperature, therefore the net heat input into the furnace per kilogram of fuel burnt can be written as:

$$H_{in} = \text{heat in fuel burnt} + \text{heat in hot combustion air} \\ = NCV + m_{ha} (c_{ha} t_{ha} - c_a t_a)$$

where

$$H_{in} = \text{net heat input in the furnace in kJ/kg fuel burnt} \\ NCV = \text{net calorific value of the fuel in kJ/kg} \\ m_{ha} = \text{hot combustion air flow in kg/kg fuel burnt} \\ c_{ha} = \text{specific heat of the air at } t_{ha} \text{ in kJ/kg}^\circ\text{C} \\ t_{ha} = \text{temperature of the hot air in } ^\circ\text{C}$$

$$c_a = \text{specific heat of the ambient air in kJ/kg}^\circ\text{C} \\ t_a = \text{ambient air temperature in } ^\circ\text{C}.$$

The heat absorbed by the surrounding walls can be defined by the corrected Stefan-Boltzmann equation and thus the following equation can be written:

$$H_{out} = \text{heat absorbed by the furnace walls} + \text{heat carried by the gases leaving the furnace}$$

$$= \frac{20.53 \cdot Fr \cdot BBSA}{m_f} \left[\left(\frac{T_e}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] + m_g (c_e t_e - c_a t_a)$$

where

$$20,53 = \text{Stefan-Boltzmann constant in kJ/h m}^2 \text{ } ^\circ\text{K}^4 \\ F_r = \text{correction factor for geometry and emissivity (dimensionless); typical values are given in Table 1} \\ BBSA = \text{total effective black body surface area of the furnace in m}^2 \\ m_f = \text{amount of fuel burnt in kg/h} \\ T_e = \text{absolute temperature of the exit gas in } ^\circ\text{K} \\ T_w = \text{absolute temperature of the furnace walls in } ^\circ\text{K} \\ m_g = \text{mass of flue gas produced per kg fuel burnt in kg/kg} \\ c_e = \text{specific heat of the gas at temperature } t_e \text{ in kJ/kg}^\circ\text{C} \\ t_e = \text{exit gas temperature in } ^\circ\text{C}.$$

Formulae to determine the specific heat of air and flue gas are given in a later section.

Table 1
Typical correction factors for furnace geometry and emissivity.

Fuel	Correction factor
Bituminous coal	0,81
Bagasse	0,72
Wood	0,72
Oil	0,85
Gas	0,65

The above equation cannot be solved explicitly and recourse has to be made to an iterative technique. The model uses two linear relations to find the solution. The calculation procedure is described in step form and is illustrated in Figure 1.

Step 1: H_{in} is calculated.

Step 2: The objective is to determine t_e , that is, t_1 which results in $H_{out} = H_{in}$. Hence, t_2 is supplied as the initial guess. It is suggested that 950°C be used. From this value of $t_2, H_2 = H_{out}(t_2)$ can be calculated.

Step 3: Adjust the assumed value of t_2 by 50° from the relationships:

$$t_3 = t_2 + 50 \text{ if } H_2 < H_{in}$$

$$t_3 = t_2 - 50 \text{ if } H_2 > H_{in}$$

Step 4: Determine the value $H_3 = H_{out}(t_3)$.

Determine the new estimate t_4 from the linear relationship given by the equation:

$$t_4 = t_2 + (t_3 - t_2) \cdot (H_{in} - H_2) / (H_3 - H_2)$$

From the value of t_4 determine $H_4 = H_{out}(t_4)$.

Step 5: Adjust t_4 by 5°C in accordance with the following relationships:

$$t_5 = t_4 + 5 \text{ if } H_4 < H_{in}$$

$$t_5 = t_4 - 5 \text{ if } H_4 > H_{in}$$

From the value of t_5 determine $H_5 = H_{out}(t_5)$.

Step 6: Calculate the new value for the exhaust gas temperature entitled t_6 from the linear relationship below:

$$t_6 = t_4 + (t_5 - t_4) \cdot (H_{in} - H_4) / (H_5 - H_4)$$

From the value determined for t_6 determine $H_6 = H_{out}(t_6)$.

Step 7: Establish that the value determined for the exhaust gas temperature is within 0,5°C of the correct value by the relationship given below:

$$\text{If } |H_6 - H_{in}| < |(H_5 - H_4) / 10|$$

then t_6 is within 0,5°C of the correct value.

If step 7 is unsuccessful, then t_6 is taken as the new initial guess and the procedure is repeated from Step 2. The method results in a fast convergence to the correct value.

Flue gas component properties

General

The thermal conductivity, dynamic viscosity and mean specific heat for the components found in the flue gas can be determined from:

$$\text{Property} = A + Bt + Ct^2 + Dt^3$$

where t = gas temperature in °C.

The values of A, B, C and D are given in Table 2. They have been derived by curve fitting the values given in the relevant VDI tables and are valid for gas temperatures of up to 1 300°C.

The maximum deviation over the temperature field considered is less than 1%.

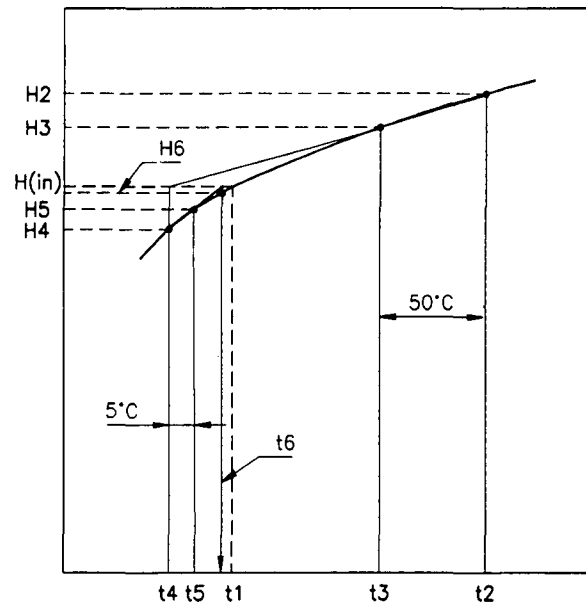


Figure 1. Evaluation of furnace exit temperature.

Table 2
Data for flue gas component properties.

Component	A	B	C	D
Thermal conductivity, mW/m°C				
CO ₂	14,146	0,084035	-1,8718x10 ⁻⁵	4,17281x10 ⁻⁹
N ₂	24,6665	0,063338	-1,8259x10 ⁻⁵	3,92097x10 ⁻⁹
O ₂	24,626	0,072635	-1,3432x10 ⁻⁵	2,46825x10 ⁻⁹
H ₂ O	15,4632	0,087818	2,68456x10 ⁻⁵	-1,28258x10 ⁻⁸
H ₂	169,9763	0,426419	-1,50978x10 ⁻⁴	9,17918x10 ⁻⁸
Air	24,53075	0,070718	-1,51675x10 ⁻⁵	-1,26061x10 ⁻⁹
SO ₂	14,146	0,084035	-1,8718x10 ⁻⁵	4,17281x10 ⁻⁹
Dynamic viscosity, kg/ms x 10 ⁶				
CO ₂	13,7639	0,049807	-2,10096x10 ⁻⁵	5,69835x10 ⁻⁹
N ₂	16,8846	0,044049	-1,8591x10 ⁻⁵	5,8883x10 ⁻⁹
O ₂	19,269	0,054237	-2,41903x10 ⁻⁵	7,80475x10 ⁻⁹
H ₂ O	7,5245	0,044734	-6,74737x10 ⁻⁶	2,9352x10 ⁻⁹
H ₂	8,5577	0,018133	-3,14966x10 ⁻⁶	5,64858x10 ⁻¹⁰
Air	17,4244	0,046762	-2,06231x10 ⁻⁵	6,70797x10 ⁻⁹
SO ₂	13,7639	0,049807	-2,10096x10 ⁻⁵	5,69835x10 ⁻⁹
Mean specific heat, kJ/kg°C				
CO ₂	0,819372	5,24968x10 ⁻⁴	-2,7566x10 ⁻⁷	6,1792x10 ⁻¹¹
N ₂	1,03768	1,410147x10 ⁻⁶	1,20329x10 ⁻⁷	-4,59217x10 ⁻¹¹
O ₂	0,908963	1,42472x10 ⁻⁴	-2,01556x10 ⁻⁹	-1,39593x10 ⁻¹¹
H ₂ O	1,85076	1,32829x10 ⁻⁴	2,39327x10 ⁻⁷	-9,0209x10 ⁻¹¹
H ₂	14,29387	6,5992x10 ⁻⁴	-4,76564x10 ⁻⁷	3,01067x10 ⁻¹⁰
Air	1,00269	3,4628x10 ⁻⁵	8,94269x10 ⁻⁸	-3,63247x10 ⁻¹¹
SO ₂	0,594236	3,2344x10 ⁻⁴	-1,5249x10 ⁻⁷	2,7831x10 ⁻¹¹

Specific heat of flue gas

The specific heat of the flue gas is determined from the summation of the products of the mass fraction of each component by its respective specific heat at the relevant temperature, or:

$$c_p = \sum_{(k)} m_k \times c_k$$

with:

- c_p = mean specific heat of the flue gas in kJ/kg°C
 m_k = mass fraction of the flue gas component in kg/kg
 c_k = mean specific heat of the flue gas component in kJ/kg°C.

Thermal conductivity of flue gas

The thermal conductivity of the flue gas can be determined by the formula:

$$k_g = (\sum_i x_i r_i) \times Z \times 1,085$$

with:

- k_g = thermal conductivity of the flue gas in W/m°C
 k_i = thermal conductivity of the flue gas component in W/m°C
 r_i = volumetric fraction of the component in m³/m³
 Z = $1 + 0,402 y^{0,511} (1 - y)$

with:

- y = volumetric fraction of H₂O in m³/m³.

The volumetric fraction of the component is determined from the mass fraction of the component, from:

$$r_i = m_i \times R_i / R_g$$

where

- r_i = volumetric fraction of the component in m³/m³
 m_i = mass fraction of the component in kg/kg
 R_i = gas constant of the component in Nm/kg °C
 $R_{CO_2} = 188$ $R_{O_2} = 260$ $R_{H_2} = 4120$ $R_{SO_2} = 130$
 $R_{N_2} = 297$ $R_{H_2O} = 462$ $R_{Air} = 287$
 R_g = gas constant of flue gas in Nm/kg°C = $\sum_{(k)} m_k \times R_k$ (k)

Dynamic viscosity of flue gas

The dynamic viscosity of the flue gas can be determined from:

$$\mu_g = \frac{\sum_k \mu_k \cdot m_k / \sqrt{M_k}}{\sum_k m_k / \sqrt{M_k}}$$

with:

- μ_g = dynamic viscosity of flue gas in kg/ms
 μ_k = dynamic viscosity of flue gas component in kg/ms

- m_k = mass fraction of flue gas component in kg/kg
 M_k = molecular mass of flue gas component in kg/kmol (see Table 3).

Table 3
Molecular masses (kg/kmol) of the most common flue gas components.

CO ₂ = 44,01	N ₂ = 28,16	H ₂ O = 18,02
O ₂ = 32,00	H ₂ = 2,02	SO ₂ = 64,06
Air = 28,96		

Thermal conductivity, specific heat and dynamic viscosity of steam

These steam properties are used in the superheater calculations, and can be calculated with the following equation:

$$\text{Property} = A + Bt + Ct^2 + Dt^3$$

with:

t = temperature in °C.

Table 4 gives the values of A, B, C and D for each respective property. They were derived by least square approximation of the steam property values tabulated in several reference works. The formula results in the following property units:

- thermal conductivity k in W/m°C x 10⁻³
- specific heat C in kJ/kg °C
- dynamic viscosity μ in kg/ms x 10⁻⁶.

The approximation gives values within one per cent of the values tabulated in most of the relevant literature for steam temperatures of up to 800°C. The specific heat and dynamic viscosity are independent of pressure, but the value of the thermal conductivity increases with increasing pressure.

The values given in Table 4 yield properties at atmospheric pressure. The value of the thermal conduction can be adjusted for pressure by increasing it with 1% for every 200 kPa above atmospheric pressure. However, for normal sugar industry applications (pressure less than 5 000 kPa), the difference in overall heat transfer coefficient in a superheater calculation with pressure adjustment for thermal conductivity is less than 1%. (The data are not valid for wet steam.)

Table 4
Steam property constants.

	A	B	C	D
k	15,4601747	0,0852373851	4,19974607 x 10 ⁻⁵	-3,18960693 x 10 ⁻⁸
c	1,84473235	3,84201681 x 10 ⁻⁴	5,02265908 x 10 ⁻⁷	-2,68518607 x 10 ⁻¹⁰
μ	7,83915182	0,0426546666	-2,19229917 x 10 ⁻⁶	-6,80037084 x 10 ⁻¹⁰

Saturated steam pressure and temperature

The saturated steam pressure for $t^{\circ}\text{C}$ can be calculated from:

$$P = 22\,120 e^z$$

and

$$z = \frac{\sum_{i=1}^5 A[i] X^i}{(1-X)(1+4,16711732X+20,9750676 X^2)} - \frac{X}{X^2 10^9 + 6}$$

with:

$$\begin{aligned} P &= \text{saturated steam pressure in kPa absolute} \\ X &= \frac{374,15 - t}{647,3} \end{aligned}$$

The matrix $A[i]$ is defined as:

$$A[i..5] = (-7,691234564; -26,08023696; -168,1706546; 64,23285504; -118,9646225).$$

The above calculation is also used to determine the saturated steam temperature for a given steam pressure of P kPa absolute, in the following manner:

An arbitrary low value is assumed for the steam temperature (e.g. 10°C) and for the adjustment value (e.g. 50°C). The saturated steam pressure is then calculated using the formula given above, and compared with the given pressure P . If the calculated saturated steam pressure is smaller than pressure P , then the assumed steam temperature gets augmented by the adjustment value, otherwise the adjustment value is halved and then subtracted from the assumed steam temperature. The saturated steam pressure is again calculated and the whole process is repeated until the calculated and given saturated steam pressures have converged to within a preset value (typical within 0,5 kPa). By halving the adjustment value every time the calculated pressure exceeds the given pressure, conversion is obtained very quickly.

Non-luminous radiative coefficient

The presence of water vapour and carbon dioxide in the flue gas is important in boiler calculations. They are selective radiators that both emit and absorb radiation in certain wavelengths that are not visible, hence the term non-luminous. The non-luminous radiation is dependent on gas temperature, the partial pressure of the gaseous constituents and the geometry.

The non-luminous coefficient h_r can be calculated as shown below. The method is based on the article by Sharan (1963).

$$h_r = h_{r300} \times K_t \times K_E$$

with:

$$h_{r,300} = 4,1868 P_x \left[\frac{t_x + 71,5 - 28,5 \frac{\ln(x/0,015625)}{\ln 2}}{1271,5 - 28,5 \frac{\ln(x/0,015625)}{\ln 2}} \right] \text{kJ/m}^2 \text{h}_C$$

$$h_r = \text{radiation coefficient in kJ/m}^2 \text{h } ^{\circ}\text{C}$$

with:

$$t_g = \text{gas temperature in } ^{\circ}\text{C}$$

$$x = \text{MBL} \times [P_H (P_C + P_H)]^{0,5}$$

$$\text{MBL} = \text{mean beam length in m}$$

$$P_H = \text{partial pressure of H}_2\text{O}$$

$$P_C = \text{partial pressure of CO}_2$$

$$P_X = a_0 + a_1 x + a_2 x^2 + a_3 x^3 + a_4 x^4 + a_5 x^5$$

with:

$$a_0 = 10,248755 \quad a_1 = 259,0015$$

$$a_2 = -521,4298 \quad a_3 = 583,8360$$

$$a_4 = -303,8581 \quad a_5 = 58,109$$

$$K_t = \text{temperature correction factor}$$

$$= K_{t1} = \left[\frac{T_g^t - T_w^t}{T_g^t - 573^t} \right] \times \left[\frac{t_g - 300}{t_g - t_w} \right] \text{ for wall temp. } < 300_C$$

$$= (0,73 \times K_{t1} - 0,719) \times \frac{(1100)^{0,25}}{(t_x)} + 1 \text{ for wall temps. between } 300 \text{ and } 600_C$$

$$T_g = \text{log mean gas temperature in } ^{\circ}\text{K}$$

$$T_w = \text{log mean wall temperature in } ^{\circ}\text{K}$$

$$K_E = \text{wall emissivity correction factor (taken = 1).}$$

For the boiler model calculation, the total gas pressure can be considered to be atmospheric. Hence, the partial pressures of the H_2O and CO_2 components can be taken equal to their volumetric fractions.

Internal heat transfer coefficient

The internal heat transfer coefficient for turbulent gas flow inside a tube can be determined from the Dittus and Boelter equation:

$$N_u = 0,023 R_e^{0,8} P_r^{0,4}$$

or, after introducing the relevant definitions and solving for h_i :

$$h_i = 0,023 d^{-0,2} k^{0,6} G^{0,8} c^{0,4} \mu^{-0,4}$$

where:

$$h_i = \text{internal heat transfer coefficient in W/m}^2\text{ } ^{\circ}\text{C}$$

$$d = \text{inside diameter of the tube in m}$$

$$k = \text{thermal conductivity in W/m}^{\circ}\text{C}$$

$$G = \text{gas mass flux in kg/m}^2\text{s}$$

$$c = \text{specific heat in J/kg}^{\circ}\text{C}$$

$$\mu = \text{dynamic viscosity in kg/m s}$$

$$k = \text{thermal conductivity in W/m}^{\circ}\text{C}.$$

The equation is used for calculating the internal heat transfer coefficient on a superheater and for calculating the internal heat coefficient on the gas side of the air heater. Properties of steam and flue gas are determined at the log mean temperatures.

Convective coefficient for cross flow

The convective coefficient for gas flow across a bundle of tubes can be determined from:

$$N_u = K \cdot R_e^{0,6} \cdot P_r^{0,33}$$

or:

$$h_c = K \cdot D^{-0,4} \cdot G^{0,6} \cdot c^{0,33} \cdot \mu^{-0,27} \cdot k^{0,67}$$

with:

- h_c = convective coefficient in $W/m^2 \text{ } ^\circ C$
- K = 0,287 for in-line tube arrangement
= 0,32 for staggered tube arrangement
- D = outside diameter of tube in m
- G = flue gas mass flux in kg/m^2s
- c = specific heat of flue gas in $J/kg \text{ } ^\circ C$
- μ = dynamic viscosity of flue gas in kg/ms
- k = thermal conductivity of flue gas in $W/m \text{ } ^\circ C$.

The properties of the flue gas are in this case determined for the film temperature, defined as follows:

$$t_f = \frac{t_w + t_g}{2}$$

with:

- t_f = film temperature in $^\circ C$
- t_w = log mean wall temperature in $^\circ C$
- t_g = log mean gas temperature in $^\circ C$.

The formula is valid for turbulent flow with a Reynolds number greater than 2 000 and for tube banks at least 10 rows deep.

An arrangement factor has to be applied to the result to adjust for the geometry of the bank. This factor is dependent on the ratio between longitudinal, as well as transversal pitching and the diameter of the tube. For normal industrial boiler constructions, this factor is around 0,98 to 1,0.

Adjustment factors for the tube bundles less than 10 rows deep and arrangement factors can be found in the specialised literature.

Heat transferred to the cavity walls

The heat transferred to the cavity walls is based on the relationships for turbulent heat transfer to planar surfaces and for gases flowing along the longitudinal axis of tubes. The characteristic dimension becomes the hydraulic diameter which is defined as four times the ratio of the volume to the peripheral surface area. The following relationship has been used for obtaining the convective coefficient of heat transfer to the cavity walls:

$$N_u = 0,023 P_r^{0,4} R_e^{0,8}$$

Under most gas conditions associated with the boiler design, little accuracy will be lost by assuming the Prandtl number to be 0,6. By making this assumption and adapting the equation for use with more familiar units, the equation becomes:

$$h_c = 0,161 D_h^{-0,2} G^{0,8} \mu^{0,2} c_p$$

where:

- D_h = hydraulic diameter in m
- G = gas mass flux in kg/hm^2
- μ = dynamic viscosity in kg/ms
- c_p = specific heat in $kJ/kg^\circ C$
- h_c = convective coefficient in $kJ/h \text{ m}^2 \text{ } ^\circ C$.

Iterative calculations

In general, the heat transferred throughout the boiler is based on:

$$\text{heat lost by gas} = \text{heat transferred to heating surfaces}$$

or:

$$M_g (c_i t_i - c_o t_o) = U A \text{ LMTD}$$

where:

- M_g = mass of gas in kg/h
- c_i = specific heat of gas at inlet in $kJ/kg \text{ } ^\circ C$
- t_i = gas inlet temperature in $^\circ C$
- c_o = specific heat of gas at outlet in $kJ/kg \text{ } ^\circ C$
- t_o = gas outlet temperature in $^\circ C$
- U = overall heat transfer coefficient in $kJ/hm^2 \text{ } ^\circ C$
- A = total heating surface in m^2
- LMTD = log mean temperature differential between gas and heating surface in $^\circ C$.

The overall heat transfer coefficient is determined from:

$$U = 1 / (1 / h_o f_o + t / k + d_o / h_i d_i f_i)$$

or, since $k / t \ll h_o f_o$:

$$U = 1 / (1 / h_o t_o + d_o / h_i d_i f_i)$$

where:

- $h_o = h_r + h_c$
- and
- h_r = radiative coefficient in $kJ/hm^2 \text{ } ^\circ C$
- h_c = convective coefficient in $kJ/hm^2 \text{ } ^\circ C$
- f_o = external fouling factor (dimensionless)
- d_o = external tube diameter
- d_i = internal tube diameter
- h_i = internal heat transfer coefficient in $kJ/hm^2 \text{ } ^\circ C$
- f_i = internal fouling factor (dimensionless).

The heat transfer coefficients can be determined by using the methods described above. However, both the left and right hand sides of the heat transfer equations contain temperature dependent values and therefore a trial and error technique is used to solve the equation. The model uses an approach technique as described below and illustrated in Figure 2.

Step 1: Assume a gas exit temperature t_1 in °C.

Step 2: With t_1 calculate the heat lost by the gas Q_{1a} (left hand side of the equation) and the heat transferred to the heating surface Q_{1b} (right hand side of the equation). If $Q_{1a} > Q_{1b}$, then assume $t_2 = t_1 - D_t$, else assume $t_2 = t_1 + D_t$. The temperature difference D_t can be any value but is typically around 10°C.

Step 3: With t_2 calculate the heat lost by the gas Q_{2a} (left hand side of the equation) and the heat transferred to the heating surface Q_{2b} (right hand side of the equation).

Step 4: Calculate $t_3 = (b_2 - b_1) / (a_1 - a_2)$

where:

$$a_1 = (Q_{1a} - Q_{2a}) / (t_1 - t_2)$$

$$b_1 = Q_{1b} - a_1 t_1$$

$$a_2 = (Q_{1b} - Q_{2b}) / (t_1 - t_2)$$

$$b_2 = Q_{1b} - a_2 t_1$$

The value t_3 determined in Step 4 will be converging towards the to-be-determined temperature.

The above steps can be repeated with the new assumed exit temperature $t_1 = t_3$ until the difference between t_1 and t_3 becomes acceptably small. A sophistication on the above method is to change the value D_t depending on the difference between t_3 and t_1 .

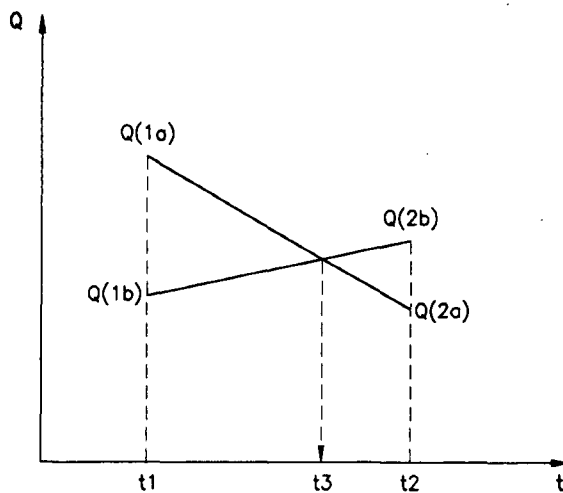


Figure 2. Converging method for iterative calculations.

Conclusions

The methods described have proven to be reliable in their application to industrial boilers and, more specifically, the industrial boilers normally found in the sugar industry. Using the approximations to determine gas properties and heat transfer coefficients in computer programs avoids the need for look-up tables and/or curves. The approximations also facilitate calculations, in particular those of an iterative nature.

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