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# Optimum Distributions of Heating Surface Areas in Industrial Boiler

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**Abstract**

Industrial boiler consists of a furnace and a set of heat exchangers, which are evaporator, superheater, boiler bank, economizer, and air heater. Combustion of bagasse and air in the furnace produces thermal energy that is used to evaporate feed water and increase steam temperature in the superheater. Hot flue gas from the furnace also contributes to increasing steam temperature in the superheater. Hot flue gas then passes through the boiler bank, resulting in additional water evaporation. After that, it passes successively through the economizer, in which feed water temperature is increased, and the air heater, in which air temperature is increased, before being exhausted to the atmosphere. It is known qualitatively that the performance of the boiler is improved with the increase of the heating surface area of each component. Qualitative effects of increasing surface areas, however, require a boiler model that takes into account their effects. The unavailability of a suitable model in the open literature is the main reason why this paper is written. The proposed model assumes that superheater is divided into radiative superheater and convective superheater. Radiative superheater receives direct radiation from the furnace. Convective superheater, boiler bank, economizer, and air heater are convective heat exchangers. The model is used to determine the optimum distribution of the surface areas of superheater, economizer, and air heater that results in the minimum installation cost for a specified boiler efficiency and the optimum distribution of the surface areas that results in the maximum boiler efficiency for a specified installation cost.

**Keywords:** Boiler, Heat exchanger, Energy system, Optimization

## I. INTRODUCTION

Boiler is an important component in engineering systems such as thermal power plants and cogeneration systems. The function of a boiler is to produce superheated steam that is expanded in a steam turbine for power production. Analysis of such a system requires a boiler model. A widely used model is a black-box model, which disregards the components of the boiler. Such a model considers only the inputs to the boiler, which are feed water, fuel, and are, and the outputs, which are superheated steam, flue gas, and ash [1]. A black-box model can be used to show the effects of flue gas temperature, excess air ratio, feed water temperature, fuel moisture content, and fuel heating value on the boiler efficiency, defined as the enthalpy increase of feed water as it turns into superheated steam divided by the heating value of the fuel consumed by the boiler. Using this boiler model in an energy system analysis requires a given value of boiler efficiency or a correlation between the boiler efficiency and input parameters [2].

A boiler consists of a furnace and set of heat exchangers. The boiler efficiency is undoubtedly affected by the heating surface areas of these heat exchangers. Although the effects of heating surface areas on boiler efficiency can be more or less qualitatively predicted, their quantitative effects have to be determined from a boiler model that is more sophisticated than black-box models. Examples of these models are available in the literature. Kakac [3] provided the detail of the model of each component of a fossil fuel boiler, along with sample calculations. Diez et al. [4] developed a model of utility boiler for on-line simulation. Cantrell and Idem [5] modelled a boiler as a set of heat exchangers, and used the effectiveness-NTU method to analyse its performance under fouling conditions. Moghari et al. [6] used the model of natural gas-fired boiler to investigate heat transfer behaviour within the boiler. Recently, Hajebzadeh et al. [7] proposed a detailed model of a coal-fired boiler, validated it with measured data, and showed that the model can predict the behaviour of a boiler operating at partial loads.

Previous models have been developed for coal-fired boilers. In this paper, these models are modified in order to develop a model of industrial boiler. The difference between industrial boilers and coal-fired boilers is that bagasse boilers do not have reheater because the temperature of flue gas in bagasse boilers is significantly lower than that in coal-fired boilers. Furthermore, boiler bank, which is not found in coal-fired boilers, is installed after superheater in industrial boilers to produce additional saturated steam. A model of industrial boiler can be found in a book by Stultz and Kitto [8]. However, their model requires interpolations using proprietary charts, and is not suitable for computerized simulation. Therefore, an alternative industrial boiler model is developed for simulation purpose in this paper. The following sections give the description of industrial boiler, and present mathematical models of the main components of the boiler. The proposed boiler model is then used to determine the optimum distributions of heating surface areas in a reference boiler

## II. COMPONENTS OF INDUSTRIAL BOILER

An illustration of bagasse boiler system is shown in Fig. 1. The main components of the boiler are furnace (F), evaporator (EV), superheater (SH), boiler bank (BB), economizer (EC), and air heater (AH). Solid lines denote fuel, air, and flue gas, whereas and dashed lines denote feed water and steam. Bagasse and heated air are fed to F where combustion occurs. Some of the thermal energy from combustion is transferred to EV, which consists of water tubes forming part of the furnace walls, causing partial evaporation of pressurized feed water. Flue gas from F flows successively past SH, BB, EC, and AH. Heat transfer from flue gas to steam in SH results in the increase of the temperature of saturated steam in SH so that the output of SH is superheated steam. Heat transfer from flue gas to feed water in BB causes partial evaporation, which results in the production of additional saturated steam. Heat transfer from flue gas to feed water in EC increases the temperature of feed water before it enters the steam drum (not shown in Fig. 1). Heat transfer from flue gases to air in AH increases the air temperature before it enters F. After leaving AH, flue gas is exhausted to the atmosphere.

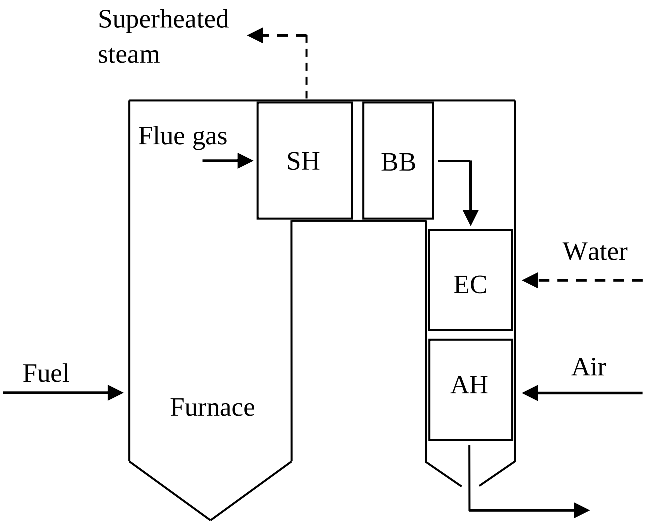


**Fig. 1.** Main components of industrial boiler

Material flows in the industrial boiler system are shown in Fig. 2. Flue gas leaves F at temperature $T_{g1}$. It is used to increase steam temperature in SH from the saturated steam temperature ($T_v$) to $T_s$. It should be noted that SH also receives thermal energy by direct radiation from hot gas in F. Flue gas leaving SH at $T_{g2}$ flows to BB, EC, and AH, and its temperature is reduced, respectively, to $T_{g3}$, $T_{g4}$, and $T_{g5}$. Heat transfer from flue gas causes the increase of feed water temperature from $T_{wi}$ to $T_{we}$ in EC and the increase of air temperature from $T_{ai}$ to $T_{ae}$ in AH. Steam drum (D) receives subcooled feed water at a mass flow rate $m_w$ from EC, and saturated steam at mass flow rates $m_{s1}$ and $m_{s2}$ from EV and BB. It returns saturated liquid water at the same mass flow rates to EV and BB, and sends saturated steam at a mass flow rate $m_s$ to SH. Water evaporation in EV is due to radiative heat transfer from flue gas in F, whereas water evaporation in BB is due to convective heat transfer from flue gas. In order to maintain the concentration of dissolved solids in feed water at a safe level, it is assumed that some of the feed water is blowdown water. It should be noted that, in an actual operation, the inputs to D from EV and BB are mixtures of saturated steam and saturated liquid water, which are separated in D. In other words, some saturated liquid water is recirculated through D. In this simplified model, the recirculated saturated liquid water is ignored, and inputs to D from EV and BB are assumed to be saturated steam.

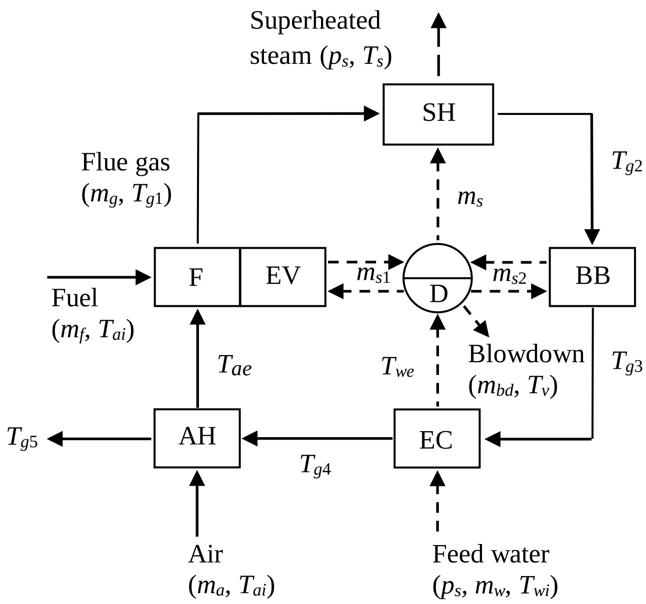


**Fig. 2.** Material flows in industrial boiler system

## III. MATHEMATICAL MODEL

The model of bagasse boiler consists of models of its components. These components can be modelled with varying levels of complexity. The approach taken in this paper is to use a simplified model that is sufficient to capture the essential characteristics of the component.

### III.I Furnace

Combustion between fuel and air occurs in the furnace, along with radiative and convective heat transfer processes. It is necessary to employ CFD simulation for design and optimization of furnace [9]. But if the furnace is viewed as a boiler component that supplies thermal energy to the boiler, a simplified model, in which the temperature of flue gases is assumed to be uniform, is sufficient. This approach has been taken by many researchers in analysing energy conversion systems. For example, Hajebzadeh et al. [7] and Zhang et al. [10] used this approach in analysing coal-fired boilers.

An objective of a furnace analysis is to determine the furnace exit gas temperature because flue gases at the furnace exit provide thermal energy to all heat exchangers in the boiler. It may be determined by using a method suggested by Blokh [3],

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which gives a relation between the furnace exit gas temperature and the adiabatic flame temperature. Chandok et al. [11] proposed a method of estimating furnace exit gas temperature using artificial neural network. Alternatively, it can be determined from energy balance in the furnace, which is expressed as

$$m_f LHV + (1 - x_A) m_f c_{pf} (T_{ai} - T_r) + m_a c_{pa} (T_{ae} - T_r) = \\ m_g c_{pg} (T_{g1} - T_r) + m_f c_{pash} (T_{g1} - T_{ai}) + Q_{loss} + Q_{EV} + Q_{SH} \quad (1)$$

where $LHV$ is the lower heating value of fuel, $x_A$ is the mass fraction of ash in fuel, $T_r$ is the reference temperature (25°C), $Q_{loss}$ accounts for the heat transfer due to radiation and convection between the boiler shell and the ambient air, and $c_{pf}$, $c_{pa}$, $c_{pg}$, and $c_{pash}$ are, respectively, specific heat capacities of fuel, air, flue gas, and ash. $Q_{EV}$ and $Q_{SH}$ in Eq. (1) are radiation heat transfer to the evaporator and the superheater, respectively.

Assume that the composition of bagasse is known. The higher heating value ($HHV$) can be determined from the formula proposed by Qian et al. [12].

$$HHV = 8.7352 \times 10^4 \left( \frac{1}{3} x_C + x_H + \frac{1}{8} x_S \right) \quad (2)$$

where $x_C$, $x_H$, and $x_S$ are, respectively, mass fractions in bagasse of carbon (C), hydrogen (H), and sulfur (S). In order to determine the lower heating value of bagasse, the amount of water resulting from the combustion of 1 kg of bagasse must be known. If the moisture content of bagasse is $x_M$, the complete combustion of 1 kg of bagasse produces $9x_H + x_M$ kg of water. Therefore, the lower heating value of bagasse is

$$LHV = HHV - (9x_H + x_M) \Delta i_r \quad (3)$$

where $\Delta i_r$ is the latent heat of water evaporation at the reference state ($2.442 \times 10^3$ kJ/kg).

The amount of excess air required for the complete combustion of fuel is $\phi$. It is assumed that $\phi = 0.3$. Once the excess air is known, the mass flow rates of air ($m_a$) and flue gas ($m_g$) can be determined from

$$m_a = (1 + \phi) AFR \, m_f \quad (4)$$

$$m_a = (1 - x_A) m_f + m_a \quad (5)$$

where $AFR$ is the stoichiometric air-fuel ratio.

$$AFR = 11.44 x_C + 34.32 x_H + 4.29 (x_S - x_O) \quad (6)$$

The furnace is modelled as a rectangular enclosure. Part of the vertical walls is installed with evaporator tubes. The remaining part is assumed to be insulated. Heat transfer in the furnace is predominantly radiative. Radiation exchange occurs between hot gases, which is the product of combustion, and the evaporator tubes. Radiation exchange also occurs between hot gases and the imaginary exit plane at the top surface. All of the radiation heat transfer through this exit plane reaches the superheater. This model suggests that the net radiation heat transfer from the hot gas to the evaporator is

$$Q_{EV} = \sigma A_v \left( \varepsilon_g T_{g1}^4 - \alpha_g T_{EV}^4 \right) \quad (7)$$

where $\varepsilon_g$ is the emissivity, and $\alpha_g$ is the absorptivity of the gases, which are determined from correlations given by Leckner [13]. $A_v$ is the effective vertical surface area of the enclosure, and $T_{EV}$ is the temperature of the vertical surfaces, which is assumed to be equal to $T_v + 5$. Similarly, the net radiation heat transfer from the hot gases to the superheater is

$$Q_{SH} = \sigma A_t \left( \varepsilon_g T_{g1}^4 - \alpha_g T_{SH}^4 \right) \quad (8)$$

where $A_t$ is the effective area of the exit plane, and $T_{SH}$ is the temperature of the superheater tubes.

### III.II Steam drum

The supply of subcooled feed water at temperature $T_{we}$ to the steam drum, as shown in Fig. 2, results in condensation of some of saturated steam sent to the steam drum from the evaporator and the boiler bank. At steady state, the amount of condensed steam ($\Delta m_s$) is determined from energy balance.

$$\Delta m_s = \frac{m_w c_{pw} (T_v - T_{we})}{\Delta i_s} \quad (9)$$

where $\Delta i_s$ is the latent heat of water evaporation at boiler pressure $p_s$, and $c_{pw}$ is the specific heat capacity of water. This means that the amount saturated water that must be evaporated in the evaporator and the boiler bank is

$$m_{s1} + m_{s2} = m_s + \frac{m_w c_{pw} (T_v - T_{we})}{\Delta i_s} \quad (10)$$

Even though feed water contains a very small fraction of dissolved solids, the operation of the steam drum will increase the concentration of dissolved solids if there is no blowdown. It is assumed the mass flow rate of blowdown that is required to maintain the concentration of dissolved solids at a safe value is 2% of steam flow rate ($m_s$). Therefore, $m_w = 1.02 m_s$.

### III.III Evaporator

Evaporator tubes are located at the walls of furnace. Radiative heat transfer from furnace causes partial evaporation of pressurized feed water in evaporator. The mass flow rate of evaporated water is determined from

$$Q_{EV} = m_{s1} \Delta i_{vl} \quad (11)$$

The rate of steam generation in the evaporator ($m_{s1}$) can be expressed in terms of the total rate of steam generation ($m_s$) using Eq. (10).

### III.IV Superheater

The superheater consists of tube bundles. Steam flows inside the tubes, and flue gas flow outside. It may be assumed that the tube thickness is negligible. Let the internal heat transfer coefficient between steam and tube walls of the superheater be $h_{SH,i}$, and the external heat transfer coefficient between tube walls of the superheater and flue gas be $h_{SH,o}$. The overall heat transfer coefficient is determined from

$$\frac{1}{U_{SH}}=\frac{1}{h_{SH,i}}+\frac{1}{h_{SH,o}} \tag{12}$$

Superheater is a mixed convective-radiative heat exchanger. Previous investigations by Taler et al. [14] and Sobota [15] have assumed that radiation heat transfer can be taken into account by including radiative heat transfer coefficient in $h_{SH,o}$. However, it has been shown by Kakac [3] that there is also direct radiative heat transfer from hot gases in the furnace to the superheater. Diez et al. [4] proposed a model of counter-flow heat exchanger that accounts for direct radiative heat transfer. This model is adopted as the model of superheater in this paper. It yields the following equations.

$$\ln\left(\frac{T_{g2}-T_v+k}{T_{g1}-T_s+k}\right)=U_{SH}A_{SH}\left(\frac{1}{m_s c_{pv}}-\frac{1}{m_g c_{pg}}\right) \tag{13}$$

$$k=\frac{Q_{SH}}{U_{SH}A_{SH}}\left[\frac{1}{m_g c_{pg}/m_s c_{pv}-1}+\frac{1}{h_{SH,o}/h_{SH,i}+1}\right] \tag{14}$$

$$m_s c_{pv}(T_s-T_v)=m_g c_{pg}(T_{g1}-T_{g2})+Q_{SH} \tag{15}$$

where $c_{pv}$ is the specific heat capacity of steam.

### III.V Boiler bank, economizer, and air heater

Heat transfer from flue gas to the boiler bank, economizer, and air heater is predominantly convective. Energy balance in the boiler bank results in the following equation.

$$m_{s2}\Delta i_{vl}=m_g c_{pg}(T_{g2}-T_{g3}) \tag{16}$$

If the temperature of the boiler bank tubes is assumed to be uniform, the heat transfer equation for the boiler bank may be expressed as

$$m_{s2}\Delta i_{vl}=\frac{U_{BB}A_{BB}(T_{g2}-T_{g3})}{\ln[(T_{g3}-T_v)/T_{g2}-T_v]} \tag{17}$$

where $A_{BB}$ is the boiler bank surface area. Since the internal heat transfer coefficient ($h_{BB,i}$) between feed water and tube walls of the boiler bank is much larger than the external heat transfer coefficient ($h_{BB,o}$) between tube walls and flue gas, $U_{BB}$ is approximately equal to $h_{BB,o}$.

The economizer and air heater are modelled as counter-flow heat exchangers. The analysis of this heat exchanger can be found in a heat transfer textbook [16]. It can be shown that the result of the analysis yields the following equations.

$$\ln\left(\frac{T_{g4}-T_{wi}}{T_{g3}-T_{we}}\right)=U_{EC}A_{EC}\left(\frac{1}{m_w c_{pw}}-\frac{1}{m_g c_{pg}}\right) \tag{18}$$

$$m_w c_{pw}(T_{we}-T_{wi})=m_g c_{pg}(T_{g3}-T_{g4}) \tag{19}$$

$$\ln\left(\frac{T_{g5}-T_{ai}}{T_{g4}-T_{ae}}\right)=U_{AH}A_{AH}\left(\frac{1}{m_a c_{pa}}-\frac{1}{m_g c_{pg}}\right) \tag{20}$$

$$m_a c_{pa}(T_{ae}-T_{ai})=m_g c_{pg}(T_{g4}-T_{g5}) \tag{21}$$

where $A_{EC}$ and $A_{AH}$ are, respectively, the economizer and air heater surface areas. Since the internal heat transfer coefficient ($h_{EC,i}$) between feed water and tube walls of the economizer is much larger than the external heat transfer coefficient ($h_{EC,o}$) between tube walls and flue gases, $U_{EC}$ is approximately equal to $h_{EC,o}$. For the air heater, the overall heat transfer coefficient is determined from

$$\frac{1}{U_{AH}}=\frac{1}{h_{AH,i}}+\frac{1}{h_{AH,o}} \tag{22}$$

where $h_{AH,i}$ is the internal heat transfer coefficient between air and tube walls of the air heater, and $h_{AH,o}$ is the external heat transfer coefficient between tube walls of the air heater and flue gas. It is assumed that tube thickness in the air heater is negligible.

### III.VI Heat transfer coefficients

External heat transfer coefficient of the superheater ($h_{SH,o}$) is the sum of convective ($h_c$) and radiative ($h_r$) heat transfer coefficients. The expression of the radiative heat transfer coefficient is given by Kakac [3].

$$h_r=5.11\times10^{-11}\alpha_g T_{ga}^3\left[\frac{1-(T_{SH}/T_{ga})^4}{1-(T_{SH}/T_{ga})}\right] \tag{23}$$

where $T_{ga}=(T_{g1}+T_{g2})/2$. The temperature of the superheater tubes ($T_{SH}$) is determined from the assumption that the superheater tubes have a uniform temperature.

$$T_{SH}=\frac{T_s e^{h_{SH,i}A_{SH}/m_s c_{pv}}-T_v}{e^{h_{SH,i}A_{SH}/m_s c_{pv}}-1} \tag{24}$$

Convective heat transfer coefficient is a function of Reynolds number, which is proportional to mass flow rate. It is assumed that reference values $h_{cr}$, $h_{SHr,i}$, $h_{SHr,o}$, $h_{BBr,o}$, $h_{ECr,o}$, $h_{AHr,i}$, and $h_{AHr,o}$ corresponding to the reference flow rates of steam, air, and flue gases ($m_{sr}$, $m_{ar}$, $m_{gr}$) are known. For other flow rates that differ from the reference flow rates, the heat transfer coefficients are determined from

$$h_c=h_{cr}\left(\frac{m_s}{m_{sr}}\right)^{0.6} \tag{25}$$

$$h_{SH,i}=h_{SHr,i}\left(\frac{m_s}{m_{sr}}\right)^{0.8} \tag{26}$$

$$h_{BB,o}=h_{BBr,o}\left(\frac{m_g}{m_{gr}}\right)^{0.6} \tag{27}$$

$$h_{EC,o}=h_{ECr,o}\left(\frac{m_g}{m_{gr}}\right)^{0.6} \tag{28}$$

$$h_{AH,i}=h_{AHr,i}\left(\frac{m_a}{m_{ar}}\right)^{0.8} \tag{29}$$

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$$h_{AH,o} = h_{AHr,o}\left(\frac{m_g}{m_{gr}}\right)^{0.6} \quad (30)$$

According to Rein [17], reference values of heat transfer coefficients are $h_{cr}$ = 0.065 kW/m.K, $h_{SHr,i}$ = 1.25 kW/m.K, $h_{BBr,o}$ = 0.07 kW/m.K, $h_{ECr,o}$ = 0.065 kW/m.K, $h_{AHr,i}$ = 0.045 kW/m.K, and $h_{AHr,o}$ = 0.045 kW/m.K.

## IV. OPTIMUM DISTRIBUTIONS OF HEATING SURFACE AREAS

The 21 primary variables in the industrial boiler system are mass flow rate of fuel ($m_f$), excess air ratio ($\phi$), steam pressure ($p_s$), steam temperature ($T_s$), mass flow rate of steam in the superheater ($m_s$), mass flow rate of steam in the boiler bank ($m_{s2}$), air temperatures into and out of the air heater ($T_{ai}$ and $T_{ae}$), feed water temperatures into and out of the economizer ($T_{wi}$ and $T_{we}$), temperatures of flue gas ($T_{g1}$, $T_{g2}$, $T_{g3}$, $T_{g4}$, and $T_{g5}$), furnace heating surface areas ($A_v$ and $A_t$), superheater surface area ($A_{SH}$), boiler bank surface area ($A_{BB}$), economizer heating surface area ($A_{EC}$), and air heater surface area ($A_{AH}$). After eliminating intermediate variables, there are 10 equations [Eqs. (1), (11), (13), (15) – (21)] in the system. This means that 11 of the variables must have specified values so that the solution of the system can be found. Known variables in a boiler analysis are $m_f$, $\phi$, $p_s$, $T_{ai}$, $T_{wi}$, $A_v$, $A_t$, $A_{SH}$, $A_{BB}$, $A_{EC}$, and $A_{AH}$. Results of an analysis enables the determination of boiler efficiency from

$$\eta = \frac{m_s(i_s - i_{wi})}{m_f HHV} \quad (31)$$

It can be seen from Eq. (31) that the value of boiler efficiency is fixed if $m_f$, $m_s$, and $T_s$ are specified. This means that, at a specified value of boiler efficiency, one of the three heating surface areas ($A_{SH}$, $A_{EC}$, and $A_{AH}$) is a free parameter. If $A_{EC}$ is a free parameter, the system now has $m_{s2}$, $T_{ae}$, $T_{we}$, $T_{g1}$, $T_{g2}$, $T_{g3}$, $T_{g4}$, $T_{g5}$, $A_{SH}$, and $A_{AH}$ as the 10 unknowns. Each distribution of heating surface areas, defined by $A_{SH}$, $A_{EC}$, and $A_{AH}$. It is assumed that the cost of installing a heat exchanger is the product of the unit cost and the surface area of the heat exchanger. This assumption is valid for superheater, economizer, and air heater because they have relatively large surface areas. Assume that the nominal unit costs of superheater, economizer, and air heater surface areas are, respectively, 500, 120, and 100 \$/m$^2$. The total cost of installing the surface areas of superheater, economizer, and air heater is

$$C_{total} = 500A_{SH} + 120A_{EC} + 100A_{AH} \quad (32)$$

Out of the infinite number of distributions of heating surface areas that yield a given value of boiler efficiency, there exists the optimum distribution that corresponds to the minimum cost. It is to be found by simulation.

If the total installation cost is specified, along with $m_s$ and $T_s$, $\eta$ becomes an unknown. A value of $A_{EC}$ represents a distribution of heating surface areas because $A_{SH}$ and $A_{AH}$ are dependent on $A_{EC}$. There is an infinite number of distributions of heating surface areas corresponding to the specified total installation cost. There exists the optimum distribution that yields the maximum boiler efficiency, which can be found by simulation.

## V. RESULTS AND DISCUSSION

One of the largest industries that use industrial boilers is the cane sugar industry. Bagasse is the remainder of sugar cane after juice is extracted. It is the main fuel that the industrial boiler uses to produce process steam and generate power in the cogeneration system of the cane sugar industry. The composition of bagasse according to Rein [17] is 22.04% carbon, 21.07% oxygen, 2.72% hydrogen, 0.15% nitrogen, 0.02% sulphur, 52.00% moisture, and 2.00% ash.

In order to demonstrate the determination of the optimum distribution of heating surface areas that minimizes the total installation cost and the determination of the optimum distribution of heating surface areas that maximizes the boiler efficiency, a reference boiler is considered. Parameters of this boiler are shown in Table 1.

**Table 1.** Parameters of the reference boiler

| | |
|---|---|
| $p_s$ (MPa) | 4.5 |
| $T_s$ (°C) | 480 |
| $m_s$ (kg/s) | 11.11 |
| $m_f$ (kg/s) | 5.12 |
| $T_{ai}$ (°C) | 30 |
| $T_{wi}$ (°C) | 105 |
| $A_v$ (m$^2$) | 3.5 |
| $A_t$ (m$^2$) | 1.2 |
| $A_{SH}$ (m$^2$) | 262 |
| $A_{BB}$ (m$^2$) | 1052 |
| $A_{EC}$ (m$^2$) | 496 |
| $A_{AH}$ (m$^2$) | 1077 |
| $T_{ae}$ (°C) | 118 |
| $T_{we}$ (°C) | 161 |
| $T_{g1}$ (°C) | 609 |
| $T_{g2}$ (°C) | 483 |
| $T_{g3}$ (°C) | 271 |
| $T_{g4}$ (°C) | 167 |
| $T_{g5}$ (°C) | 107 |
| $\eta$ (%) | 72.0% |
| $C_{total}$ (\$) | 298388 |

Figure 3 shows that, in order to maintain the value of 72.0% for boiler efficiency (η), $A_{SH}$ and $A_{AH}$ must have specific values that are dependent on $A_{EC}$. Therefore, a value of $A_{EC}$ represents a distribution of heating surface areas. Figure 4 shows that the total cost of heating surface areas reaches the minimum value of $293998 when $A_{EC}$ is 757.4 m$^2$, $A_{SH}$ is 273.5 m$^2$, and $A_{AH}$ is 663.6 m$^2$. The minimum cost is 1.5% less than the total cost of the reference boiler.

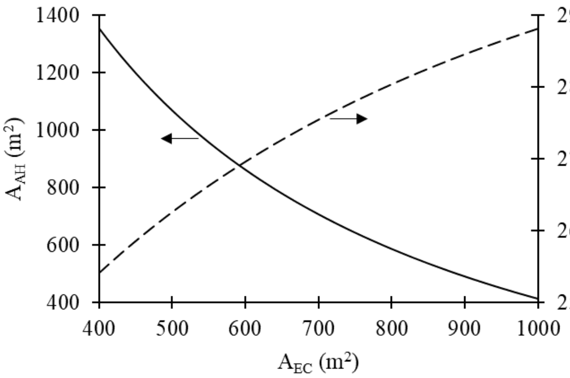


**Fig. 3.** Different distributions of heating surface areas that yield the boiler efficiency of 72.0%

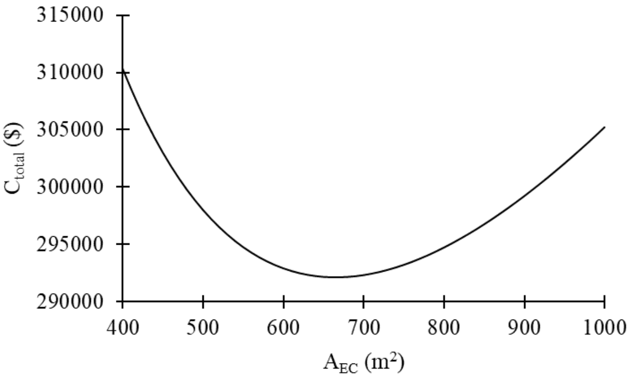


**Fig. 4.** Variation of total cost of heating surface areas with distribution of heating surface areas corresponding to the boiler efficiency of 72.0%

If the total cost is fixed at $301082, Fig. 5 shows that $A_{SH}$ must increase, and $A_{AH}$ must decrease as $A_{EC}$ increases in order to maintain the total cost at this value. Therefore, a value of $A_{EC}$ represents a different distribution of heating surface areas in this case. Figure 6 shows the variation of boiler efficiency with distribution of heating surface areas. It can be seen that the boiler efficiency is maximum at the optimum distribution of heating surface areas, in which $A_{EC}$ is 689.0 m$^2$, $A_{SH}$ is 274.7 m$^2$, and $A_{AH}$ is 783.5 m$^2$. The maximum efficiency is 72.1%, which is 0.1% larger than the efficiency of the reference boiler.

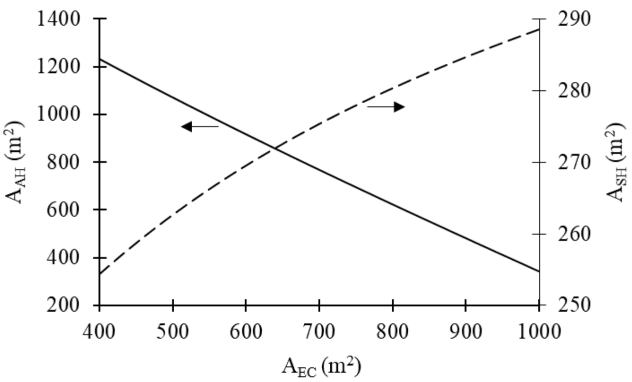


**Fig. 5.** Different distributions of heating surface areas corresponding to the total cost of $298388 for heating surface areas

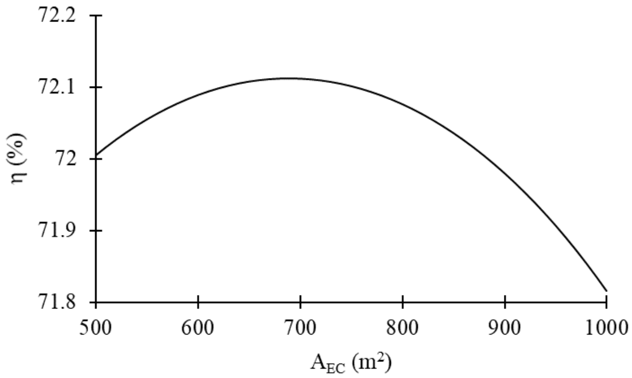


**Fig. 6.** Variation of boiler efficiency with distribution of heating surface areas corresponding to the total cost of $298388 for heating surface areas

## VI. CONCLUSION

Industrial boiler consists of furnace, evaporator, steam drum, superheater, boiler bank, economizer, and air heater. A model of industrial boiler that takes into account these components is developed in this paper. This model is used to determine the optimum distribution of heating surface areas of superheater, economizer, and air heater under given conditions. The model shows that there are infinitely many distributions of the heating surface areas that correspond to a specified value of boiler efficiency, and there are infinitely many distributions of the heating surface areas that correspond to a total cost of installing these surface areas. Furthermore, it is shown that there exist the optimum distribution that results in the minimum total cost for a given value of boiler efficiency and the optimum distribution that results in the maximum boiler efficiency for a given value of total cost. Simulation results for a reference boiler indicate that the optimum distribution that minimizes the total cost increases the total cost by 1.5% compared with a non-optimum distribution, and that the optimum distribution that maximizes the boiler efficiency increases the boiler efficiency by 0.1% compared with a non-optimum distribution.

**REFERENCES**

[1] Rusinowski H, Stanek W. Neural modelling of steam boilers. Energy Convers Manage. 2007; 48: 2802-2809.

[2] Kljajic M, Gvozdenac D, Vukmirovic, S. Use of neural networks for modeling and predicting boiler's operating performance. Energy. 2012; 45: 304-311.

[3] Kakac S. Boilers, Evaporators, and Condensers. New York: Wiley; 1991.

[4] Diez LI, Cortes C, Campo A. Modelling of pulverized coal boilers: review and validation of on-line simulation techniques. Appl Therm Eng. 2005; 25: 1516-1533.

[5] Cantrell C, Idem S. On-line performance model of the convection passes of a pulverized coal boiler. Heat Trans Eng. 2010; 31: 1173-1183.

[6] Moghari M, Hosseini S, Shokouhmand H, Sharifi H, Izadpanah S. A numerical study on thermal behavior of a D-type water-cooled steam boiler. Appl Therm Eng. 2012; 37: 360-372.

[7] Hajebzadeh H, Ansari ANM, Niazi S. Mathematical modeling and validation of a 320 MW tangentially fired boiler. Appl Therm Eng. 2018; 146: 232-242.

[8] Stultz SC, Kitto JB. Steam: Its Generation and Use. Barberton: Babcock & Wilcox; 1992.

[9] Karim MR, Naser J. CFD modelling of combustion and associated emission of wet woody biomass in a 4 MW moving grate boiler. Fuel. 2018; 222: 656-674.

[10] Zhang G, Xu W, Wang X, Yang Y. Analysis and optimization of a coal-fired power plant under a proposed flue gas recirculation mode. Energy Convers Manage. 2015; 102: 161-168.

[11] Chandok JS, Kar IN, Tuli S. Estimation of furnace exit gas temperature (FEGT) using optimized radial basis and back-propagation neural networks. Energy Convers Manage. 2008; 49: 1989-1998.

[12] Qian H, Guo X, Fan S, Hagos K, Lu X, Liu C, Huang D. A simple prediction model for higher heat value of biomass. J Chem Eng Data. 2015; 61: 4039-4045.

[13] Leckner B. Spectral and total emissivity of water vapour and carbon dioxide. Combust Flame 1972; 19: 33-48.

[14] Taler D, Trojan M, Dzierwa P, Kaczmarski K, Taler J. Numerical simulation of convective superheaters in steam boilers. Int J Therm Sci. 2018; 129: 320-333.

[15] Sobota T. Improving steam boiler operation by on-line monitoring of the strength and thermal performance. Heat Trans Eng. 2018; 39: 1260-1271.

[16] Incropera FP, DeWitt DP. Introduction to Heat Transfer (Third edition). New York: Wiley; 1996.

[17] Rein P. Cane Sugar Engineering. Berlin: Verlag; 2007.