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PERFORMANCE MEASUREMENTS OF A BALL TYPE HIGH-CYCLE REGENERATIVE BURNER SYSTEM (HRS) AT SSAB TUNNPLÅT AB

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Key words: regenerative burners, burners, reheating furnaces, temperature measurements, iron and steel industry

SUMMARY

In accordance to a project called “HTAC tests at KTH”, this report presents measurements carried out in a ball type HRS at SSAB TUNNPLÅT AB. These measurements aiming in figure out energy balance, temperature efficiency and heat recovery rate for various cycling time. The results were encouraging. The temperature efficiency and heat recovery rate during a normal cycling time, 60 seconds, were in order of 87.7% ±2% and 75.9% ±2.3% respectively.

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INNEHÅLLSFÖRTECKNING

1. Introduction

The major advantages of High Temperature Air Combustion HiTAC technology are to provide a significant energy savings, thus CO2 and NOx emission reduction compared to any conventional combustion. Furthermore, HiTAC increases the production rate and contributes in enhancing the product quality in most industrial applications due to heat transfer enhancement and more uniform temperature distribution inside the furnace.
High-cycle Regenerative burner systems (HRS) are used in the application of HiTAC combustion. These burners are capable to supply combustion air with temperatures above the self-ignition temperature of the fuel. This can be achieved by using the regenerative heat exchanger that recovers the heat from the flue gases and use it to increase the temperature of the combustion air. The performance of the heat exchanger is an important factor to achieve the desired temperature of the combustion air. Energy balance measurements must be carried out to evaluate the performance of the heat exchanger.
Application of the HiTAC in different types of furnaces in Japan as reported by Yasuda et al. (2000) and Mori (2001) resulted also in big industrial interest in European countries. Simulation and measurements of both types of HRS systems, ball and honeycomb, has performed by Hasegawa et al. (1997) and Tada et al. (1991). However, most of the measurement carried out so far has ignored the effect of the dynamic behavior of both the HRS and the measuring equipment and some of them lacks for uncertainty analysis.
One major difficulty in carrying out these measurements is the dynamic behaviour of both the measuring system and the operational nature of the burning systems caused by switching devices of the regenerators. Therefore, it is important to recalculate the measured parameters in order to determine the real values at any given time to get a high quality data. Another measuring problem, concerning temperature measurements, is the effect of thermal radiation from the walls on the thermocouples. Hence, thermal equilibrium cannot be achieved with the medium whose temperature is to be measured and resulting in numerous errors in measurements.
The aim of this report is to present measurement results aiming to figure out the performance of a ball type HRS system that been performed in the 4th and 5th of June 2002 at SSAB TUNNPLÅT, Sweden. They are installed at one preheating furnace and formed only a small portion of the total heating capacity of the furnace. Energy balance, temperature efficiency and heat recovery rate of HRS has been figured out at various cycling time and furnace temperature while taking into account the dynamic effect of the system.

2. Experimental apparatus

2.1 HRS system description

HRS burner system was mounted in the dark zone, bottom of Zone no. 2, of the Fce 302 preheating furnace as seen in figure 1. The furnace is preheating steel slabs with a total capacity of 300 T/hr.
The model of the HRS system is RCB-HM-40 manufactured by CHUGAI RO CO. LTD – Japan. The system has a pair of burners, where one is firing and the other acts as the regenerator media, each incorporating a ceramic heat exchanger containing 670 kg of 14 mm diameters ceramic balls. Regenerative material specifications are tabulated in table 1. Moreover, The heating capacity of HRS burner system is 2097 KW and forms only small portion of the total heating capacity of the furnace, which is 170 MW.
Heavy fuel oil no. 5 is used in this firing system. Table 2 shows the results of a fuel property test made in 2002-04-08. Furthermore, The standard cycling time is 60 seconds but can be set for different intervals.

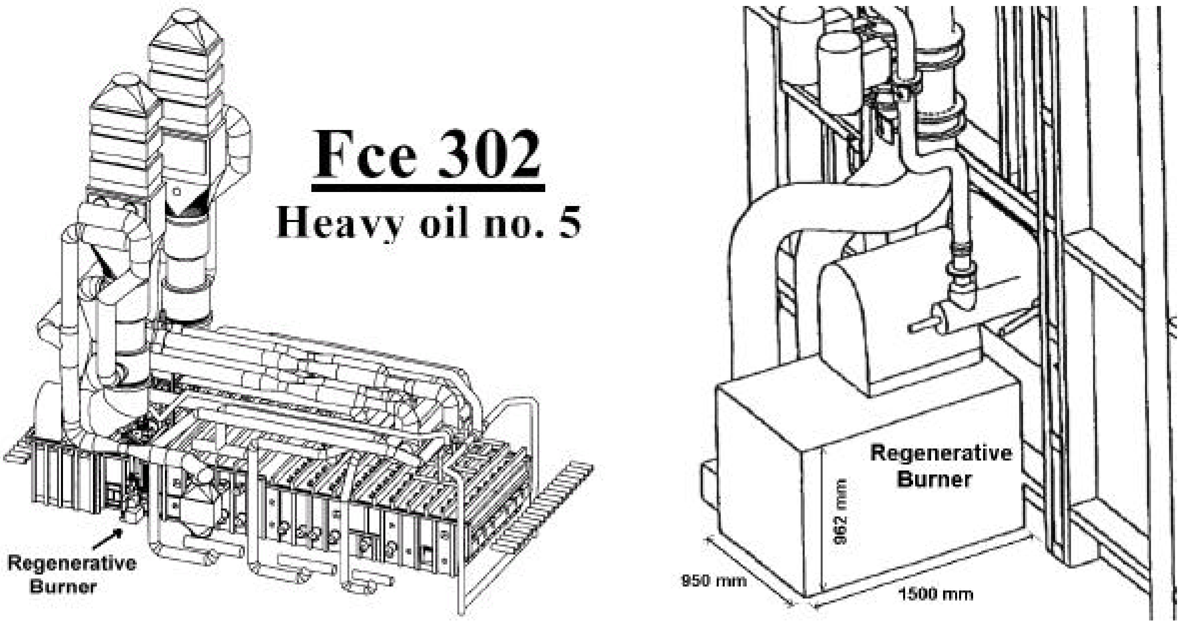

Figure 1: 3D-drawings of the HRS system and Fce 302 preheating furnace.

Table 1: Specifications of ceramic ball regenerative material

Material of balls	Alumina (AL2O3: more than 99.7%)	
Bulk density of balls	Approx. 2160	kg/m3
Specific heat	0,245 (at 420°C)	Btu/lb.F
Specific heat	0,294 (at 1340°C)	Btu/lb.F
Specific heat	0,302 (at 1880°C)	Btu/lb.F
Thermal conductivity	3,0 (at 200°C)	Btu-in/ft2.hr.F
Thermal conductivity	5,8 (at 600°C)	Btu-in/ft2.hr.F
Thermal conductivity	8,7 (at 1000°C)	Btu-in/ft2.hr.F
Thermal conductivity	11,6 (at 1400°C)	Btu-in/ft2.hr.F
Average diameter of ball	(13 - 15) average 14	mm
Nominal diameter	13 (1/2)	mm (inch)
Ball room size	700Wx1250Lx510H	mm

Table 2: Constituents and properties of heavy fuel oil no. 5.

Fuel content, mass fraction (%)		**Some fuel properties**		
Carbon	87,1	Density (at 15°C)	0.9263	kg/L
Hydrogen	12,3	Viscosity (at 80°C)	42,78	mm^2/s
Sulphur	0,36	Viscosity (at 50°C)	184,7	mm^2/s
Nitrogen	< 0,3	HHV	45,38	MJ/kg
Oxygen	< 0,3	LHV	42,77	MJ/kg
		LHV	11,01	KWH/L
		LHV	11,89	KWH/kg

2.2. Measuring techniques

The cycling time of the HRS burners can be selected. Tests were being performed at three different cycling times. In each case, flow and temperature measurement signals of flue gases and combustion air in both sides of the regenerators were being logged every 250 ms interval. The measurements were being performed on both burners, the southern and northern burners.
For temperature measurements in the hot side of the regenerator, a double shield pyrometer was made on site to fit in the measuring hole. It was mounted on the southern burner. Flue gas composition of flue gases were measured and logged as well.
Digital signals of the switching valve and fuel oil shut off valve were logged as well together with the fuel flow rate. It was found that there is 7 seconds delay time for the fuel valve to open after switching over between burners. The actual cycle time was also found 7 seconds later than the set cycling time. Which means that the actual cycle time is 67 seconds when it is set to 60 seconds.
To insure same operating conditions in furnace, signals from one thermocouple[1] mounted at the inner wall were logged as well. In fact, it was difficult to have the same temperature through out the tests due to production requirements. As a result, each case was repeated 2 to 3 times to ensure the same operating condition as possible for the three cases. Table 1 shows the wall temperature in each case.

2.3. Experimental conditions

The measurements were performed with three different cycling times, 50, 60 and 70 seconds. The operating conditions of these three cases are shown in table 3.

Table 3: Operational conditions in the three studied cases

Cycling Time (s)	Fuel flow rate (L/h)	Fuel type	Combustion air flow rate (Nm^3/h)[*2]	Furnace wall Temperature[1] (°C)	Combustion air temp T_{ai} (°C)	Combustion air temp T_{ao} (°C)
50	194.8	Fuel oil # 5	2520	970	95.8	962.5
60	195.1	Fuel oil # 5	2518	950	90.9	939
70	195.6	Fuel oil # 5	2521	1000	95.7	1006

3. Data Analysis Methodology

3.1 Dynamic response consideration

One major difficulty in carrying out these measurements is the dynamic behaviour of the heat exchanger, causing all parameters of which are to be measured vary with time. In practice, no sensor can respond instantaneously to a change in the input signal but requires some time to reach its steady state value because of its time constant. This delay is due to the inertia of the sensor, for instance, thermal inertia in case of a temperature sensor. Hence, the output readings of sensors do not represent the actual values of parameters in which are being measured, thus resulting in numerous errors.

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[1] Signals from a thermocouple mounted at the inner wall 2 meters away from the south burner. This thermocouple has tag no. DB-2TT02-1S. It was used as an indication of steady state throughout one test and similar conditions between tests.

[*2] Normal conditions are at T= 273.15 ^{o}K and P=101325 pa (absolute). This is valid in the whole report.

To obtain accurate measurements as possible, a certain measures had to be considered regarding both, the measuring instruments and data analysis. Regarding measuring instruments, two major measures had to be considered. First, all measuring instruments were have as small time constant as possible. Furthermore, the time constants of all sensors were figured out experimentally and before starting energy balance calculations. After that, a special frequency response analysis was performed to compensate with the time constant data, which was measured previously.
Evaluation of measured data from non-steady state measurements was carried out when the real value of the measurement had to be calculated taking into account the inertia of the measuring device. The real value of the measurement was calculated as below.

$$U_R = K\frac{dU}{dt} + U_M \qquad (1)$$

Where U_R is the actual value of the input signal, dU and dt are the change of input signal and the change in time respectively, U_M is the measured signal and K is a constant depending on the inertia of the measurement system (Time constant). Several measurements were performed to determine the time constant for several thermocouples and other measuring devices. An example of what the measured and calculated temperatures looks like can be seen in Figure 2.

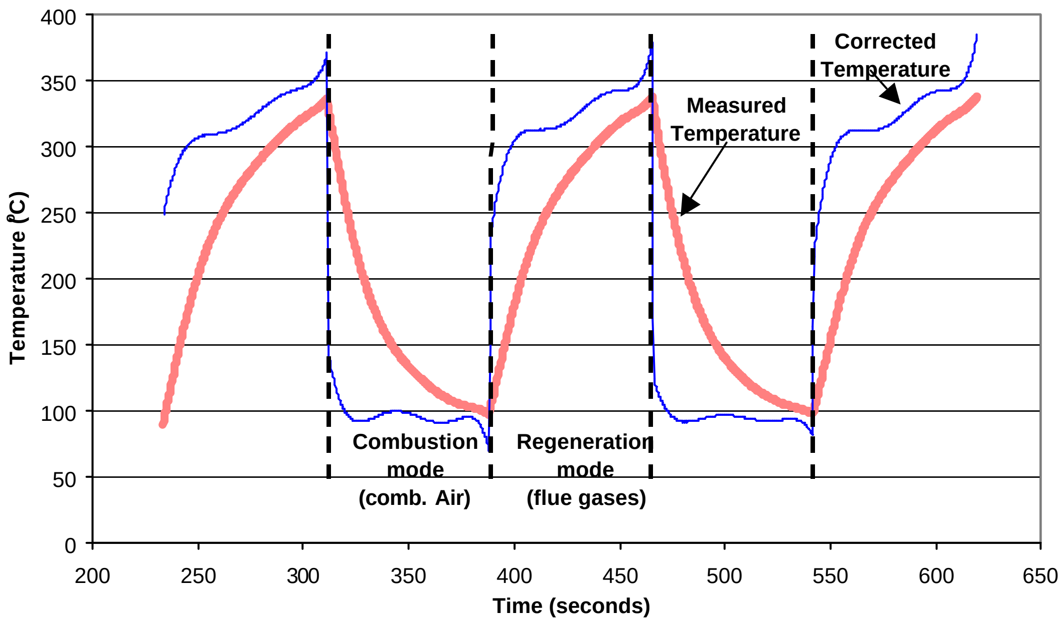

Figure 2: Measured and corrected values of combustion air and flue gases temperatures. The cycle time is 70 seconds.

3.2. Radiation effect on thermocouple consideration

Another measuring problem, concerning temperature measurements, is the effect of thermal radiation from the walls on the thermocouples. Hence, thermal equilibrium cannot be achieved with the medium whose temperature is to be measured and resulting in numerous errors in measurements. A suction pyrometer was used in this case to shield the thermocouple from radiation and increase the heat transfer coefficient between the medium of which to be measured and the thermocouple, thus avoiding a big amount of error and getting results close as possible to true values.

3.3. Energy balance and HRS performance parameters

Figure 3 shows the regenerative heat exchanger as an energy system. The system includes the ceramic balls, insulation and casing. The system time has been chosen to be two cycles; one cycle the burner operates in combustion mode and another for regeneration mode. The heat passes through the system boundary are the following:

- Surface radiation and convection losses: $\dot{Q}_L$
- Physical enthalpy in cold combustion air inlet: $\dot{Q}_{ai} = \dot{m}_{a\ a}\ Cp_{ai}\ T_{ai}$
- Physical enthalpy in hot combustion air outlet: $\dot{Q}_{ao} = \dot{m}_a\ Cp_{ao}\ T_{ao}$
- Physical enthalpy in hot flue gases inlet: $\dot{Q}_{gi} = \dot{m}_g\ Cp_{gi}\ T_{gi}$
- Physical enthalpy in cold flue gases outlet: $\dot{Q}_{go} = \dot{m}_g\ Cp_{go}\ T_{go}$

where $\dot{m}$, Cp and T are mass flow rate, specific heat and temperatures respectively. The subscripts $_a$, $_g$, $_i$ and $_o$ denotes combustion air, flue gases, inlet and outlet respectively.

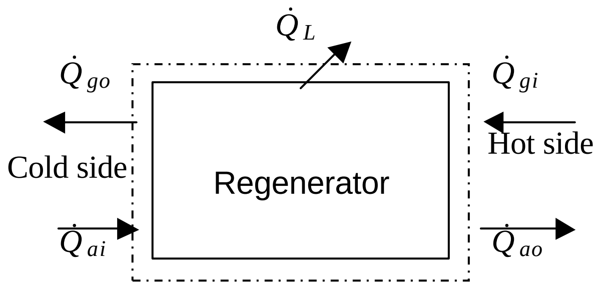

Fig. 3. Regenerative heat exchanger as an energy system

The *cold side temperature efficiency* h_{ta} is defined as below (Hasegawa et al. 1997):

$$h_{ta} = \frac{T_{ao} - T_{ai}}{T_{gi} - T_{ai}} \quad (2)$$

The *heat recovery rate* h_g is the ratio between the useful energy, energy recovered by combustion air, to the total available energy in flue gases.

$$h_g = \frac{\dot{m}_a \left(Cp_{ao} \cdot T_{ao} - Cp_{ai} \cdot T_{ai} \right)}{\dot{m}_g \cdot Cp_{gi} \cdot T_{gi}} \quad (3)$$

The *flue gas suction rate* is defined as the ratio between flue gases passes through regenerator, $\dot{m}_g$, to the total flue gases flow rate generated by the combustion process.

4. Results

Measurement results for the three different cycling times are tabulated in table 4 below. These measurements belong to the northern regenerative burner.

Table 4: Measurements results of the three studied cases

Measured parameter	**Values**			**Unit**
Cycle time	**50**	**60**	**70**	**[s]**
Furnace wall temperature	970	950	1000	[°C]
Flue gases temperature inlet, T_{gi}	1070.4	1058	1100.8	[°C]
Flue gases temperature outlet, T_{go}	286.8	279.8	305.1	[°C]
Combustion air temperature outlet, T_{ao}	962.5	939.0	1006.0	[°C]
Combustion air temperature inlet, T_{ai}	95.8	90.9	95.7	[°C]
Fuel flow rate, (instantaneous)	194.8	195.1	195.6	[L / hr]
Firing rate, (instantaneous)	2145	2148	2154	KW
Fuel flow rate (average)	171.5	176.4	178.2	[L / hr]
Air flow rate, $\dot{m}_a$	2520	2518	2521	[Nm^3/hr]
Actual O2 content in flue gas	6	6	6	[%]

It was reported that signals from the flue gases flow meter lacks very much for accuracy. In fact, one probable problem in measuring such media is the condensation of vapour. Therefore, these signals were ignored during calculations. However, The flue gases flow rate was then recalculated from the energy balance calculations because it was the only missing parameter in those calculations. Although regenerator surface temperature was not measured, the surface losses by radiation and convection was assumed to be 2% of the physical enthalpy in hot flue gases inlet with 100% uncertainty. This was the only assumption made that allows us to calculate the flow rate of flue gases passing the regenerator (table 5). Therefore, the calculated uncertainty of calculated mg was in order of only 2.5%. As a result of this assumption and from the combustion analysis (table 5), the flue gas suction rate was in order of 88.76, 86.54 and 91.14 for cycling time of 50, 60 and 70 seconds respectively.

In the same respect, temperature efficiencies (table 6) were in order of 88.9%, 87.7% and 90.6% and the heat recovery rates were in order of 75.5%, 75.9% and 74.7%. Figure 4 shows no effect of switching time, between 50 and 70 seconds, on both the temperature efficiency and heat recovery rate. In fact this reveals a consistency with the theoretical relation between switching time and heat transfer rate between the regenerator media and a fluid (Mochida et al. 2001). This relation for a ball type regenerator media is shown in the thin lines in figure 5 for τ = 150 sec. The bold lines in the same figure show the condition for honeycomb regenerator media, τ = 12 sec.

Table 5: Combustion analysis and specific he at calculation results

Parameter	**Values**			**Unit**
Cycle time	**50**	**60**	**70**	**[s]**
Theoretical air consumption	11.024	11.024	11.024	[Nm^3/kg fuel]
Theoretical dry flue gas	10.227	10.227	10.227	[Nm^3/kg fuel]
Theoretical CO2 content (CO2 max)	15.790	15.790	15.790	[%]
Flue gas CO2 content	11.266	11.266	11.266	[%]
Dry flue gas	14.318	14.318	14.318	[Nm^3/kg fuel]
Wet flue gas	15.684	15.684	15.684	[Nm^3/kg fuel]
Fuel oil density	0.926	0.926	0.926	[kg / L]
Flue gases flow rate (generated by combustion)	2830	2835	2842	[Nm^3/hr]
Flue gases flow rate (through regenerator) $\dot{m}_g$	**2512**	**2453**	**2590**	[Nm^3/hr]
Flue gas suction rate	**88.8**	**86.5**	**91.1**	[%]
Air factor, ?	1.371	1.371	1.371	[%]
Total water in flue gases	1.367	1.367	1.367	[Nm^3/kg fuel]
Flue gas moisture content	8.713	8.713	8.713	[%]
Wet flue gas density	1.299	1.299	1.299	[kg/Nm^3]
Combustion air density	1.292	1.292	1.292	[kg/Nm^3]
Specific heat of flue gases inlet, Cp_{gi}	1.294	1.298	1.303	[kJ/kg °K]
Specific heat of flue gases outlet, Cp_{go}	1.109	1.107	1.118	[kJ/kg °K]
Specific heat of combustion air inlet, Cp_{ai}	1.012	1.012	1.012	[kJ/kg °K]
Specific heat of combustion air outlet, Cp_{ao}	1.189	1.185	1.197	[kJ/kg °K]

Table 6: Energy balance, temperature efficiency and heat recovery rate results

Parameter	**Values**			**Unit**
Cycle time	**50**	**60**	**70**	**[s]**
Physical enthalpy in hot flue gases inlet, $\dot{Q}_{gi}$	1255.0	1215.6	1341.0	[KW]
Physical enthalpy in cold combustion air inlet, $\dot{Q}_{ai}$	87.7	83.1	87.7	[KW]
Physical enthalpy in cold flue gases outlet, $\dot{Q}_{go}$	288.2	274.2	318.8	[KW]
Physical enthalpy in hot combustion air outlet, $\dot{Q}_{ao}$	1035.1	1005.4	1089.5	[KW]
Net heat gained by combustion air, $\dot{Q}_{ao}$ - $\dot{Q}_{ai}$	947.4	922.2	1001.8	[KW]
Net heat extracted from flue gases, $\dot{Q}_{go}$ - $\dot{Q}_{gi}$	966.8	941.4	1022.2	[KW]
Surface losses, $\dot{Q}_L$ (assumed)	2.0	2.0	2.0	[%]
Surface losses, $\dot{Q}_L$ (assumed)	19.4	19.1	20.4	[KW]
Heat recovery rate, h_g	75.5	75.9	74.7	[%]
Air side temperature efficiency, h_{ta}	88.9	87.7	90.6	[%]

It is slightly difficult to see the trends in figure 4 not only because of the uncertainty of these results but also because of the small differences in furnace wall temperature of the three cases. Table 7 presents the uncertainties of these results in both absolute and relative values. Uncertainty analyses were being carried out in accordance to Holman (1994), chapter 3.

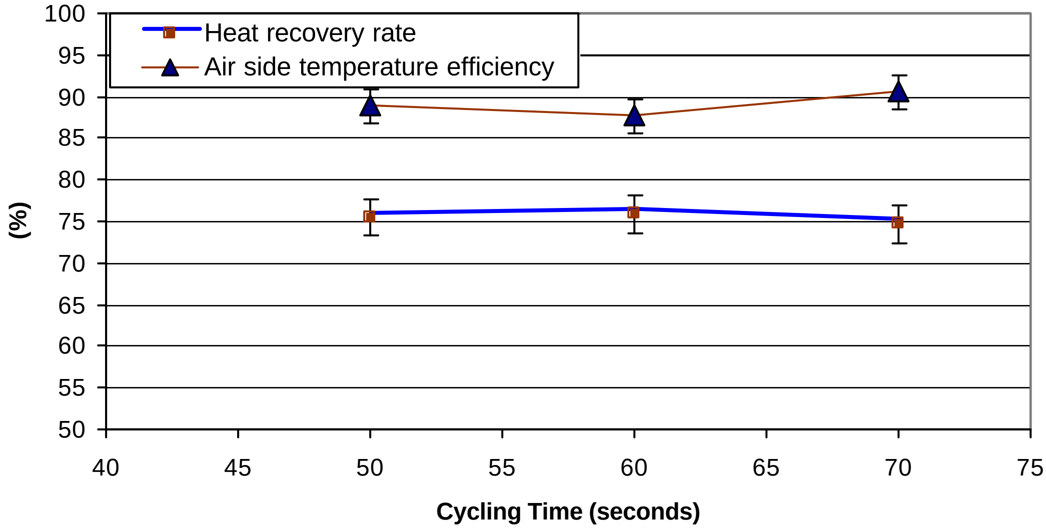

Figure 4: Temperature efficiency and heat recovery rate vs. cycling time.

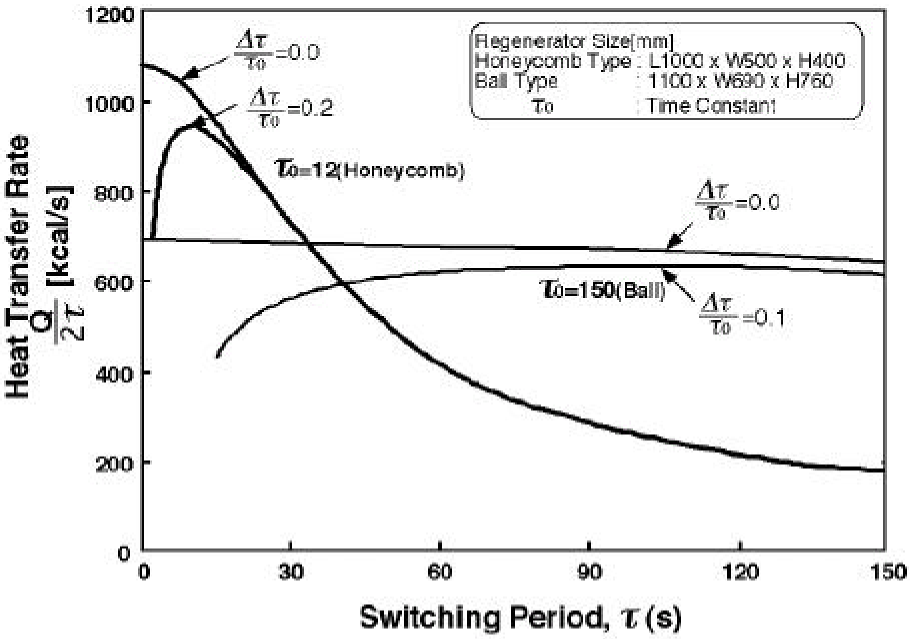

Figure 5: The effect of switching period on heat transfer rate (Mochida et al. 2001).

Table 7: Uncertainty Analysis for the results of the three studied cases

Parameter	**Relative**	**Absolute values**			**Unit**
Cycle time		**50**	**60**	**70**	**[s]**
Uncertainty in air flow rate, $U(\dot{m}_a)$	1.0%	25.2	25.2	25.2	[Nm³/hr]
Uncertainty in flue gases flow rate, $U(\dot{m}_g)$	2.5%	62.8	61.3	64.8	[Nm³/hr]
Physical enthalpy in cold air inlet, $U(\dot{Q}_{ai})$	2.4%	2.1	2.1	2.1	[KW]
Physical enthalpy in hot air outlet, $U(\dot{Q}_{ao})$	1.3%	13.5	13.2	14.1	[KW]
Physical enthalpy in hot flue gases inlet, $U(\dot{Q}_{gi})$	2.6%	33.0	32.0	35.2	[KW]
Uncertainty in surface losses, $U(\dot{Q}_L)$ (assumed)	100%	19.4	19.1	20.4	[KW]
Uncertainty in heat recovery rate, $U(h_g)$	3.0%	2.3%	2.3%	2.2%	[%]
Uncertainty in temperature efficiency, $U(h_{ta})$	2.3%	2.1	2.1	2.0	[%]

5. Conclusion

The effect of cycling time on the performance of the HRS has been measured. It is certain that the heat recovery ratio and temperature efficiency are almost not affected with cycling time for this type of regenerator. Moreover, results at a standard cycling time, 60 seconds, were encouraging. The temperature efficiency and heat recovery ratio were in order of 87.7% ±2% and 75.9% ±2.3% respectively at standard cycling time.

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