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Effect of Variation of Temperature and Speed on the Performance of Fixed Matrix Periodic Flow Regenerator

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***Abstract:*--The regenerator is a storage-type heat exchanger. The heat transfer surface or elements are usually referred to as a matrix in the regenerator. Linear regression method is applied for finding out a relation between different variables by taking up the values from either graphs or tables. Objective of the work is to find out the relation between the effectiveness of a heat exchanger to the various operating parameters like temperature, speed of rotation of wheel, mass flow rate, etc. Then, all the methodology is converted into a C++ program which takes up a set of initial values and displays the corresponding values of effectiveness and pressure drop. The effect of changing of different parameters on effectiveness, pressure drop and heat duty is observed next and corresponding effect is plotted on the graph. The trend obtained is then analyzed and possible explanations are discussed for the results.**

***Index Terms*—matrix, effectiveness, thermo hydraulic**

I. INTRODUCTION

The regenerator is a storage-type heat exchanger. The heat transfer surface or elements are usually referred to as a matrix in the regenerator. To have continuous operation, either the matrix must be moved periodically into and out of the fixed streams of gases, as in a rotary regenerator, or the gas flows must be diverted through valves to and from the fixed matrices as in a fixed matrix regenerator. The latter is also sometimes referred to as a periodic-flow regenerator, a swing regenerator, or a reversible heat accumulator.

The desired material properties for the regenerator are high volumetric heat capacity (high cp) and low effective thermal conductivity in the longitudinal (gas flow) direction. It should be noted that at very low temperatures, 20 K (368R) and below, the specific heat of most metals decreases substantially, thus affecting the regenerator performance significantly.

Linear regression method is applied for finding out a relation between different variables by taking up the values from either graphs or tables. e.g. , the value of effectiveness for the fixed matrix regenerative heat exchanger is calculated by the fig 2.28 to 2.35 in [1]. However, applying above technique on these graphs to get a relation yields the relation in the form given as,

$$\varepsilon = 56.48186459(N_{tu})^{0.18759734511}(C_c/C_h)^{0.0982195126554}(C_r/C_c)^{.0617219686585}$$

Where ε is effectiveness in percentage

For checking out the accuracy, the values are calculated by the above relation and then compared by the graphical value and corresponding value of error is noted down in table 1,

$N_{tu,o}$	C_c/C_h	C_r/C_c	ε_{graph}	$\varepsilon_{formula}$	Error (%)
4.38	0.95	3.49	82	80.084	2.34
9	0.95	1.5	87.5	87.0168	0.55
7	0.90	5	90	88.9395	1.178
2	1	2	65	67.137	3.288
5	0.7	1.5	79	75.629	4.627

Table 1: comparison of graphical and numerical values of ε

As observed from the above table the values calculated by the above relation lies in the acceptable range and hence the technique can be applied for finding out direct relations for different sets of graphs and tables.
Therefore, in this work this technique is adopted for finding out relations ranging from eq (2)-(6) and eq(12) . the values for relations are either taken from graphs present in [1] or from the table of properties of dry air taken from [9].

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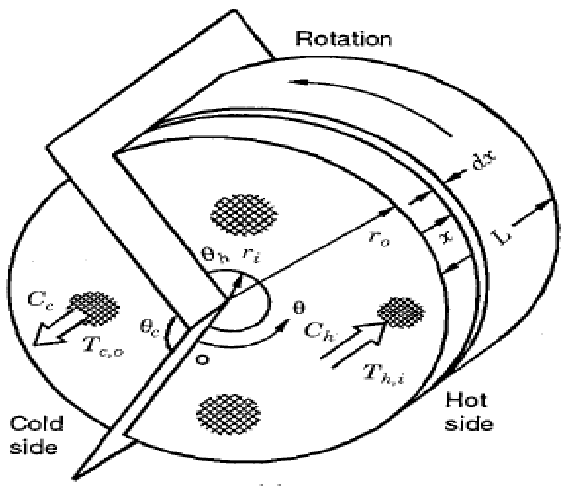
The relations are generally in the form of exponents given by the general form

$F = A_0 (B)^m (C)^n$

Where,
F= value of property
A_0=constant term
B,C =dependent variables
m, n= powers of exponent

II. GEOMETRICAL and THERMOHYDRAULIC PARAMETERS OF REGENRATIVE HEAT EXCHANGERS



(a)

Fig II.2 Rotary regenerator showing sections x and dx (From Shah, 1981)

Figure II.2 depicts a schematic view of a rotary generative heat exchanger with periodic flow. These details are estimated based on the following assumptions.

1. The regenerator operates under quasi-steady-state or regular periodic-flow conditions (i.e., having constant mass flow rates and inlet temperatures of both fluids during respective flow periods).

2. Heat loss to or heat gain from the surroundings is negligible (i.e., the regenerator outside walls are adiabatic).

3. There are no thermal energy sources or sinks within the regenerator walls or fluids.

4. No phase change occurs in the regenerator.

5. The velocity and temperature of each fluid at the inlet are uniform over the flow cross section and constant with time.

6. The analysis is based on average and thus constant fluid velocities and the thermophysical properties of both fluids and matrix wall material throughout the regenerator (i.e., independent on time and position).

7. The heat transfer coefficients (h_h and h_c) between the fluids and the matrix wall are constant (with position, temperature, and time) throughout the exchanger.

8. Longitudinal heat conduction in the wall and the fluids is negligible.

9. The temperature across the wall thickness is uniform at a cross section and the wall thermal resistance is treated as zero for transverse conduction in the matrix wall (in the wall thickness direction).

10. No flow leakage and flow bypassing of either of the two fluid streams occurs in the regenerator due to their pressure differences. No fluid carryover leakage (of one fluid stream to the other fluid stream) occurs of the fluids trapped in flow passages during the switch from hot to cold fluid period, and vice versa, during matrix rotation or valve switching.

11. The surface area of the matrix as well as the rotor mass is uniformly distributed.

12. The time required to switch the regenerator from the hot to cold gas flow is negligibly small.

13. Heat transfer caused by radiation within the porous matrix is negligible compared with the convective heat transfer.

14. Gas residence (dwell) time in the matrix is negligible relative to the flow period.

3.2.1 Geometrical Properties

For the geometrical shown in fig II.2 one may get total heat transfer area as

$$A_h + A_c = \alpha * vol = \square * \pi * d * L * t \quad (1)$$

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$$A_h/A_c=A_{fr,h}/A_{fr,c} \quad (2)$$

The effective free flow areas are,

$$A_c=p*A_{fr} \quad (3)$$

The hydraulic radius is given as,

$$r_h=p/\alpha \quad (4)$$

Thermohydraulic parameters

The rate of heat transfer may be calculated as follows

$$Q = m_c.Cpc.(T_{c,out} – T_{c,in}) = m_h.Cph.(T_{h,in} – T_{h,out}) \quad (5)$$

The overall number of transfer units,

$$N_{tu,o}=1/C_C[1/((1+hA)_c+(1/hA)_h)] \quad (6)$$

The friction factor f can be calculated by linear regression method by using values taken from graph (kays and London,1980).

$$f= 7.4466344\ (p)^{.58707552572}(Nr)^{-.2609736085} \quad (7)$$

$$N_r=4r_hG/\mu \quad (8)$$

The pressure drop on cold and hot side is given by equation,

$$\Delta P/P_1=G^2v_1/2gP_1[(1+p^2)(v_2/v_1-1)+f(\ A/A_c\ )(v_m/v_1]$$

The values are transformed in terms of independent variables and are represented as,

$$\Delta Pc = 3.55*10^3\frac{(1+Afr,h/Afr,c)}{d^2t^2\alpha^2rh^3}[(1+rh^2)+45.096(\frac{dt\alpha}{(1+Afr,h/Afr,c)})^{[illegible]}L^{[illegible]} \quad 9)$$

And

$$\Delta Ph = 2.52*10^{2}\frac{(1+Afr,c/Afr,h)}{d^2t^2\alpha^2rh^2}[0.61(1+rh^2)+25.9(\frac{dt\alpha}{(1+Afr,c/Afr,h)})^{[illegible]}L^{[illegible]} \quad 10)$$

III. OBJECTIVE/ DESCRIPTION OF THE PROBLEM

Objective of the work is to find out the relation between the effectiveness of a heat exchanger to the various operating parameters like temperature, speed of rotation of wheel, mass flow rate, etc.

Geometry of the heat exchanger and the material properties of wheel are kept fixed and the values are taken from a problem in Compact heat exchanger, Kays And London,1980 [1] given as- $A_{fr,h}$ =2.4712 m^2, $A_{fr,c}$ =0.9894 m^2, p=.725, α= 3215.2231 m^2/m^3
m=159.2136 kg, C_s=.50252 KJ/Kg K

For getting the variation in the value of effectiveness heat duty and pressure drop with respect to a single variable, all the other variables need to be kept fixed. Therefore, these constant values are also taken by [1] given as-

w_c=9.5 Kg/sec, w_h=9.6425 Kg/sec, $T_{c,in}$=440 K, $T_{h,in}$=947 K, N=26.5 RPM

The corresponding outlet temperatures can be obtained by the following relation assuming effectiveness as 85% initially,

$$\varepsilon= (t_{c,in}-t_{c,out}) / (t_{h,in}-t_{c,in}) = (t_{h,in}-t_{h,out})/ (t_{h,in}-t_{c,in}) \quad (11)$$

Now, after getting these values, the corresponding values of specific volume ,C_p, Prandtl number and surface viscosity is calculated by the relations obtained by the linear regression method discussed in the earlier section. All these values are calculated at mean temperatures of both the cold and hot side.

$T_{c,mean}=T_{c,in}+T_{c,out}/2$ and $T_{h,mean}=T_{h,in}+T_{h,out}/2$

The corresponding values of properties are calculated by the method described in section 1.1 and the values for calculation are taken from the table of properties of air from [9].

$$v=2.058635235\ (T)^{-0.0845664147775} \quad (12)$$

$$\mu=.005434T+.24\ (10^{-5})\ kg/m\ s \quad (13)$$

$$Cp=0.639473708(T)^{0.0843642463687}\ KJ/kg\ K \quad (14)$$

$$Pr=.807261563(T)^{-.0214123812024} \quad (15)$$

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The values of properties are calculated from the above relations for cold side mean temperature as well as hot side mean temperature.

Reynolds number is then calculated by the relation

$G=w/A_c$ (16)

And,

$N_r=4r_hG/\mu$ (17)

From the above values the corresponding values of f, N_{st} and h can be calculated by the fig 7-8 and fig 7-9 in [1]

Number of transfer units, N_{tu} , is calculated by the equation

$N_{tu}=N_{st}\,A/A_c$ (18)

The effectiveness of the heat exchanger can be considered as a function of N_{tu}, C_c/C_h and C_r/C_c.

The required values are found out by the definition of the terms and are given as

$C_c=w_c\,.C_{p,c}$ (19)

$C_h=w_h\,.C_{p,h}$ (20)

$C_r=N_s.m.c_s/60$ (21)

The values of effectiveness is calculated from the fig 2-28 to 2-33 in [1]

Alternatively, Apply linear regression on these graphs to get the relation amongst the quantities.

$\square= 56.48186459(N_{tu})^{0.18759734511}(C_c/C_h)^{00.0982195126554}(C_r/C_c)^{.0617219686585}$ (22)

The corresponding pressure drops and heat duty can be calculated by the relation

$\Delta P/P_1=G^2v_1/2gP_1[(1+p^2)(v_2/v_1-1)+f\,A/A_c\,v_m/v_1]$ (23)

$Q=w_hCp_h(t_{in}-t_{out}) = w_cCp_c(t_{out}-t_{in})$ (24)

Graphs showing variation of effectiveness and pressure drop with respect to

1. Variation in inlet temperature of cold side

Firstly, the inlet temperature of cold side is varied while keeping all other values constant and values as given in section 2.2

The range of values of temperature is from 318K to 698 K and the results are obtained as

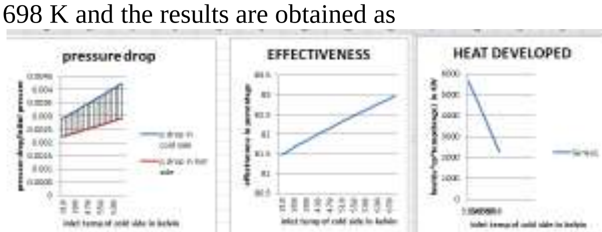

Fig III.1- variation w.r.t cold side temp

2. Variation in inlet temperature of hot side

Firstly, the inlet temperature of hot side is varied whsssile keeping all other values constant and values as given in section 2.2

The range of values of temperature is from 7000K to 1460 K and the results are obtained as

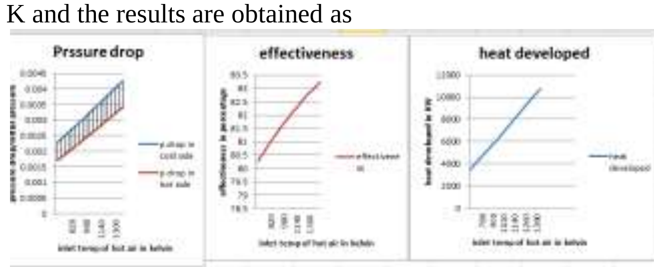

Fig III.2- variation w.r.t hot side temp

3. Variation in mass flow rate of cold side

Now, the mass flow rate of cold side is varied while keeping all other values constant and values as given in section 2.2

The range of values of flow rate is from 5 kg/sec to 15 Kg/sec and the results are obtained as

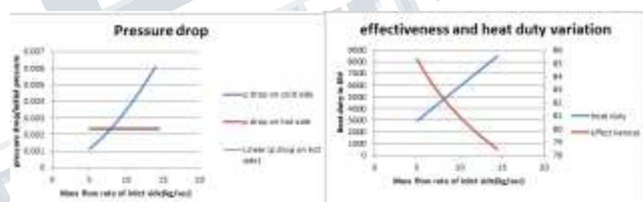

Fig III.3-variation w.r.t w_c

4. Variation in mass flow rate of hot side

Now, the mass flow rate of hot side is varied while keeping all other values constant and values as given in section 2.2

The range of values of flow rate is from 5 kg/sec to 15 Kg/sec and the results are obtained as

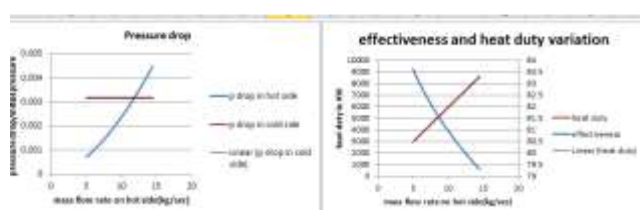

Fig III.4- variation .w.r.t w_h

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5. Variation in speed of rotor

Now, the speed of rotor is varied while keeping all other values constant and values as given in section 2.2
The range of values of speed is from 5 RPM to 100 RPM and the results are obtained as

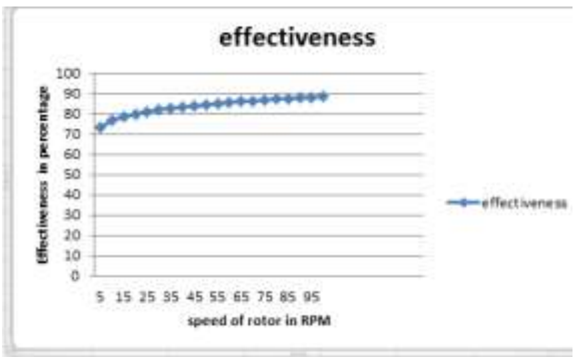

Fig III.5-variation w.r.t N

During the variation in speed of rotor, only effectiveness changes and no changes are observed in pressure drop and heat duty.

IV. ANALYSIS/DISCUSSION

The figures III.1-III.5 depict the effect of changing of a single type of parameters on the effectiveness, heat duty and the pressure drops in the cold side and hot side respectively. The trend observed generally is either increasing or decreasing depending upon the parameter which is being varied.

III.1 shows the variation w.r.t temperature $T_{c,in}$ in which the effectiveness tends to increase with increase in the temperature. This may be attributed to the higher mean temperature of the cold side and corresponding higher values of the properties of air at this temperature. This may lead to the conclusion that inlet temperature of cold side should be kept to the maximum value possible. However, this may not be feasible as because the pressure drops at both sides also tend to increase with the temperature. Higher pressure drop causes increase in power consumption and hence the operating cost. Therefore, a compromise has to be done with the value of temperature in order to keep pressure drop within the design limit.

Similar would be the case III.2 in which the inlet temperature of hot side is varied. The values of both the effectiveness and pressure drop tend to increase with the temperature and hence a compromising value is adopted for the designing.

In fig III.3, the mass flow rate of cold air is varied. There is an assumption involved with the process that the areas and speed of wheel is constant. In this case, the effectiveness tend to decrease with the increase in w_c which may be attributed to higher value of C_c. however, the heat duty increases with w in this case and therefore the intersection point of both the values is taken as the design point for this situation. Pressure drop only varies on the cold side which has to be kept under consideration during selection of parameters. Similarly, the process is taken for fig III.4.

In fig III.5 the speed of the wheel (in RPM) is varied. In this case, no change is observed in the heat duty and pressure drops considering the assumption that all other parameters are constant. Only the effectiveness of the heat exchanger varies due to change in C_r/C_c ratio. The trend observed in the graph shows that the effectiveness tends to increase with the speed upto a certain level, and the value starts to become constant for the higher values of speed. This may be attributed to the effect of speed of wheel which only causes the change in the heat capacity of the matrix. Considering all other parameters like mass, density and area ratio as constant, it may be possible that speed has very small impact on the effectiveness. Similar results were obtained by O. Buyukalaca and T. Yilmaz (2002)[9].

With the help of above work, therefore, one is able to estimate the values of different thermohydraulic parameters which would give the best possible effectiveness considering the proper feasible values of heat duty and pressure drops as well.

V. CONCLUSION AND FUTURE SCOPE

Initially the design process of the rotary regenerative heat exchanger is explained and the method is described. Least square method is adopted for formulating all the graphical and experimental data and getting a direct relation. After this, all the methodology is converted into a C++ program which takes up a set of initial values and displays the corresponding values of effectiveness and pressure drop.

The effect of changing of different parameters on effectiveness, pressure drop and heat duty is observed next and corresponding effect is plotted on the graph. The trend obtained is then analyzed and possible explanations are discussed for the results.

Further research can be done on the second law based optimization of the same type of heat exchanger which would also include entropy considerations. Furthermore, the cases of different types of matrix can also be considered for the optimization in order to get the best suitable combinations of parameters for the design of the heat exchanger.

VI. NOMENCLATURE

A, A_{HT} heat transfer area, m^2
A_{fr} frontal flow area, m^2
ε Effectiveness
A_c Effective Flow area, m^2
α heat transfer area per unit volume, m^2/m^3
C heat capacity rate (mCp), WK^{-1}
Cp specific heat of fluid, Wkg^{-1} K^{-1}
ρ density, kg m^{-3}
r_h hydraulic radius, mm
μ viscosity, N m^{-2} s^{-1}
f friction factor
Pr Prandtl number
ΔP pressure drop, N m^{-2}
Q rate of heat transfer, W
R specific gas constant, J kg_1 K_1
Re Reynolds number
N_{st} Stanton number [h/(GCp)]
p porosity
v Specific volume, m^3/kg
t wheel thickness, m
T temperature, K
U overall heat transfer coefficient, Wm^{-2} K^{-1}
G mass flux velocity (m/Aff), kg m^{-2} s^{-1}
h heat transfer coefficient, Wm^{-2} K^{-1}
L heat exchanger length, m
w mass flow rate of fluid, kg s^{-1}
m mass of matrix,kg
N speed of rotation of wheel, RPM
$N_{tu,o}$ overall number of transfer units, dimensionless
P pressure, N m^{-2}

Subscripts
c cold side air
h hot side air
in in
out outlet
fr frontal
r wheel

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