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J. KOZIOŁ*, K. BANASIAK*

MATHEMATICAL MODEL OF THE METALLURGICAL WASTE HEAT RECOVERY SYSTEM GENERATING USEFUL HEAT AND REFRIGERATION BASED ON THE EXPERIMENTALLY DETERMINED CHARACTERISTICS

MODEL MATEMATYCZNY ZIĘBNICZO-GRZEJNEGO UKŁADU ODZYSKU HUTNICZEJ ENERGII ODPADOWEJ OPARTY O DOŚWIADCZALNIE WYZNACZONE CHARAKTERYSTYKI URZĄDZEŃ

A mathematical model of the metallurgical low and medium temperature waste heat recovery system, for the most part based on experimental relationships, is presented. The utilization system consisting of a waste heat boiler, an absorption refrigerator and a cooling towers system has been analyzed, considered to cover the integrated heat and refrigeration demands. The selected results of the numerical calculations, simulating the performance of the system in the case of the changing ambient conditions as well as the exhaust gas temperature and its flow rate, has been presented.

W pracy przedstawiono model matematyczny układu odzysku średnio- i nisko- temperaturowej hutniczej energii odpadowej, oparty w większości na zależnościach wyznaczonych doświadczalnie. Przeanalizowano układ utylizacyjny składający się z kotła odzysknicowego, chłodziarki absorpcyjnej oraz systemu chłodni wentylatorowych, wytwarzający ciepło na potrzeby grzewcze oraz zimno na potrzeby chłodnicze. Przedstawiono wybrane rezultaty obliczeń numerycznych, symulujących pracę systemu utylizacyjnego w warunkach zmienności parametrów termicznych otoczenia oraz temperatury i strumienia odpadowych spalin hutniczych.

1. Introduction

Energy loss caused by the incomplete use of exergy of the metallurgical products is known as the waste energy. Reduction of the energy losses by the utilization of the waste energy resources contributes to a growth of efficiency in the metallurgical processes and lowers the amount of deleterious substances emitted to the environment.

* INSTYTUT TECHNIKI CIEPLNEJ, POLITECHNIKA ŚLĄSKA, 44-100 GLIWICE, UL. KONARSKIEGO 22

This paper discusses the problem of the utilization of the low and medium temperature waste energy, accessible in the form of the stream of the hot exhaust gases. A typical source of the low and medium temperature physical exergy in the metallurgical industry is the heat furnace in the rolling mill. A waste stream of hot gases, flowing out from the furnace with temperature considerably higher than the ambient one (300-600°C), may be applied to produce the technological steam or hot water, covering the central heating requirements, and as a driving fluid in an absorption refrigerator.

A mathematical model of the waste heat recovery system, utilizing the physical exergy of the metallurgical exhaust gases, flowing out from the steel-plant heat furnace, considered to cover the internal heating and refrigeration demands, is presented in this paper. The model, designed to simulate the annual operation of a genuine system as well as to perform the on-line calculations, which is helpful in control systems, is based on experimental and theoretical relationships and aims to determine the maximum momentary heating or refrigeration capacity, whilst considering the changing conditions of the ambient temperature and humidity and taking into account the variations in the exhaust gases stream.

2. Energy characteristics

The transition functions (characteristics) of the energy systems may be determined through the following methods [8]:

— theoretical analysis,
— statistical analysis,
— theoretical-statistical analysis,
— special measurement procedure.

The theoretical analysis, based on physical laws and rules, is usually insufficient to include the influence of all phenomena occurring during changes in the operating conditions. The statistical analysis verifies the transition function coefficients by statistical procedures and is usually applied in the case of being in possession of a sufficiently numerous set of the experiment results. The type of the approximation function is assumed a priori.

The best fit of the approximation function to the genuine performance may be achieved due to the combination of both methods called theoretical-statistical analysis. In this method the first step is to establish the transition function type, taking as a base the simplified theoretical analysis. Then, the values of the unknown coefficients are determined as a result of the statistical processes.

The special measurement procedure establishes the transition function in series of planned experiments, aiming to estimate the influence of the analyzed parameters on the overall performance.

Creating the model, in order to establish the capacity of particular devices, various characteristics have been incorporated. The characteristics represent relationships among capacity and a driving energy flow rate, thermal parameters of the working fluid and useful products, ambient conditions etc. These correlations have been ascertained experimentally and theoretically or in combinations of the above mentioned means. The application of the characteristics replaces some more complicated mathematical models and simplifies the whole system which brings to shorter times of the calculations.

3. Waste heat recovery system

The examined system of the waste heat recovery (Fig. 1) consists of the following elements:

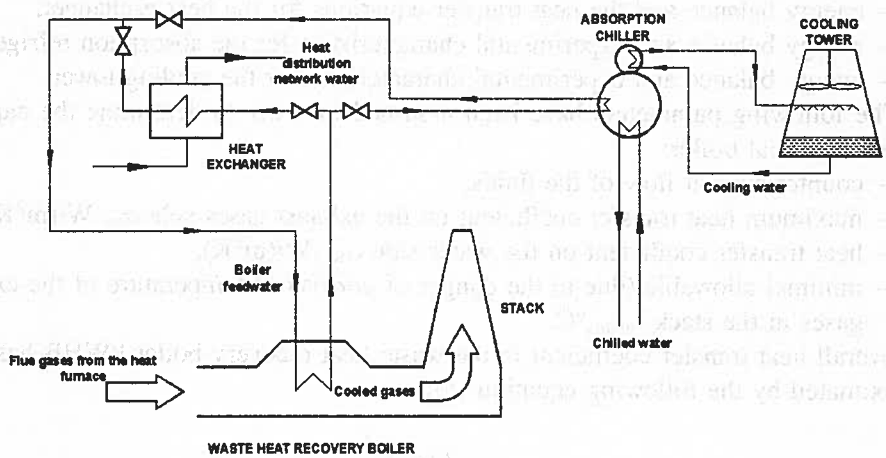

Fig. 1. Scheme of the waste heat recovery system

— hot water waste heat recovery boiler, water-tube type,
— water-to-water heat exchanger,
— single-effect lithium bromide-water absorption refrigerator,
— system of the induced-draft cooling towers.

Creating the mathematical model, in order to simplify the numerical calculations, it has been presumed that the system operates only in two independent modes:

— winter mode — occurring when ambient temperature falls below 12°C; the total amount of hot water generated in the heat recovery boiler is designated to cover the heat demands (final temperature of the heat-carrier is above the ambient one),
— summer mode — occurring when ambient temperature exceeds 12°C; the total amount of hot water generated in the heat recovery boiler is designated to drive an absorption refrigerator covering the refrigeration demands (final temperature of the heat-carrier is below the ambient one).

Such a configuration means that the waste heat recovery boiler co-operates always with only one device and the absorption refrigerator and the heat exchanger operate separately.

4. Mathematical model of the system

In order to determine the total amount of the energy carriers production, taking as a base assumptions described in the previous paragraph, a mathematical model of the examined system has been created. This model aims to simulate the performance of the waste energy recovery system in the conditions of the changing exhaust gases flow and temperature. Additionally, the impact of the ambient temperature and humidity changes may be examined.

A complete model of the system has been composed of the following elements:
— energy balance and the heat transfer equations for the waste heat boiler,
— energy balance and the heat transfer equations for the heat exchanger,
— energy balance and experimental characteristics for the absorption refrigerator,
— energy balance and experimental characteristic for the cooling tower.

The following parameters have been assumed in order to determine the capacity of the waste heat boiler:
— counter-current flow of the fluids,
— maximum heat transfer coefficient on the exhaust gases side α_g, W/(m^{2}K),
— heat transfer coefficient on the water side α_w ,W/(m^{2}K),
— minimal allowable (due to the danger of corrosion) temperature of the exhaust gases in the stack t_{gmin},°C.

The overall heat transfer coefficient in the waste heat recovery boiler kWHB has been approximated by the following equation [5]:

$$\frac{1}{k_{WHB}} = \frac{1}{\alpha_g}\left(\frac{\dot{W}_{g\,\mathrm{max}}}{\dot{W}_g}\right)^{0.6} + \frac{1}{\alpha_w}, \tag{1}$$

where:
$\dot{W}_{gmax}$ — maximal thermal capacity flow rate of the exhaust gases,
W/K $\dot{W}_g$ — momentary thermal capacity flow rate of the exhaust gases, W/K

The overall heat transfer coefficient in the heat exchanger k_{HE} has been determined with the assumption that the heat transfer coefficients on the sides of the heat distribution network and boiler feedwater are identical and the conduction resistance may be neglected [3].

It has been assumed that the values of the temperature of the heat network water are related to the ambient conditions by characteristics, reflecting the relationship between the ambient temperature and the network water temperature before/after the heat exchanger.

A single-stage, hot water driven lithium-bromide absorption refrigerator has been applied as the "cold" generation device. In order to determine the capacity of the refrigerator the energy characteristics, reflecting the relationship between the coefficient of performance and thermal parameters (especially temperature) of the working fluid in essential sites of the refrigerator, may be used. Moreover, functions approximating the correlation between the percentage of the rated refrigeration capacity and thermal conditions of the working fluid are commonly available. Typical characteristics for the single-stage lithium-bromide absorption refrigerator have been included [1, 6, 7]:

— ratio w_q of the momentary chilling capacity over the rated chilling capacity with respect to the driving water temperature and cooling water temperature (Fig. 2),

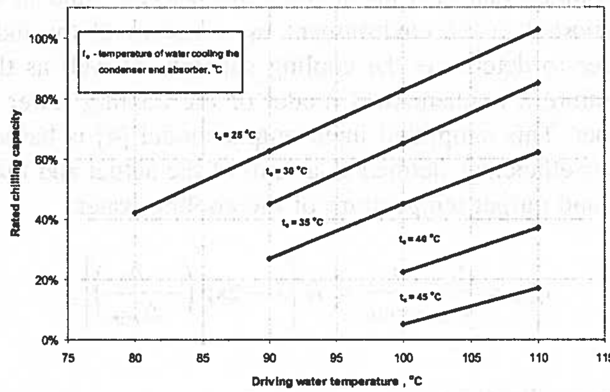

Fig. 2. Ratio of the momentary chilling capacity over the rated chilling capacity with respect to the driving water temperature and cooling water temperature (temperature of the chilled wter 6°C) [6]

— coefficient of performance COP_{ACH} with respect to the driving water temperature and cooling water temperature (Fig. 3).

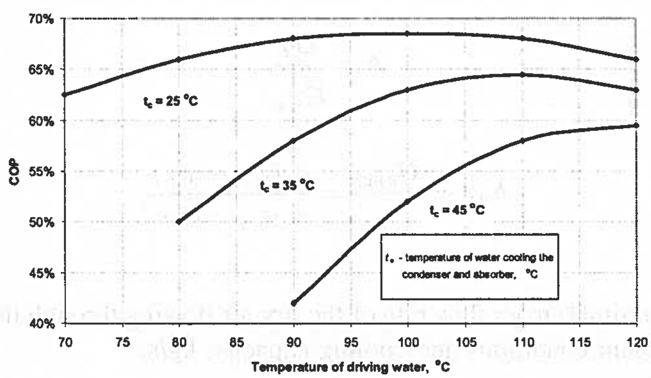

Fig. 3. Coefficient of performance with respect to the driving water temperature and cooling water temperature (temperature of the chilled water 6°C) [6]

Incorporating these characteristics into the model, the following assumptions have been made:

— value of the inlet temperature of the chilled water t_{chi} is constant,

— value of the outlet temperature of the chilled water t_{cho} is constant,

— chilling capacity varies between 10 and 100% of the rated capacity,
— the cooling water inlet temperature t_{ci} cannot be lower than the minimal admissible one,
— value of the inlet temperature of the driving water t_{di} is constant,
— correlation between the refrigeration capacity and the driving water flow rate is linear.

It has been assumed that heat taken from the working fluid in the absorber and condenser is transported to the environment by a system of the induced-draft cooling towers. In order to determine the cooling capacity as well as the cooling water inlet/outlet temperature a mathematical model of the cooling tower has been implemented in this paper. This simplified input-output model [4] is based on the concept of the cooling tower efficiency, defined as a ratio of the actual and maximal difference between the input and output temperature of the cooling water:

$$\eta_{CT} = \frac{t_{c\,o} - t_{c\,i}}{t_{c\,o} - t_{WB}} = K\left[1 - \exp\left(-\frac{\Lambda}{\Lambda_{\min}}\right)\right], \tag{2}$$

where:
η_{CT} — cooling tower efficiency, t_{ci},
$t)_{co}$ — inlet/outlet temperature of the absorption chiller cooling water, °C,
t_{WB} — Wet Bulb temperature, °C,
K — correction factor (estimated experimentally),
Λ, Λ_{min} — actual and minimal ventilation coefficient.
Ventilation coefficients are defined as follows:

$$\Lambda = \frac{\dot{G}_a}{\dot{G}_{w\,c}} \tag{3}$$

$$\Lambda_{\min} = \frac{\dot{G}_{a\min}}{\dot{G}_{w\,c}} = \frac{c_w(t_{c\,o} - t_{WB})}{i_{a\ theor} - i_{amb}}, \tag{4}$$

where:
$\dot{G}_{a\min}$, $\dot{G}_a$ — (minimal) mass flow rate of the dry air flowing through the cooling tower at specified ambient conditions and cooling capacity, kg/s,
$\dot{G}_{w\,c}$ — mass flow rate of the cooling water,
kg/s, c_w — specific heat of water, J/kgK,
i_{amb} — enthalpy of air at ambient conditions, J/kg,
i_{atheor} — enthalpy of air at temperature $t_{atheor=tco}$ and humidity 100%, flowing out from the ideal cooling tower, J/kg.

The mass flow rate of air flowing through the induced-draft cooling tower is a result of the equality of the hydraulic resistance δp and the fan differential pressure Δp_w [4]:

$$\Delta p_w = f\left(\dot{V}_a\right) = \left(\frac{\Lambda \dot{G}_{w\,c}}{\rho_{a\,i}}\right) \tag{5}$$

$$\delta p = \left(A + B\frac{\dot{G}_{w\,c}}{F_s}\right)\frac{\Lambda^2 (1 + X_{a\,i})^2 \dot{G}_{wc}^2}{2F_f^2 \rho_{a\,i}}, \tag{6}$$

where:
ρ_{ai} — density of ambient air flowing into the cooling tower, kg/m^3,
A, B — coefficients characterising the hydraulic resistance of the cooling tower airflow (estimated experimentally),
X_{ai} — mixing ratio of ambient air flowing into the cooling tower, kg$_P$H$_2$O/kg$_{\text{dryair}}$,
F_S — surface area of a cross section below the sprinkler, m^2,
F_D — surface area of the air inlet, m^2.

Since the ambient conditions, temperature and a stream of the exhaust gases are given in a form of the duration curves, the performed calculations are based on the Monte Carlo method. The randomly determined relative time is independently sampled for each of the four quantities: flow rate and temperature of the exhaust gases and the relative humidity and temperature of ambient air. For each sampling, the unknown values are determined directly from the duration curves.

According to the ambient temperature value, for each series of samplings, an operating mode (summer or winter), heat and "cold" demand and the heating water temperature (in the winter mode) is determined. Hence, after each series of the four input data samplings the next step is a solution of a nonlinear equations system, formulated dependently on the operation mode.

Each set of equations is solved using the Newton method. The basic set of nonlinear equations has a general form:

$$F_i(x_1, x_2, ..., x_n) = 0 \quad i = 1, 2, 3, ..., n, \tag{7}$$

where:
$x_1, x_2, x_3, ..., x_n$ — unknown parameters,
n — number of equations.

According to the Newton method set (7) is transformed to the form:

$$a_{i1}\delta x_1 + a_{i2}\delta x_2 + ... + a_{ij}\delta x_j + ... + a_{in}\delta x_n + f(x_1^0, x_2^0, ..., x_j^0, ..., x_n^0) = 0 \quad i, j = 1, 2, ...n, \tag{8}$$

where:
x_j^0 — initially estimated value of quantity
x_j, δx_j — correction of unknown x_j^0.
Coefficients aij are expressed by the following formula:

$$a_{ij} = \frac{\partial F_i}{\delta x_j}. \tag{9}$$

For each estimation of x_j^0 value set (8) is solved using a procedure for the solution of linear algebraic equations. The eventual values of the calculated parameters at the end of the previous step are used as starting values x_j^0 for the next Newton procedure. Due to the nonlinearity of the problem (7) an iteration process is used to obtain satisfactory accuracy of calculations.

For the summer mode the equations system consists of six equations, based on the characteristics of the absorption refrigerator (equations 10 and 11) and cooling tower (equation 12), the energy balance for the refrigerator (equation 13) and the energy balance and the heat transfer equation for the waste heat boiler (equations 14 and 15):

$$\dot{G}_{ch}\,(t_{ch\,i} - t_{ch\,o}) - \frac{\dot{G}_d}{z}\,(t_{d\,i} - t_{d\,o})\;\;COP_{ACH}\,(t_{d\,i}, t_{c\,i}) = 0 \tag{10}$$

$$\dot{G}_{ch}\,c_w\,(t_{ch\,i} - t_{ch\,o}) - \underset{ACH\;NOM}{\dot{Q}}\left(\frac{\frac{\dot{G}_d}{z}}{\underset{NOM}{\dot{G}_d}}\right) w_q\,(t_{d\,i}, t_{c\,i}) = 0 \tag{11}$$

$$t_{c\,o} - t_{c\,i} + K\,(t_{c\,o} - t_{WB})\left[1 - \exp\left(-\frac{\Lambda}{\Lambda_{\min}}\right)\right] = 0 \tag{12}$$

$$\dot{G}_{ch}\,z\,(t_{ch\,i} - t_{ch\,o}) + \dot{G}_d\,(t_{d\,i} - t_{d\,o}) - \dot{G}_{w\,c}\,z\,(t_{c\,o} - t_{c\,i}) = 0 \tag{13}$$

$$(t_{d\,i} - t_{d\,o})\left(\dot{G}_d\,c_w - H_1\,\dot{G}_g\,c_{p\,g}\Bigg|\begin{matrix} t_{g\,i} \\ t_{g\,o}\end{matrix}\right) - \left(t_{g\,i} - t_{d\,o}\right)(1 - H_1)\left(\dot{G}_g\,c_{p\,g}\Bigg|\begin{matrix} t_{g\,i} \\ t_{g\,o}\end{matrix}\right) = 0 \tag{14}$$

$$H_1 = \exp\left[-A_{WHB}\;k_{WHB}\left(\frac{1}{\dot{G}_g\,c_{p\,g}\Bigg|\begin{matrix} t_{g\,i} \\ t_{g\,o}\end{matrix}} - \frac{1}{\dot{G}_d\,c_w}\right)\right], \tag{15}$$

where: $\dot{G}_{ch}$ — mass flow rate of the chilled water, kg/s

$\dot{G}_d$ — mass flow rate of the driving water, kg/s,

z — number of absorption chillers,

t_{di}/t_{do} — driving water inlet/outlet temperature,

°C, t_{ci}/t_{co} — absorption chiller cooling water inlet/outlet temperature, °C,

$COP_{ACH}(t_{di}, t_{ci})$ — approximation of the relationship between the absorption chiller

coefficient of performance and the driving/cooling water temperature,
$\dot{G}_{d\ NOM}$ — nominal mass flow rate of the driving water, kg/s,
$\dot{G}_{ACH\ NOM}$ — nominal chilling capacity ($\dot{G}_d = \dot{G}_{dNOM}, t_{ci} = t_{ciNOM}, t_{di} = t_{diNOM}$), W,
$w_q(t_{di}, t_{ci})$ — approximation of the relationship between the absorption chiller relative capacity and the driving/cooling water temperature,
$\dot{G}_g$ — mass flow rate of the exhaust gases, kg/s,
t_{gi}/t_{go} — exhaust gases inlet/outlet temperature before/after the waste heat boiler, °C,
$c_{p\,g}\Big|_{t_{g\,o}}^{t_{g\,i}}$ — mean specific heat of the exhaust gases for the temperature range ($t_{g}o \div t_{gi}$), J/kgK,
A_{WHB} — heat exchange surface area of the waste heat recovery boiler, m^2

The unknown quantities are: $\dot{G}_{ch}, t_{ci}, t_{co}, \dot{G}_d, t_{do}$. Hence, assuming the heat exchange surface area of the waste heat boiler, the number of absorption refrigerators and cooling towers, values of the nominal mass flow rate of driving and cooling water and establishing the minimal exhaust gases temperature after the waste heat boiler, for each series of sampling, the maximum possible chilling capacity $\dot{Q}_{ch\ \max}$ may be calculated:

$$\dot{Q}_{ch\ \max} = \dot{G}_{ch}\, c_w\, (t_{ch\,i} - t_{ch\,o})\,. \tag{16}$$

For the winter mode the equations system consists of five equations, based on energy balances and heat transfer equations for the waste heat recovery boiler and heat exchanger:

$$\dot{G}_g\left(c_{p\,g}\Big|_{0°C}^{t_{g\,i}}\left(t_{g\,i} - 0°C\right) - c_{p\,g}\Big|_{0°C}^{t_{g\,o}}\left(t_{g\,o} - 0°C\right)\right) - \dot{G}_d\, c_w\,(t_{d\,i} - t_{d\,o}) = 0 \tag{17}$$

$$\dot{G}_d\,(t_{d\,i} - t_{d\,o}) - \dot{G}_h\,(t_h - t_r) = 0 \tag{18}$$

$$\left(t_{g\,i} - t_{d\,i}\right)H_1 - \left(t_{g\,o} - t_{d\,o}\right) = 0 \tag{19}$$

$$(t_{d\,i} - t_h)\,H_2 - (t_{d\,o} - t_r) = 0 \tag{20}$$

$$H_2 = \exp\left[-A_{HE}\, k_{HE}\left(\frac{1}{\dot{G}_d\, c_w} - \frac{1}{\dot{G}_h\, c_w}\right)\right], \tag{21}$$

where:
$\dot{G}_h$ — mass flow rate of the heat distribution network water, kg/s,
t_h/t_γ — temperature of the heat distribution network water after/before the heat exchanger (temperature of hot/return water), °C,
A_{HE} — heat transfer surface area of the heat exchanger, m^2.

The unknown quantities are: $\dot{G}_h, t_{go}, t_{di}, t_{do}$. Hence, assuming the heat exchange surface area of the waste heat boiler and heat exchanger, and establishing the minimal temperature of the exhaust gases after the waste heat boiler, for each series of sampling, the maximum possible heating capacity $\dot{G}_{h\max}$ may be determined:

$$\dot{Q}_{h\ \max} = G_h\, c_w\,(t_h - t_r) \tag{22}$$

The annual production of heat and "cold" is determined as an integral from the momentary heating and chilling power.

5. Results of numerical calculations

In order to simulate the performance of the metallurgical waste energy recovery system, based on the model described above, the example calculations have been per-

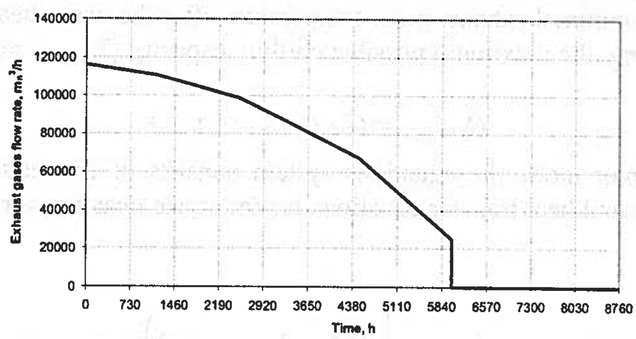

Fig. 4. Annual duration curve of the exhaust gases flow rate (data for a typical, single heat furnace)

formed. With the purpose of estimating the waste energy resources, accessible in the metallurgical industry, data for a typical heat furnace in an iron-works have been used. Typical values of the exhaust gases flow rate and temperature are presented in Fig. 4 and Fig. 5.

Performing the calculations for the defined above waste energy resources, the annual ambient humidity and temperature data for the III climatic zone in Poland [2] have been incorporated into the numerical model.

During the research aiming to determine the annual duration curves of the maximum heating and refrigeration capacity (Fig. 6) the following parameters have been examined:

— Capacity of the waste heat recovery boiler,
— Nominal refrigeration capacity,
— Nominal cooling towers system capacity.

The capacity of the waste heat boiler is connected with its heat exchange area, therefore the influence of the capacity has been examined by a change of the total heat

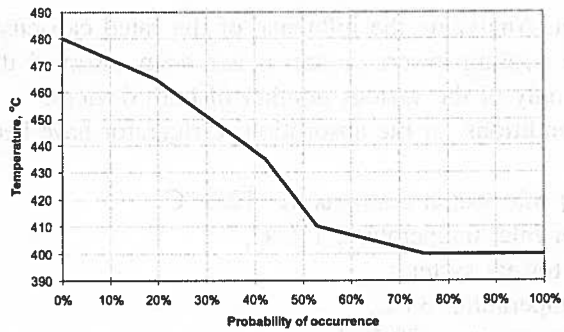

Fig. 5. Duration curve of the exhaust gases temperature with respect to the probability of occurrence (data for a typical, single heat furnace)

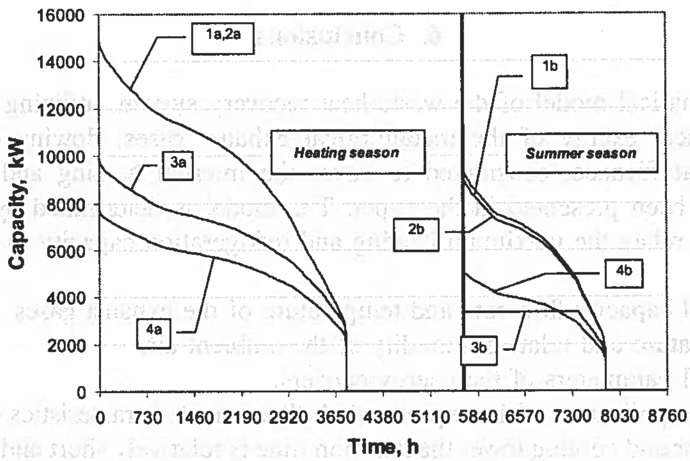

Fig. 6. Annual duration curves of the maximum heating and refrigeration capacity of the waste heat recovery system. 1a — Heating power (waste heat boiler area 1500 m^2; nominal refrigeration capacity 3 $\times$ 3.4 MW; nominal cooling tower capacity 6 $\times$ 1.3 MW). 2a — Heating power (waste heat boiler area 1500 m^2; nominal refrigeration capacity 3 $\times$ 3.4 MW; nominal cooling tower capacity 4 $\times$ 1.3 MW). 3a — Heating power (waste heat boiler area 700 m^2; nominal refrigeration capacity 1 $\times$ 3.4 MW; nominal cooling tower capacity 2 $\times$ 1.3 MW). 4a — Heating power (waste heat boiler area 500 m^2; nominal refrigeration capacity 2 $\times$ 3.4 MW; nominal cooling tower capacity 3 $\times$ 1.3 MW). 1b — Refrigeration power (waste heat boiler area 1500 m^2; nominal refrigeration capacity 3 $\times$ 3.4 MW; nominal cooling tower capacity 6 $\times$ 1.3 MW). 2b — Refrigeration power (waste heat boiler area 1500 m^2; nominal refrigeration capacity 3 $\times$ 3.4 MW; nominal cooling tower capacity 4 $\times$ 1.3 MW). 3b — Refrigeration power (waste heat boiler area 700 m^2; nominal refrigeration capacity 1 $\times$ 3.4 MW; nominal cooling tower capacity 2 $\times$ 1.3 MW). 4b — Refrigeration power (waste heat boiler area 500 m^2; nominal refrigeration capacity 2 $\times$ 3.4 MW; nominal cooling tower capacity 3 $\times$ 1.3 MW)

transfer surface area. Analyzing the influence of the rated capacity of the absorption refrigerator and the cooling towers system it has been assumed that a change in its capacity is a result only of the various number of both devices.

The nominal conditions for the absorption refrigerator have been assumed as follows:

— chilled water inlet/outlet temperature: 12/6 °C,
— driving water inlet temperature: 110°C,

and for the cooling towers system:

— hot water temperature: 35°C,
— cold water temperature: 29.5°C,
— ambient temperature: 28°C,
— ambient relative humidity: 70%.

6. Conclusions

A mathematical model of the waste heat recovery system, utilizing the low temperature physical exergy of the metallurgical exhaust gases, flowing out from the steel-plant heat furnace, considered to cover the internal heating and refrigeration demands, has been presented in the paper. The model is determined by a system of equations that relate the maximum heating and refrigeration capacity to the following parameters:

— thermal capacity flow rate and temperature of the exhaust gases,
— temperature and relative humidity of the ambient air,
— thermal parameters of the energy carriers.

Due to the application of the experimental- theoretical characteristics of the absorption refrigerator and cooling tower the iteration time is relatively short and the operation parameters are calculated almost immediately. This fact makes the presented model an appropriate tool to control the genuine systems operation on-line e.g. regulating the cooling tower fan or changing the mass flow rate of the boiler feedwater.

Additionally, the model makes it possible to optimize the technological parameters of the following devices: the waste heat recovery boiler, absorption refrigerator and cooling tower.

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