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WASTE HEAT RECOVERY OF THE REGENERATOR IN A GLASS FACTORY: A COMPARISON OF AN ANALYTICAL METHOD AND EXPERIMENTS

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ABSTRACT

One of the excellent methods for recovering waste heat from combustion furnaces is using regenerators. However, regenerators used in the glass industry are complex systems owing to the transient nature of their operating cycle. In this work, the heat transfer characteristics of the regenerators in a glass melting furnace are numerically and experimentally studied. A one-dimensional heat transfer model is proposed to investigate the heat transfer characteristics of real regenerators. With the design and operating parameters based on experimental data obtained from a real glass factory, the numerical results show a relatively good agreement with the measured results. The trends of the outlet temperatures in heating and cooling processes obtained from the numerical study at the convergent condition are relatively consistent with the measurements. In addition, the difference in the thermal recovery efficiency of the regenerator between the numerical and experimental results is only 4.3%. Thus, the proposed simple model is a believable representation for predicting the heat transfer characteristics of the regenerator in the glass or steel industry.

Keywords: *Glass melting furnace; regenerator; recovery efficiency; waste heat recovery; heat transfer.*

1. INTRODUCTION

Heat energy plays an important role in our lives due to its applications in many sectors [1-3]. It can be transferred from one object to another by the mechanism of conduction, convection, and radiation [4]. Accurate analysis of heat transfer characteristics of any system is critical for successfully designing heat exchangers [5-7].

Recovering industrial waste heat can be achieved via numerous methods, such as using combustion exhaust gas to preheat combustion air or feed-water in industrial boilers. Such methods can help significantly reduce fossil fuel, in addition to decreasing associated operating costs and pollutant emissions [8]. Among heat exchangers, regenerators (recuperators) are widely used in many industrial sectors, such as the cryogenic, metallurgical, chemical processing and glass industries. They are indeed attractive due to their wide range of temperature and compactness over which they can be used. For instance, a glass industry regenerator operates at temperatures of up to 1650K, whereas a cryogenic regenerator can operate close to a few Kelvin. A typical glass melting furnace consists of regenerators in which checkers of refractory bricks have been stacked in the regenerator chambers [9]. In one cycle the checker is heated up by flue gases, subsequently, in the following stage (20 – 30 min) the heat is transferred to combustion air. That means the furnace needs to be provided with 2 air regenerators minimum as illustrated in Fig. 1. When one regenerator works as an accumulator, another one works as a heater. It is called the reversal period. During the shifting process between the regenerators, the burners are changing too. It lasts 60 - 120 seconds. The reversal period should be as short as possible due to the cooling of the bath. In addition, since the regenerator of a glass melting furnace is undergone a cyclic operation, after certain cycles, it will reach a convergence condition in which the temperature of exit gas or bed temperature of regenerators is theoretically equal in its cycles.

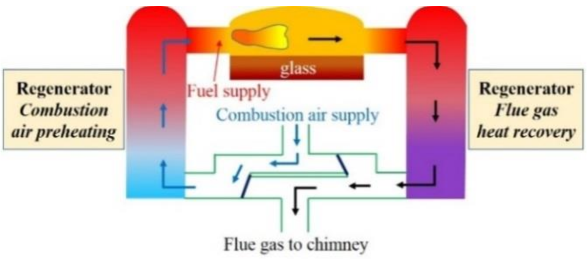

Figure 1. *A combustion system of a side-port regenerative glass furnace [10]*

A typical glass melting furnace consumes about 70–80% of the total thermal energy utilized in a glass plant. A well-designed regenerator can recover about 60 – 65% of the total input heat to the regenerator. Therefore, the efficient operation of a regenerator plays an important role in the economics of a glass plant operation. There is a great deal of previous literature on the general properties of thermal regenerators. In particular, performance estimations for regenerators which were discussed by Schack [11] and Trinks [12]. The transient temperature profiles of the flue gas, packed bed, and air during the heating and cooling processes at various switching times were investigated by Lin et al.[13]. Furthermore, increasing the switching time will decrease the regenerator's thermal effectiveness. In addition, the effect of variation in the exit temperature and heat recovery for packed-bed regenerators is studied by Yu et al. [14] using a one-dimensional transient numerical model. They reported that the fuel type should be considered as designing of a regenerative furnace in which a double preheating furnace should be used for the fuels with low- or medium-caloric value while the fuels with high-caloric value are suitable for both single or double preheating furnaces. In Zarrinehkafsh and Sadrameli's study [15], a model was recommended to investigate the thermal recovery efficiency in regenerators involving spherical particles, assuming constant gas velocity throughout the regenerator. Their analyses were based on the experimental data obtained for a regenerator packed with alumina balls. They concluded that the difference between the effectiveness results could be firstly due to the experimental errors in the parameter measurements and secondly due to the heat losses from the main bed which had not been taken into account in the mathematical model. The deposition of exhaust gas components in the regenerators of glass furnaces was investigated by Beerkens and Waal [16] using simulated flue gases from glass furnaces on a laboratory scale. Performance analysis for a glass furnace regenerator is also investigated by Vishal Sardeshpande et al. based on the regenerator's mass and energy balance [17]. The regenerator's outlet temperature is a key parameter in evaluating the regenerator performance. They reported that the model can be used for predicting performance deterioration factors like blockage, air leakage, and wall losses. A similar approach was also studied by Sardeshpande et al. [18]. They concluded that their simulation model can be used to calculate the energy performance of a given furnace design.

In general, an accurate estimation of the heat transfer characteristics and performance of regenerators when designing a glass melting furnace for the glass manufacturing industry is important and useful for designing engineers. Although there have been some studies on simulation on heat transfer characteristics of regenerators for industrial applications, these analytical models are quite complex. As a result, in the industry, a simple model has been usually preferred by designing engineers during a design process because it can provide a quick solution and low simulation cost. Therefore, in this work, we propose a simple analytical model which is capable of predicting the temperature distribution of solid phase and exit gas temperature variations as well as a regenerator's performance for different furnace loading conditions. Based on initial parameters obtained from measurements in a real system, a comparison between the numerical results and experimental data is made. It is expected that the model can be also extended for the prediction of performance of different packed regenerators.

2. REGENERATORS IN THE GLASS FACTORY

The general arrangement of regenerators of the glass melting furnace in the glass factory located in Taichung, Taiwan (Gin Jye Ming Glass Co., Ltd) is presented in Fig. 2. The system has two regenerators through which hot and cold air flow alternately. A whole of each regenerator consists of six checker chambers with a total size 7.8×5.6×17m (length × width × height). As the combustion products pass through one chamber, the checker bricks absorb heat from the combustion gas and increase their temperature. This is called the "heating process" or "hot period". The flow of air is then adjusted so that incoming combustion air passes through the hot checker, called the "cooling process" or "cold period", which transfers heat to the combustion air entering the furnaces. Two chambers are used, thus while one is absorbing heat from the exhaust gases, the other is transferring heat to the combustion air. The direction of air flow is altered every 20 min. In other words, the operating time of the cooling process or heating process is 20 min. Thebrick is made of 97% MgO material and its properties are presented in Table 1 (obtained from the supplier). The shape of each block and its arrangement inside each regenerator are also depicted in Fig. 3.

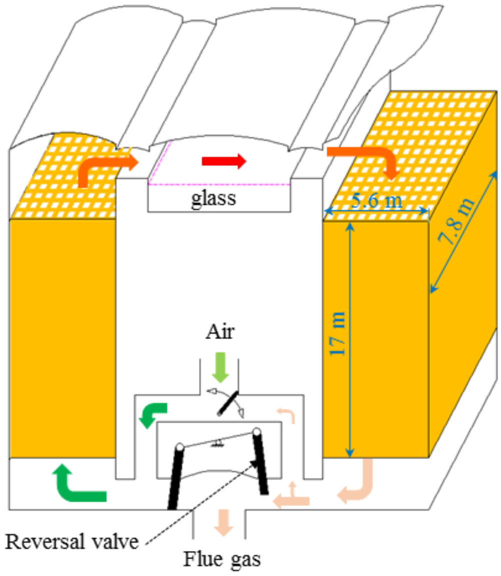

Figure 2. *Diagram of the regenerative glass melting furnace*

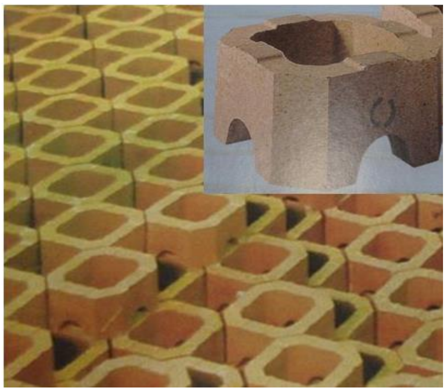

Figure 3. *The brick shape andits arrangment inside regenerator of the glass melting furnace*

Table 1. *Properties of the bricks in the bed of the regenerator*

Flue size (mm)	140
Height (mm)	150
Thickness (mm)	40
Density (kg/m^3)	1742
Heat capacity (J/kg.K)	961.4

The glass melting furnace requires high operating temperatures (about 1240°C–1315°C at the top of regenerators) which are recorded by furnace monitoring systems. Monitors are usually present at various positions in the furnace roof and in the body of the molten glass. The commonly recorded temperatures are measured by thermocouples which are located at the crown and bottom of the checker bed. The furnace's operating parameters, which include air flow rate, fuel flow rate, and oxygen flow rate are measured by flow meters and also reported on the monitoring screen. In order to use for our simulation model, these values are averaged over a one-month duration of the experimental program. The operating parameters and fuel properties are presented in Table 2.

***Table 2.** Furnace operating conditions*

Mass flow rate	
Fuel, $kg.hr^{-1}$	2100
Combustion air, $kg.hr^{-1}$	25749
Oxygen, $kg.hr^{-1}$	556.6
Fuel components,%	
C	88
H	11
Density of oil, $g.cm^{-3}$	0.97

In addition, for comparison between numerical and experimental results, the measured data including inlet and outlet gas temperature are collected during the normal operation of the real glass melting furnace in the Gin Jye Ming Glass Co., Ltd. Originally, there were five thermocouples located at each inlet or outlet port of the regenerators. Then, the averaged values are used for comparison.

3. ANALYTICAL MODEL

For our study, a one-dimensional transient model is used to simulate the thermal exchange between the regenerators and the fresh incoming air or the flue gas. The basic model assumptions are the following:

1. The inlet flow condition is fixed;

2. Mass flow rate does not change with time;

3. The thermal accumulation of gas in the regenerator is ignored;

4. The pressure is kept at ambient levels;

5. The heat transfer coefficient is constant during the periods;

6. Conductive and irradiative energy exchange mechanisms are negligible compared to convective heat exchange;

7. Adiabatic systems towards the surroundings;

Based on the arrangement of the regenerators in the glass factory, the cross-sectional area is presented in Fig. 4. In comparison, the structure of these regenerators is similar to the packed bed of regenerators studied by Mario Amelio and Pietropaolo Morrone [19]. Referring to the control volume, the thermal exchange in a packed bed with a forced fluid flow along x-direction can be described as [20]:

$$m_a \times C_{pa} \times \frac{dT_a}{dx} = h_{ib} \times p \times (T_b - T_a) \quad (1)$$

$$m_b \times C_b \times \frac{dT_b}{dt} = m_a \times C_{pa} \times (T_a - T_b) \times \left(1 - e^{-\frac{L_b}{\lambda_{ib}}}\right) \quad (2)$$

Where m_a, m_b, C_{pa}, and C_b are air mass flow rate, mass of brick bed, specific heat capacity of air, and specific heat capacity of brick bed, respectively. T_a and T_b are the temperatures of the air (or gas) and the brick bed while h_{ib} and p are convective heat transfer coefficient and the heat transfer area per unit length along the direction of gas flow.

$$p = \frac{A_p}{L_b}$$

$$\lambda_{ib} = \frac{h_{ib} \times p}{m_a \times C_{pa}}$$

A_P is the total heat transfer area between the gas flow and the bed, L_b is the length of the brick bed.

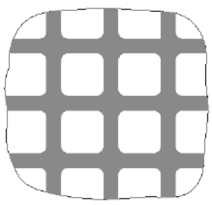

Figure 4. *Cross-sectional area of the structure bed of the regenerators*

In this work, the Eqs. (1) and (2) are numerically solved in MATLAB. In order to do that, the one-dimensional transient model is converted into a zero-dimensional transient model by assuming that the brick beds can be divided into many layers in the x-direction where the length of each layer is small enough so that the temperatures of gas and brick of each layer are only dependent on time. The layer number is initially studied to ensure that the simulation results do not depend on the number of the divided layers. With the assumption, the temperatures of exit gas and brick of each layer are presented by the following equations (3) and (4).

$$T_b - T_{ia} = \left(T_{b0} - T_{ia}\right) \times e^{-\frac{t}{\tau_b}} \quad (3)$$

$$T_{ea} - T_b = \left(T_{ia} - T_b\right) \times e^{-\frac{L_b}{\lambda_{ib}}} \quad (4)$$

where: $\dfrac{1}{\tau_b} = \dfrac{m_a \times C_{pa}}{m_b \times C_b} \times \left(1 - e^{-\frac{L_b}{\lambda_{ib}}}\right)$

$T_b$is the varying temperature of the solid phase (brick bed) after exchanging with the inlet gas at temperature T_{ia}. T_{ea} is the changing temperature of the exit gas in contact with the solid in a period of time *t*.

The choice of the heat transfer coefficient is a crucial factor. Indeed, various correlations present in the literature yield very different values. Therefore, in the present work, the model is a fixed packed bed assumed with a flow through a rectangular long duct, so h_{ib} is determined on the following basic interrelation [21].

$$Nu = \frac{h_{ib} \times D_h}{k} = 0.023 \times \mathrm{Re}_D^{4/5} \times \mathrm{Pr}^n \quad (5)$$

where: Pr = 0.6 ~0.7

$$\mathrm{Re}_D = \frac{u \times D_h}{\nu}$$

n = 0.4 for cooling process and n = 0.3 for heating process. u, D_h, and ν are air superficial velocity, hydraulic diameter of brick, and air viscosity.

For simulation and comparison, the initial parameters including air flow rate, fuel flow rate, and oxygen flow rate are taken from experimental data. The inlet temperatures in both heating and cooling processes are fixed at 200 0C and 1310 0C, respectively, in the simulation work. The properties of fuel and brick were based on suppliers.

In addition, the mass flow rates of air and flue gas in the “hot period” and “cold period”, respectively, are not similar. This is because the flue gas in the hot period consists of the mass of air, fuel, and oxygen, whereas the cold period consists only of the air mass. Based on experimental results, the combustion process of the burner is considered a complete reaction. The reaction equation is presented in Eq. (6) as follows.

$$CH_{1.5} + 1.2632(O_2 + 3.76N_2) + 0.1118O_2 \rightarrow CO_2 + 0.75H_2 + 4.7496N_2 \quad (6)$$

Moreover, the heat capacities of air and flue gas are considered as temperature dependent variables. The air is based on the heat capacity values of ideal gases given in the textbook [11] at different temperatures, from 300 – 1600K. A linear assumption of the relationship was made as shown in Fig. 5.

For the flue gas side, combustion products include CO2, H2O, and N2. Therefore, the heat capacity of flue gas is calculated based on the mole percentages of CO2, H2O, and N2 in the combustion products.

The Eqs. (7), (8), and (9) present the temperature dependence of heat capacities of each component [22].

$$C_{pCO_2} = -3.7357 + 30.529\theta^{0.5} - 4.1034\theta + 0.024198\theta^2 \quad (7)$$

$$C_{pH_2O} = 143.05 - 183.54\theta^{0.25} + 82.751\theta^{0.5} - 3.6989\theta \quad (8)$$

$$C_{pN_2} = 39.060 - 512.79\theta^{-1.5} + 1072.7\theta^{-2} - 820.4\theta^{-3} \quad (9)$$

where: $\theta = \frac{T}{100}$(K).

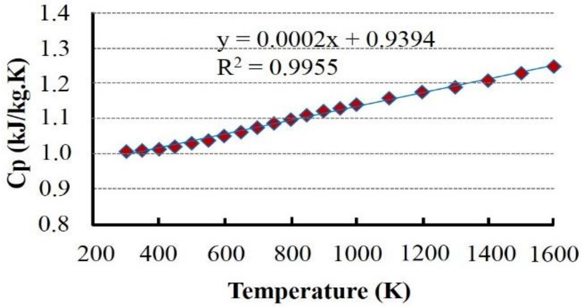

Figure 5. *Relationship between specific heat capacity and temperature of air*

A key variable from the viewpoint of thermal process characteristics is the thermal recovery efficiency, which represents the ratio between the energy transferred to the solid phase from the gas and the maximum energy that the regenerator is able to accumulate [23]. The gas energy would release to the solid if it were to leave the accumulator at the same temperature Ti at which the cold gas enters the "cold period":

$$\eta_{rec} = \frac{\int_{0}^{\tau}\int_{T_C}^{T_o} C_{pa}(T)\times dT \times dt}{\tau \int_{T_C}^{T_i} C_{pa}(T)\times dT} \qquad (10)$$

where T_o is the instantaneous temperature of the gas exiting from the regenerator and T_c is the inlet hot gas temperature of the "hot period". More often, as shown below, temporally averaged values are conveniently used for limited periods of time.

4. RESULTS AND DISCUSTION

Figures 6 and 7 show both temperature distributions of the solid phase and temperature profiles of the gas phase inside the regenerator versus location along the height of the bed (as illustrated in Fig. 2) at the convergent condition in the cooling and heating processes, respectively. The inlet temperatures are fixed at 200 °C and 1310 °C in the cooling and heating processes, respectively. Remember that, at the convergent state, the temperature profiles of the bed and gas will be repeated every cycle. It can be seen from the figures that there is a slight variation in gas and solid matrix temperatures along the packed bed during heating and cooling processes in the 20min of the cycle. The largest variation in temperature of the solid phase occurs on the first and last layers in the cold and hot periods, respectively. For the cooling process, the temperatures of the bed and combustion air will decrease over time due to the heat transfer from the bed to the air. As the heat storage of the bed decreases, the heat gains of the air decline, too. In contrast, the temperatures of the bed and flue gas get higher over the heating process. This is because the bed absorbs the heat from the higher temperature gas. It is clear that the heat transfer characteristics of the regenerator can be obtained from the analytical model at the convergent condition. The model will be validated by comparing the numerical and experimental results.

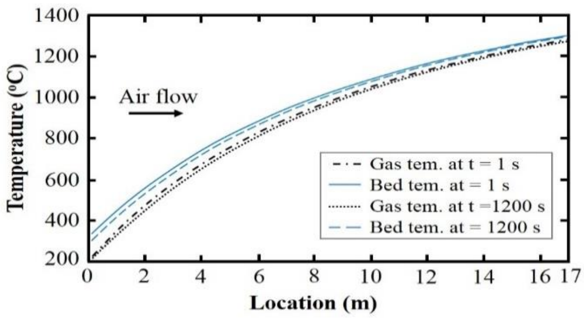

Figure 6. *Variation of gas and bed temperatures during cooling process at convergent condition*

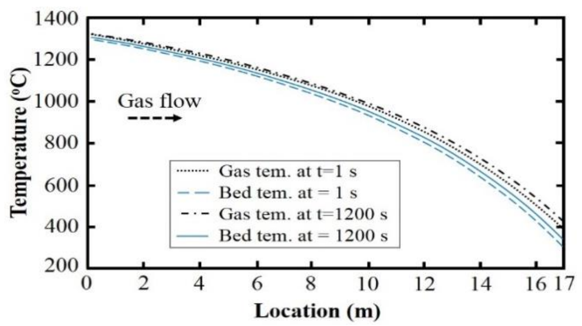

Figure 7. *Variation of gas and bed temperatures during heating process at convergent condition*

The first comparison made between the experiments and simulation results was the outlet air temperature in the heating process, as shown in Fig. 8. The trend of the experimental and numerical results are in a good agreement after 320s, whereas a large discrepancy is observed at the beginning. This phenomenon is caused by the lapse in time required for the real system to alternate the gas's direction. Hence, within the first 60s, the measured temperatures were actually inlet temperatures of the previous cold period. Following, the outlet temperature significantly increased in the next 3 min due to the increase in the flow rate of the flue gas from zero to stable flow. This phenomenon is similar to the measured inlet temperature profile at the inlet port as shown in Fig.9.In particular, the real inlet temperature in the heating process increased from about 1270°C at the beginning of the process to 1300°C after 320 s while it is fixed at 1310°C for the simulation case. It is obvious that the outlet temperatures of the experimental and numerical results are similar when the inlet temperatures for both cases are comparable.

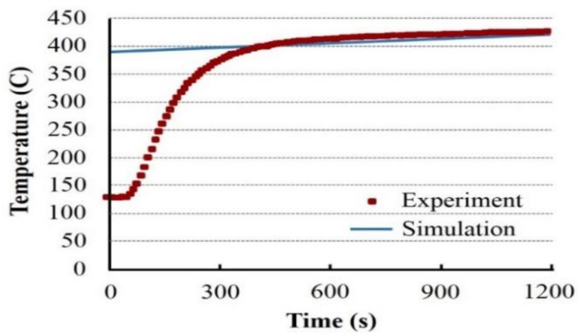

Figure 8. *Outlet gas temperature variation in the heating process*

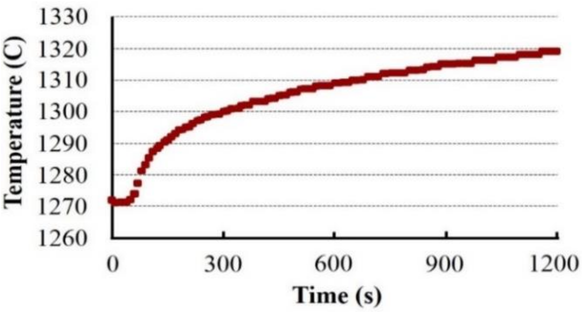

Figure 9. *Themeasured inlet temperature profile in the heating process*

Figure 10 shows the exit air temperature of the simulation model and the experiments in the cooling process. As observed, the overall trend of the outlet temperatures for both cases decreases between 0 to 1200s (20-minute cycle of the "cold period"). The measured values considerably decrease from 1320°C to 1287°C, especially at the early stage (~ 240 s),while the simulation temperature gradually lessens. The reason causing the observed variation between the numerical and experimental results is the difference in the inlet temperatures of the cooling process. Similar to the heating process, the inlet temperature in the simulation is fixed at a constant value (200 °C) while it is varied in the real system during the cooling process as seen in Fig. 11. In fact, in the first 60 s, the inlet temperature for the cooling process of the real system is the outlet temperature of the previous heating process (~ 450 °C). After that, more and more combustion air comes and mixes with the outlet temperature that makes the inlet temperature significantly decrease in the next 260 s. Finally, when the flow rate of the incoming combustion air becomes stable, the inlet temperatures in the real system and simulation are similar. Accordingly, there is a relatively good agreement in outlet air temperatures after 320 s in the cooling process.

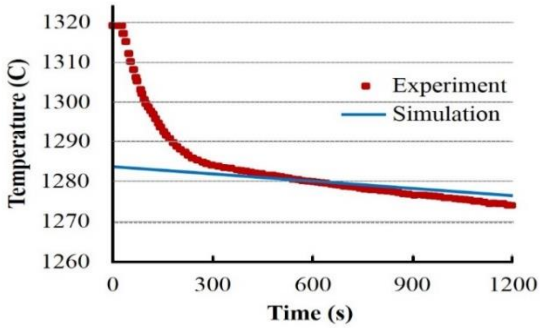

Figure 10. *Outlet gas temperature profile in cooling process*

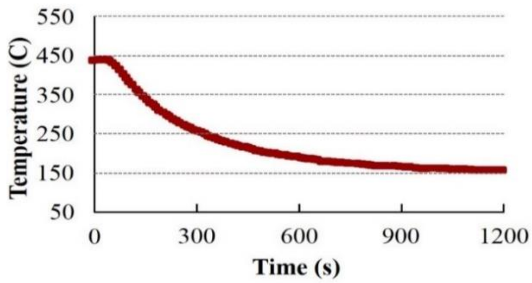

Figure 11. *Inlet gas temperature profile in cooling process*

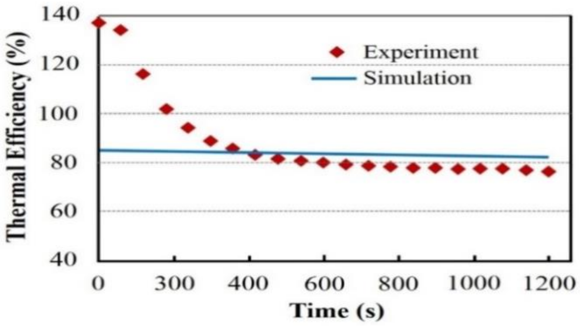

Figure 12. *Thermal recovery efficiency of regenerator for simulation and experiment*

The comparison of the thermal recovery efficiency between numerical and experimental results is presented in Fig. 12. It is found that their thermal recovery efficiencies lightly decrease during the 20-minute cycle although there is an unusual trend observed in the measurement results. The significant drop in the efficiency at the first 320 s of the cycle is due to the real working conditions of the regenerator that requires time for switching the gas direction between heating and cooling processes as discussed above. However, after 350s, the change in both simulation and experimental efficiencies is very similar. The difference in the thermal recovery efficiencies is only 4.3%, indicating that the analytical model can be used to estimate the thermal recovery efficiency of large regenerators in a real glass melting furnace. It is believed that no heat loss through the walls of the bed involved in the analytical model could be the main reason for a little higher recovery efficiency of the numerical results compared to that of the experimental results [15].

5. CONCLUSIONS

A mathematical model for regenerators used in glass melting furnaces has been developed and numerically solved. The influences of the mass flow rates and specific heat capacities of the air and combustion products which are temperature-dependent on heat transfer characteristics between gas and solid phases were also considered in the model. The analyses revealed that when the initial conditions are similar, a good agreement between calculation and measurement is seen as exhibited by an error value of only 4.3% thermal recovery efficiency. Moreover, it is believed that the operating principle of the system is a primary reason for the major differences in exit gas temperature between the theory and practice. However, the unsatisfactory issue only happens in a short time and does not have much effect on the final results. Therefore, the proposed model in this study is a believable representation for predicting the heat transfer characteristics of regenerators in the glass or steel industry.

Nomenclature

L_b = length of each bed layer (m)

C_b = specific heat of the brick bed (J/(kg K))

C_{pa}= specific heat of air (J/(kg K))

D_h = Hydraulic diameter of brick (m)

h_{ib} = convective heat transfer coefficient (W/m^2.K)

k_a= thermal conductivity of air (W/m .K)

m_a = air mass flow rate (kg/s)

m_b = mass of brick bed (kg)

Nu = Nusselt number (-)

p = the heat transfer area per unit length along the direction of gas flow

Pr = Prandtl number (-)

Re = Reynolds number (-)

T_a = gas temperature (K)

T_{ia} = Inlet gas temperature (K)

T_b = Bed temperature (K)

u = air superficial velocity (m/s)

Greek Symbols

ν = air viscosity (kg/(m s))

ρ = air density (kg/m^3)

ρ_w = brick density (kg/m^3)

t = cycle duration (s)

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