Available online at ejournals.uofk.edu

UofKEJ Vol. 1 Issue 1 pp. 50-57 (June 2011)

UNIVERSITY OF KHARTOUM ENGINEERING JOURNAL (UOFKEJ)

Influence of Airflow Leakage on Rotary Regenerator Performance Factors

W. Kisha[1], K. N. Abdalla[1], A. A. Rabah[2*]

[1]*Department of Mechanical Engineering, Faculty of Engineering, University of Khartoum, Khartoum, Sudan. (E-mail:niji84@hotmail.com; E-mail:kamalnasr@yahoo.com)*
[2*]*Department of Chemical Engineering, Faculty of Engineering, University of Khartoum, Khartoum, Sudan (E-mail:rabahss@hotmail.com)*

Abstract: This work investigated the influence of airflow leakage on the regenerator performance factors of effectiveness (ε), airflow capacity (C^*), matrix capacity (Cr^*) and number of transfer unit (*NTU*). A sophisticated experimental setup that facilitates precise measurement of apparent airflow temperature, rate and pressure drop is used. The measurements covered a wide range of rotational speeds (0 - 24 rpm), regeneration temperatures (40 – 70 °C) and airflow rates (160-500 kg/h). The range of apparent performance factors is for C^* (0.95-1.00), Cr^* (0.21-62.53), *NTU* (3.5-12.76) and ε (0.17-0.78). Significant difference between the real and apparent performance factors was found even under small leakage percent. The deviation of the real from the apparent effectiveness expressed as absolute average percent deviation (*AAPD*) exceeded 6. In over 300 data points *AAPD* for C^*, Cr^* and *NTU* exceeded 12, 7 and 7, respectively. The work also provided a relationship that facilitates the determination of the real airflow temperature and the rate from the apparent (i.e. measured) values.

Keywords: Rotary regenerator; Leakages; Performance factors

1. INTRODUCTION

Rotary regenerator (regenerator in short) is extensively used in heating, ventilation and air conditioning (HVAC) systems, particularly in solar desiccant air conditioning systems. Fig. 1 shows the location of regenerator in the solar desiccant air conditioning system. A comprehensive review in the regenerator characteristics such as matrix material of construction and geometry, channel size, number of channels per surface area is found in Rabah *et al.* [1]. Definition of regenerator performance factors of effectiveness (ε), airflow capacity (C^*), matrix capacity (Cr^*) and number of transfer units (*NTU*) and the operation conditions which include rotational speed, regeneration temperature, and airflow rates are found in ANSI/ASHRAE Standard 84-2008 [2].

Research on regenerator is focused mainly on three areas: (1) development of correlation (2) influence of the regenerator characteristic and operation conditions on performance factors and (3) leakage control and influence on the performance factors. The development of correlation is based on data obtained from numerical solution of the generator differential equations. The most widely known numerical solution using finite difference method is that of Lambertson [3]. He presented tables for ε as a function of C^*, *NTU*, Cr^* and $(hA)^*$. His solution covered $1 \le NTU \le 10$, $0.1 \le C^* \le 1.0$, $1 \le Cr^* \le \infty$, and $0.25 \le (hA)^* \le 1$. The numerical results of Lamberston [3] were used later by Kays and London [4] to derive a correlation for regenerator effectiveness. Ba´clic [5] has obtained approximate solution to the regenerator differential using Galerkin method. Ba´clic [5] presented his solution in a form of a correlation relating the effectiveness to two dimensionless groups [$\varepsilon = f(NTU,Cr^*)$]. His numerical solution covered a wide range of $1 \le NTU \le 500$ and $1 \le Cr^* \le 1$. Rabah and Kabelac [6] have solved the regenerator differential equations using analytical methods. Their solution produced a relationship between effectiveness and the dimensionless groups of C^*, *NTU* and Cr^*.

The influence of the regenerator characteristics and operation condition on effectiveness has been studied using both theoretical and experimental methods. Shang and Besant [7] have studied the influence of the matrix pore size on the regenerator performance factors. Rabah *et al.* [1] have investigated the influence of rotational speed, airflow rate and regeneration temperature on effectiveness.

The airflow carryover and pressure leakages have made the measurement of the real airflow rate and temperature quite challenging. In the recent years, significant progress has been made in the design of seals and clearance and pressure leakage was minimized [8]. However, Shah and Skiepko [9] have said that a significant reduction could occur in the thermal energy transfer to the fluid even with reasonably small (5% or so) leakages. Shah and Skiepko [9] and Shah

[10] have made extensive investigations on the influence of leakage on regenerator performance factors. They have provided numerical scheme for leakage estimation. Han and Kim [11] has made experimental investigation on the influence of leakage on regenerator performance factors. They have developed a correlation for carryover and pressure leakages as a function of rotational speed and ventilation power respectively.

The leakage of regenerator cannot be absolutely eliminated and its idealization definitely leads to false airflow rates and temperatures and subsequently the regenerator performance factors of ε, C^*, *NTU* and Cr^*. Hence, the airflow leakages at the inlet and outlet of both sides of the regenerator had to be quantified and accounted for in the energy balance. This work is aimed to investigate the influence of leakage on regenerator performance factors.

2. EXPERMENTAL SETUP

The experimental setup employed in this work is described in details in Rabah *et al.* [1]. Here, only a brief description of the experimental rig is given. Fig. 2 shows a schematic representation of the experimental setup. It consists of a regenerator (R), an electric heater (H) and two sets of air fans (F1) and (F2) installed in a wind tunnel. The wind tunnel is insulated using Ampflex foam of 10 mm thickness. The regenerator is of type RRS-K-C-16-505/655-400 manufactured by Klingenburg. Table 1 shows the specification of the regenerator.

3. MEASUREMENTS AND UNCERTAINITIES

Fig. 2 shows the location where airflow temperature, pressure and pressure drop are measured. Fig. 3 shows the location of the airflow temperature measurement at the exits of the regenerator. The volumetric airflow was measured via an orifice plate. The orifice plate and its location were constructed in accordance with the DIN EN ISO 5167-2 standard. The pressure drop across the orifice plate is measured using a differential pressure transmitter. All sensors are calibrated and the combined uncertainty U is determined as:

$$U = \sqrt{U_{inst}^2 + U_{cal}^2 + U_{rand}^2} \qquad (1)$$

where U_{inst} is the instrumentation error, U_{cal} is the calibration error, which is the error of the reference device and U_{rand} is the random or precision error which is defined in accordance with Deutsche Kallibriendienst (DKD) as:

$$U = \frac{t_{\lambda,95\%}\sigma}{\sqrt{N}} \qquad (2)$$

where t is the student test at a 95% confident interval, σ is the standard deviation, N is the number of data points and λ is the degree of freedom $= N - 1$.

Table 1. Generator specifications

Wheel mass (kg)	7.5	Wheel length (m)	0.2
Wheel diameter (m)	0.4	Channel hydraulic diameter (mm)	1.4
Folie thickness (mm)	0.08	Wave height (mm)	1.6
Wavelength (mm)	4.2	Heat transfer area to volume ration (m^2/m^3)	1410
Density (kg/m^3)	2700	Specific heat (kJ/kg K)	0.92

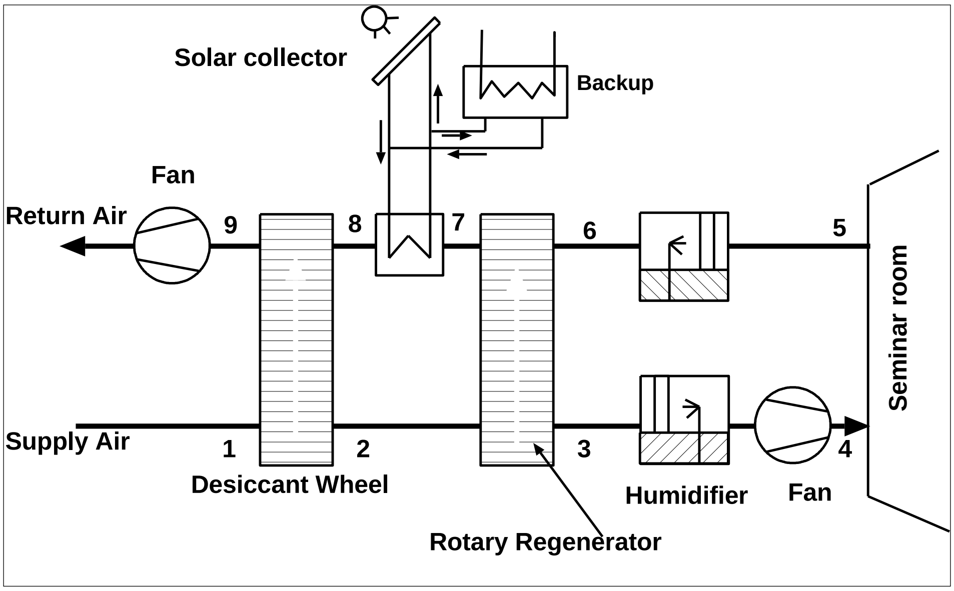

Fig. 1. Solar desiccant air conditioning system

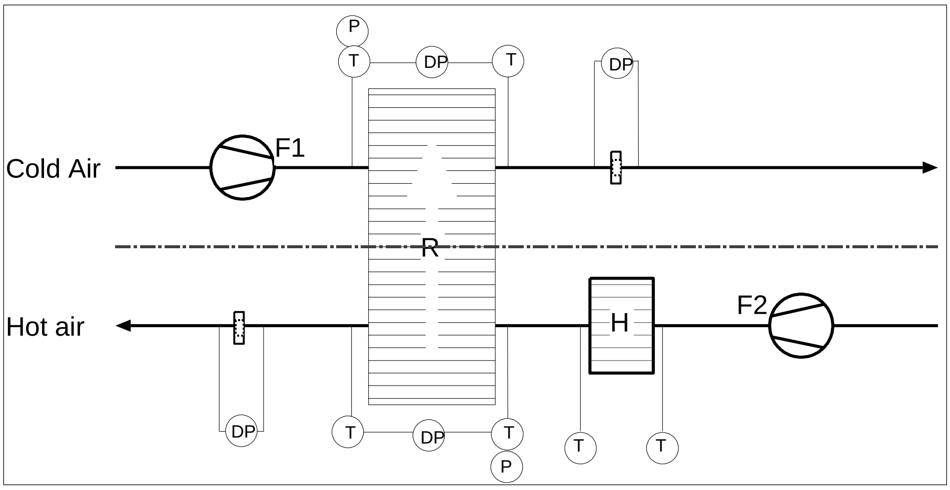

Fig. 2. Schematic of experimental setup

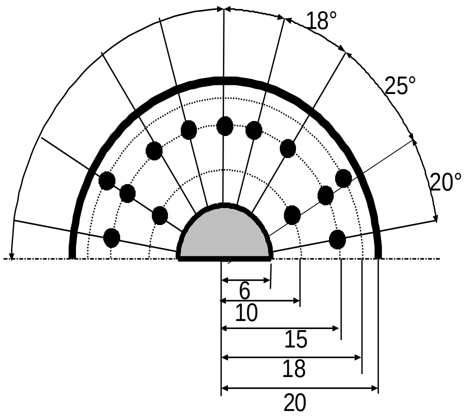

Fig. 3. Temperature measurement grid

Illustration example 1: The K-Type thermocouple is calibrated against a standard platinum resistance thermometer (SPRT) of type PT25. SPRT is calibrated by the 'Physikalisch Technischen Bundesanstalt (PTB)' to an uncertainty of 2.0-4.0 mK in the temperature range of 0-120 ℃. For the purpose of calibration the thermal source of dry block calibrator is used. For all the thermocouples employed in this work, a level of uncertainty in the range of 0.10 - 0.25 K is realized. This range is equivalent to 0.5-1.0% for the temperature range of 20 – 70 ℃; where 20 ℃ is the average room temperature during the measurement and 70 ℃ is the maximum regeneration temperature used in this work.

For a parameter that is not directly measured such as airflow rate, uncertainty is calculated from law of propagation of error. For example, if the parameter y is a function of variables x's as:

$$y = y(x_1, x_2, x_3, \ldots.) \tag{3}$$

the uncertainty $U(y)$ is

$$U(y) = \left\{\sum \left[c(x)U(x)\right]^2\right\}^{0.5} \tag{4}$$

where $U(x)$ and $c(x)$ are the uncertainty and the sensitivity coefficient of the variable x. The sensitivity coefficient is defined as:

$$c(x) = \frac{\partial y}{\partial x} \tag{5}$$

Illustration example 2: To measure the volumetric airflow, orifice flow equation is used

$$Q = C_d A_n Y \sqrt{\frac{2\Delta P}{\rho\left(1-\beta^4\right)}} \tag{6}$$

where Q is volumetric airflow, C_d is the orifice discharge coefficient, A is the area at the orifice exist, ΔP is measured pressure drop across the orifice, Y is the expansion factor, ρ is air density at the upstream of the orifice, $\beta = d/D$ is the contraction ratio and d and D are the orifice and duct diameter respectively. With only ΔP and ρ as variables, the uncertainty in the airflow rate in according with Eq. 4 is

$$\frac{U(Q)}{Q} = \sqrt{\left[\frac{U(\Delta P)}{2\Delta P}\right]^2 + \left[\frac{U(\rho)}{2\rho}\right]^2} \tag{7}$$

Table 2 shows the uncertainty level of the measurement devices.

Table 2. Measurement range and uncertainty

Temperature (°C)	20-70	0.1-0.25 K
Differential pressure (mbar)	0-19.99	0.1-0.5 %
Wheel speed (rpm)	0-24	0.1-0.2 %
Mass flow rate (kg/h	100-500	1-2 %

3. LEAKAGE MODEL

There are two types of leakages: carryover and pressure leakages. Carryover leakage is defined as the transport of the trapped gas in the voids of the matrix to the other fluid side just after the switching from hot stream to cold stream, and vice versa, due to matrix rotation. Although it cannot be eliminated, in general it is negligibly small [10], [11]. Pressure leakage is due to the pressure difference between the cold and hot streams in the regenerator. There are two types of pressure leakage radial and peripheral as shown in Fig. 4. The peripheral seal is made tight to eliminate the leakage between the housing and the wheel, hence preventing the flow bypass from the regenerator inlet to the regenerator outlet on each gas side (in the axial direction). In this work, besides the carryover leakages, the axial bypass (peripheral) leakage is considered small in comparison with the radial leakage.

Fig. 5 shows the physical model for radial pressure leakage. The points 1', 2', 3' and 4' represent the apparent airflow temperatures and rates. The variables at these points are assigned the subscript "a". The points 1, 2, 3 and 4 represent the real airflow temperature and rates.

with reference to Fig. 5 the following assumptions are considered:

1. The carryover and peripheral pressure leakages are neglected [10].
2. The pressure at the cold side is higher than that at the hot side; hence the leakage is from the cold to the hot side. This is due to the presence of electric heater which increases the pressure drop at the hot side at the same power of air ventilators.
3. The splitting at points 1 and 2 and the mixing at points 3 and 4 are adiabatic.
4. The splits at the cold section (points 1 and 2) are isothermal i.e. the apparent and real temperatures are equal [$T_{cia} = T_{ci}$ and $T_{coa} = T_{co}$].
5. Since the front and rear seals and clearances are identical and the pressure drop across the regenerator is small it may be assumed that the leakage airflow rates are equal $M_{ciL} = M_{coL}$.
6. Since the carryover leakage is neglected the real airflow rate at the inlet and outlet are equal.
7. The airflow specific heat is constant.

with these assumptions in mind the material and energy balance at the splitting and mixing points yield

$$\dot{M}_{coa} = \dot{M}_c - \dot{M}_{coL} \tag{8}$$

$$\dot{M}_{cia} = \dot{M}_c + \dot{M}_{ciL} \tag{9}$$

$$\dot{M}_{hia} = \dot{M}_h - \dot{M}_{coL} \tag{10}$$

$$\dot{M}_{hoa} = \dot{M}_h + \dot{M}_{ciL} \tag{11}$$

$$\dot{M}_{hia} T_{hia} = \dot{M}_h T_{hi} - \dot{M}_{coL} T_{co} \tag{12}$$

$$\dot{M}_{hoa} T_{hoa} = \dot{M}_h T_{ho} + \dot{M}_{ciL} T_{ci} \tag{13}$$

The energy balance for the matrix, assuming the regenerator is adiabatic and exchange no work with the surrounding, is

$$\dot{Q}_c = \dot{Q}_h = \dot{M}_c c_p (T_{co} - T_{ci}) = \dot{M}_h c_p (T_{hi} - T_{ho}) \tag{14}$$

There are in total seven equations [Eqs. (8)- (14)], six known (measured) quantities (T_{hia}, T_{hoa}, T_{ci}, T_{co}, M_{coa}, M_{hoa}) and seven unknown quantities (T_{hi}, T_{ho}, M_h, M_c, $M_{ciL} = M_{coL}$, M_{cia}, M_{hia}). Hence the system of equations is closed. Algebraic solution of Eqs. [(8)- (14)] yields the real airflow rate and temperature in term of apparent airflow properties.

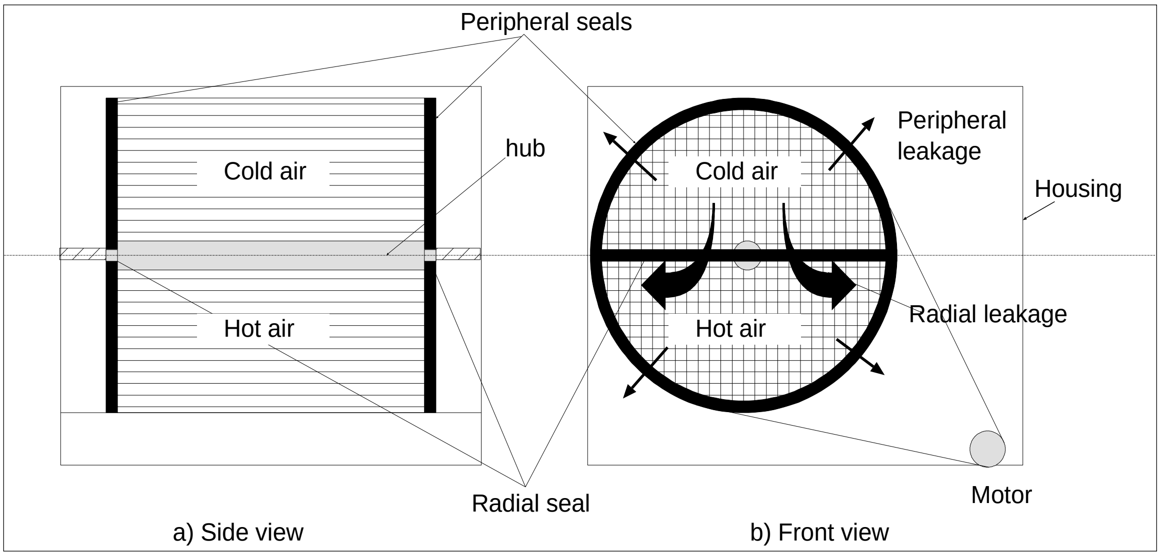

Fig. 4. Seals locations and leakages directions

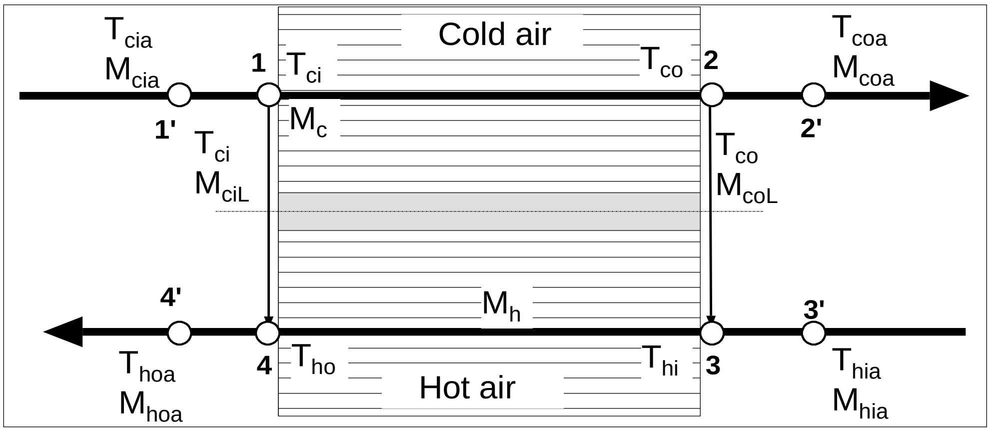

Fig. 5. Physical model of pressure leakage

$$\dot{M}_h = \frac{1}{2}\frac{\dot{M}_{hoa}(T_{hia}+T_{hoa}-2T_{ci})+\dot{M}_{coa}(T_{co}-T_{ci})}{T_{hia}-T_{ci}} \quad (15)$$

$$\dot{M}_h = \frac{1}{2}\frac{\dot{M}_{hoa}(T_{hia}-T_{hoa})+\dot{M}_{coa}(T_{co}+T_{ci}-2T_{hia})}{T_{hia}-T_{ci}} \quad (16)$$

$$T_{hi} = \frac{\dot{M}_{coa}(2T_{hia}-T_{co})(T_{co}-T_{ci})+\dot{M}_{hoa}[T_{hia}(T_{hoa}-2T_{ci})+T_{co}(T_{hia}-T_{hoa})]}{T_{hia}-T_{ci}} \quad (17)$$

$$T_{ho} = \frac{\dot{M}_{coa}(T_{ci})(T_{co}-T_{ci})+\dot{M}_{hoa}[T_{hoa}(T_{hia}-T_{ci})+T_{hia}(T_{hoa}-T_{ci})]}{T_{hia}-T_{ci}} \quad (18)$$

where T is in °C and M is in kg/s.

By exchanging the airflow temperature and rate of the hot side by those of cold side Eqs. [(16)-(18)] can be used for the case where the leakage moves from the hot side to cold side.

4. DATA REDUCTION

5.1 Effectiveness

In accordance with testing standards such as ANSI/ASHRAE Standard 84- 2008 [2] the regenerator effectiveness is

$$\varepsilon = \frac{C_c(T_{co}-T_{ci})+C_h(T_{hi}-T_{ho})}{2C_{\min}(T_{hi}-T_{ci})} \quad (19)$$

where the temperature is the bulk temperature. For constant airflow properties and a uniform velocity, the bulk temperatures is estimated as:

$$T = \frac{\int_0^{\pi}\int_0^{R} T(r,\theta) r dr d\theta}{\int_0^{\pi}\int_0^{R} r dr d\theta} \quad (20)$$

The integration given by equation 20 is evaluated numerically as:

$$T_{co} = \frac{1}{N}\sum\sum T_{co,ij} \qquad T_{ho} = \frac{1}{N}\sum\sum T_{ho,ij} \quad (21)$$

where i stands for r, j for μ, $T_{ij} = T(r;\ \theta)$ and N =13 is the number of measurement nodes.

5.2 Dimensionless Groups

The main parameters that influence the effectiveness of the regenerator are C^*, Cr^* and NTU. These dimensionless groups are defined as:

- The airflow heat capacities ratio $C^* = \frac{C_{\min}}{C_{\max}}$

where

$$C_{\min} = \begin{cases} C_c & C_c < C_h \\ C_h & C_h < C_c \end{cases}$$

$$C_{\max} = \begin{cases} C_c & C_c > C_h \\ C_h & C_h < C_c \end{cases}$$

$$C_i = \dot{M}_i cp$$

For balanced flow C^* = 1.

Matrix capacity $Cr^* = \dfrac{C_m}{C_{\min}}$

where $C_m = M_m cp_m \omega$, c_{pm} is the specific heat capacity of the matrix, M_m is the mass of the matrix in kg and ω is the rotational speed in revolution per second. The mass of the matrix is $M_m = \rho_m V$, where ρ is matrix density and V is the solid volume of the matrix.

- The number of heat transfer unit is

$$NTU = \frac{UA}{C_{\min}} \tag{22}$$

where

$$\frac{1}{UA} = \frac{1}{h_c A_c} + \frac{1}{h_h A_h} \tag{23}$$

The local heat transfer coefficient is calculated using Shah [10] correlation for narrow channels. It should be remembered that the conduction resistance is neglected in Eq. (23) because of the thin wall and high thermal conductivity of the matrix.

5. RESULTS AND DISCUSSION

To study the influence of leakage on the regenerator parameters, numerous quality measurements based on statistical error analysis are computed. These include percent deviation (*PD*), average absolute percent deviation (AAPD), minimum absolute percent deviation (APD_{min}) and maximum absolute percent deviation (APD_{max}). The percent deviation is defined as:

$$PDi = \frac{X_{a,i} - X_i}{X_{a,i}} \tag{24}$$

where $X_{a,i}$ and X_i are the apparent and real regenerator performance factor, respectively.

The real and apparent regenerator performance factors for all measurement are evaluated using the model shown in section 5. The experimental results include over 300 data points covering a wide range of operation conditions. Due to space limitation not all data are given. About 120 data points are measure. Table 3 gives the range of performance factors and the statistical parameters for all data points.

6.1 Influence of Leakage on the Regenerator Parameters

Fig. 6 shows the airflow rate that escaped from the cold side to the hot one. The amount of leakage represents about 2 to 7 % of the apparent airflow rate and it increases with airflow rate. Table 4 shows, as an example, the influence of leakage on C^*, Cr^* and *NTU*. The values of the real parameters, together with statistical parameters, are given in Table 4. It can be seen that the apparent C^*~ 1, while the real $C^* < 1$. This means that what is considered as balanced is in fact unbalanced. The APD_{min}, APD_{max} and *AAPD* in C^* are 7.22, 13.13 and 10.1, respectively. For this particular sample the *AAPD* for the regenerator capacity and the number of transfer unit are 5.61 and 5.64, respectively.

Table 2. Range of Deviation and Performance Factors

Performance factor	Real		Apparent		Deviation		
	Min	Max	Min	Max	APD_{min}	APD_{max}	AAPD
C^*	0.78	0.95	0.95	1.00	1.04	22.00	12.81
Cr^*	0.21	67.41	0.21	62.53	0.00	13.85	7.49
NTU	3.66	14.10	3.50	12.76	0.50	13.95	7.46
ε	0.17	0.83	0.17	0.78	0.00	15.00	6.62

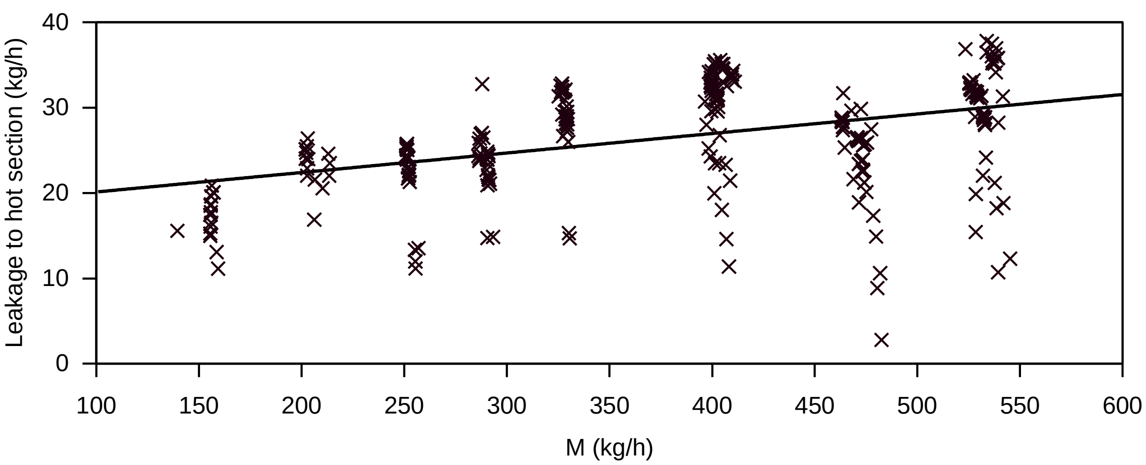

Fig. 6. Airflow leakage rate (the solid line represents the mean error)

Table 4. Influence of leakage on regenerator parameters

	C^*			Cr^*			NTU		
rpm	Real	Apparent	PD	Real	Apparent	PD	Real	Apparent	PD
0.72	0.9	0.97	7.22	0.58	0.56	-3.57	3.7	3.55	-4.23
1.21	0.86	0.99	13.13	0.99	0.92	-7.61	3.79	3.52	-7.67
1.21	0.89	0.99	10.1	0.96	0.91	-5.49	3.69	3.5	-5.43
2.41	0.90	0.99	9.09	1.93	1.84	-4.89	3.69	3.52	-4.83
4.82	0.89	0.99	10.1	3.87	3.67	-5.45	3.7	3.51	-5.41
7.23	0.89	0.99	10.1	5.82	5.51	-5.63	3.71	3.51	-5.70
9.64	0.89	0.99	10.1	7.78	7.35	-5.85	3.72	3.51	-5.98
12.05	0.89	0.99	10.1	9.72	9.20	-5.65	3.72	3.52	-5.68
14.46	0.89	1.00	11.0	11.65	11.01	-5.81	3.71	3.51	-5.7
16.87	0.89	0.99	10.1	13.61	12.88	-5.67	3.72	3.52	-5.68
19.28	0.89	0.99	10.1	15.58	14.73	-5.77	3.72	3.52	-5.68
21.69	0.89	0.99	10.1	17.48	16.53	-5.75	3.71	3.51	-5.70
24.10	0.89	0.99	10.1	19.47	18.41	-5.76	3.72	3.52	-5.68
AAPD			10.1			5.61			5.64
APD_{min}			7.22			3.57			4.23
APD_{max}			13.13			7.61			7.67

6.2 Influence of Leakage on Effectiveness

Fig. 7 shows the influence of airflow leakage on real temperatures at the hot side. The leakage error (the different between the real and the apparent temperatures) varies between 0.25 to 2.5 K. It should be remembered that the cold side airflow temperatures are not affected by the leakage (cf. section 4). Table 5 shows a calculation sample of effectiveness and the other regenerator performance factors. The rest of the result about 300 data points covering wide range of $0{:}8 \le C^* \le 0.95$, $0.8 \le Cr^* \le 35$ and $3 \le NTU \le 14$. As can be seen in the given sample (Table 5) AAPD is about 5. For wide range of measurement AAPD is in the range of 5 to 11.

6. CONCLUSIONS

The influence of leakage on regenerator performance factors is experimentally investigated. Significant difference between the real and apparent performance factors is found even under small leakage percent. Simple equations that facilitate the calculation of the real airflow temperature using the measured (apparent) airflow rate and temperature are developed. The equations are useful particularly in testing and rating regenerator.

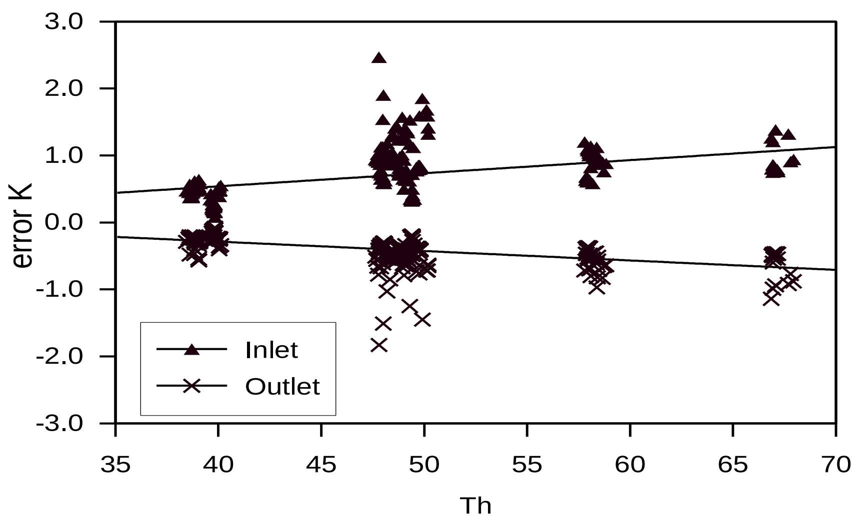

Fig. 7. Influence of leakage on hot side temperatures

Table 5. Influence of air flow leakage on regenerator effectiveness

	C*		Cr*		NTU		ε	
rpm	Real	PD	Real	PD	Real	PD	Real	PD
0.72	0.9	8.16	0.59	-5.36	3.74	-4.47	0.44	0
1.21	0.87	12.12	0.99	-6.45	3.79	-6.76	0.63	-5
1.21	0.86	13.13	0.99	-7.61	3.78	-7.39	0.64	-4.92
2.41	0.89	11.00	1.94	-6.01	3.70	-5.71	0.74	-5.71
4.82	0.89	11.00	3.88	-6.01	3.71	-6	0.78	-5.41
7.23	0.89	11.00	5.83	-6.00	3.71	-6	0.79	-5.33
9.64	0.90	9.09	7.76	-5.58	3.71	-5.4	0.80	-5.26
12.05	0.89	10.1	9.71	-5.66	3.71	-5.4	0.80	-5.26
14.46	0.89	10.1	11.65	-5.62	3.71	-5.4	0.80	-5.26
16.87	0.89	10.1	13.58	-5.6	3.71	-5.7	0.80	-5.26
19.28	0.89	10.1	15.53	-5.43	3.71	-5.4	0.80	-3.9
21.69	0.89	11	17.49	-5.74	3.72	-5.98	0.81	-6.58
24.10	0.89	10.1	19.43	-5.6	3.72	-5.68	0.81	-5.19
AAPD		10.24		5.85		5.63		4.85
APD_{mim}		6.32		5.26		7.39		0
APD_{max}		13.13		7.61		3.59		6.58

REFERENCES

[1] Rabah, A. A.; Fekete, A.; Kabelac, S., 2009, "”Experimental Investigation on a Regenerator Operating at Low Temperatures,” *ASME Journal of Thermal Sciences and Engineering Applications*, 1 (4),pp: 041004-13

[2] ASHRAE. 2008. "ASHRAE Standard 84-2008, Method of Testing Air-To-Air Heat Exchangers”. *Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers Inc*.

[3] Lambertson, T. J., 1958, "Performance of a periodic flow heat exchanger,” Trans. ASME, 80, pp. 586-592.

[4] Kays, W. M., London, A. L., 1984,”*Compact heat exchanger,*” McGraw-Hill, New York.

[5] Ba´clic, B.S., 1985, "The application of the Galerkin method to the solution of symetricic and balanced counter flow regenerator problem”, *ASME J Heat Transfer*, 107, pp. 214-221.

[6] Rabah, A. A.; Kabelac, S., 2010, "A simplified solution of the regenerator periodic problem: the case for air conditioning” (Forschung im Ingenieurwesen) *Journal of Engineering Research*. DOI 10.1007/s10010-010-0126-z.

[7] Shang, W., Besant, R. W., 2004, "Measurement of the pore size variation and its effect in energy wheel performance,” *ASHRAE Transaction*, 110 (1), pp. 410-421.

[8] Skiepko, T., 1989,"Effect of reduction in seal clearances on leakages in a rotary heat exchanger,” Heat Recovery Systems and CHP, 9(6) , pp. 553-559.

[9] Shah, R. K. and Skiepko, T., 1999,"Influence of leakage distribution on the thermal performance of a rotary regenerator,” *Appl. Thermal Engng*, 19, 685-705.

[10] Shah, R. K., 1981, "Thermal design theory of regenerators,” In: Kakac, S., Bergles, A. E., Mayinger, F., 1981, "*Heat exchangers: Thermal-hydraulic fundamentals and design*" pp. 721-763, Hemisphere, New York.

[11] Han, Hwa-Taik, Kim, Min-Kyu, 2005, "An Experimental Study on Air Leakage and Heat Transfer Characteristics of a Rotary-type Heat Recovery Ventilator,” *International Journal of Air-Conditioning and Refrigeration*, 13(2), pp.83-88

Nomenclature

A	Heat transfer area, Constant
cp	Specific heat capacity
C	Heat capacity
h	Heat transfer coefficient
L	Length
M	Mass flow rate
r	Radius
ρ	Density
ε	Effectiveness
ω	Rotational speed
C^*	Heat capacity ratio
Cr^*	Matrix to fluid heat capacity ratio
NTU	Number of transfer units