ROTARY REGENERATOR DESIGN THEORY AND OPTIMISATION

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Abstract

A review of the simplest and most frequently used design theory for counterflow rotary regenerators is given. A new correlation for the effectiveness of a balanced regenerator is proposed. A computer program was written for the thermal evaluation and design of counterflow rotary regenerators. The program is used to optimise some of the operating conditions and design parameters if the effectiveness is specified. It is shown by an example how the disk rotational speed, flow split ratio and matrix wall thickness can be optimised. Five heating surfaces that were experimentally tested for their heat transfer and pressure drop characteristics are compared at their respective optimum operating points.

Nomenclature

A	area	m^2
C	heat capacity rate	W/K
c	specific heat	J/kgK
D	diameter	m
f	fanning friction factor	
G	mass flux	kg/sm^2
h	convective heat transfer coefficient	W/m^2K
K	loss coefficient	
k	thermal conductivity	W/mK
L	disk height	m
m	mass	kg
N	rotational speed	rev/s
Ny	characteristic heat transfer parameter	m^{-1}
Δp	pressure differential	N/m^2
p	absolute pressure	Pa
q	heat transfer rate	W
R	thermal resistance	K/W
Ry	characteristic flow parameter	m^{-1}
s	surface spacing	m
T	temperature	K
W	mass flow rate	kg/s
x	leakage factor or fraction	

Greek symbols

α	area ratio	
β	area density	m^2/m^3
δ	plate thickness	m
ε	effectiveness	
ϕ	flow split ratio	
μ	dynamic viscosity	kg/sm
ρ	density	kg/m^3
σ	porosity	

Dimensionless Groups

A_k^*	matrix solid area ratio	$A_{k,min}/A_{k,max}$
C^*	heat capacity ratio	C_{min}/C_{max}
C_r^*	matrix heat capacity ratio	C_r/C_{min}
$(hA)^*$	convection conductance ratio	$(hA)_{min}/(hA)_{max}$
Ntu_o	modified number of heat transfer units	$\frac{1}{C_{min}}\left[\begin{array}{c}1/(hA)_{min}\\+1/(hA)_{max}\end{array}\right]$
Pr	Prandtl number	$\mu c_p/k$
λ	longitudinal conduction parameter	$\frac{k_w A_{k,t}}{LC_{min}}$

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Subscripts

c	cold gas side or contraction
e	equivalent or expansion
fr	frontal
h	hot gas side or hydraulic
i	inlet
k	refer to matrix solid area for longitudinal conduction
m	mean or average or matrix
max	maximum
min	minimum
o	outlet
p	pressure
r	rotor
s	seal
w	matrix wall
∞	recuperator or ideal rotary regenerator rotating at infinite speed

Introduction

Since its invention the development of the rotary regenerator has been continuous, covering the principles of thermodynamics, fluid mechanics and structural engineering. The idea of using a slowly rotating drum containing a tightly packed heating surface matrix to transfer heat between two gas streams is still the basis of present-day designs. Today the rotary regenerator is a carefully engineered product which is highly reliable. The diameter of the drum can range from 350 millimetres to more than 15 metres. This causes the heating matrix to weigh anything from a few kilograms to several hundreds of tons.[1]

The challenge in regenerator design today is the optimisation of the operating conditions and the development of improved heating surfaces. Several new heating surfaces were experimentally tested for their pressure drop and heat transfer characteristics. The characteristics of a heating surface have a profound influence on the size and optimum operating conditions of the regenerator. For purposes of illustration, five of the surfaces are compared in a specific application. A review of the usual design procedures is given. An integral part in the performance evaluation procedures of a regenerator is the calculation of the effectiveness of a balanced regenerator. A novel version of the correlation of Lambertson[2] to account for the effect of the matrix heat capacity ratio (C_r^*) is presented.

Design theory for counterflow regenerators

To date, no complete solution for the differential equations governing the heat transfer process in a regenerator exists. The differential equations are subject to the normal idealisations made in the design theory. The idealisations are listed by Shah[3] and Bahnke & Howard.[4] The differential equations can be made dimensionless to arrive at the dimensionless groups for the effectiveness – number of transfer units (ε-Ntu_o) or the reduced length – reduced period (A-Π) methods of solution. The ε-Ntu_o method is usually used for rotary regenerators and the A-Π method for fixed matrix regenerators. This paper will focus on the ε-Ntu_o method. It can be shown that the effectiveness of a counterflow regenerator is a function of six dimensionless variables.[3]

$$\varepsilon = f\left(Ntu_o, (hA)^*, C_r^*, C^*, \lambda, A_k^*\right) \qquad (1)$$

The most important assumption in the derivation of the above relation is that there is no pressure or carry over leakage. In a direct contact type recuperator the effectiveness is only a function of Ntu and C^*. The parameters $(hA)^*$ and C_r^* arise because of the heat storing nature of a regenerator. The longitudinal heat conduction effect is accounted for by λ and A_k^*. An advantage of this particular choice of dimensionless parameters is that the influence of $(hA)^*$ and A_k^* on the effectiveness are small and that the solution method parallels that of a recuperator for the case of infinite C_r^* (infinite rotational speed). A program was written to solve the effectiveness as a function of the six above-mentioned parameters. The numerical scheme of Bahnke & Howard[4] was used. This was done for convenient evaluation of the approximate solution methods presented in the next section.

Approximate solutions for an ideal regenerator

If the assumption is made that there is no pressure or carry over leakage, the regenerator is said to be 'ideal'.

Solutions for the case of zero longitudinal heat conduction

In the case of zero longitudinal heat conduction the functional relationship for the effectiveness (equation 1) becomes:

$$\varepsilon = f\left(Ntu_o, C_r^*, C^*, (hA)^*\right) \qquad (2)$$

For the special case of infinite rotational speed, i.e. $C_r^* = \infty$ an analytical solution is available. The effectiveness is given by the well-known relation,

$$\varepsilon_\infty = \frac{1 - e^{-Ntu_o(1-C^*)}}{1 - C^* e^{-Ntu_o(1-C^*)}} \qquad (3)$$

For $C^* = 1$ equation (3) reduces to: $\varepsilon = (Ntu_o)/(1 + Ntu_o)$. The above relation is identical to that of an indirect contact type recuperator. Lambertson[2] correlated the effect of C_r^*. Kays &

London[5] later modified the exponent of C_r^* from 1.87 to 1.93.

$$\varepsilon = \varepsilon_\infty \left(1 - \frac{1}{9C_r^{*1.93}}\right) \quad (4)$$

where ε_∞ is calculated from equation (3).

Razelos[6] proposed the following method to calculate the effectiveness of an unbalanced ($C^* < 1$) regenerator. He disregarded the influence of $(hA)^*$ in the range $0.25 < (hA)^* < 4$. The specified values of Ntu_o, C^* and C_r^* are used to calculate $Ntu_{o,e}$ and $C_{r,e}^*$ for an equivalent balanced regenerator ($C^* = 1$).

$$Ntu_{o,e} = \frac{2Ntu_oC^*}{1+C^*} \text{ and } C_{r,e}^* = \frac{2C_r^*C^*}{1+C^*} \quad (5)$$

Therefore the effectiveness of the balanced regenerator using $Ntu_{o,e}$ and $C_{r,e}^*$ with $C^* = 1$ in eq. (4) is:

$$\varepsilon_e = \frac{Ntu_{o,e}}{1+Ntu_{o,e}} \left(1 - \frac{1}{9C_{r,e}^{*1.93}}\right) \quad (6)$$

The method by Razelos[6] for calculating the effectiveness of an unbalanced regenerator requires the effectiveness of a balanced regenerator. Eq. (6) was found to be less accurate at high Ntu_o numbers. Numerical results for a balanced, symmetric regenerator ($C^* = (hA)^* = 1$) for the complete range of $C_r^* > 1$ and $0.5 < Ntu_o < 500$ were used to extend eq. (4) for $C^* = 1$ as follows:

$$\varepsilon_e = \frac{Ntu_{o,e}}{1+Ntu_{o,e}} \left(1 - \frac{1}{AC_{r,e}^{*\,n}}\right) \quad (7)$$

where $A = 15.78/Ntu_{o,e} + 3.2678 \ln(Ntu_{o,e}) + 0.0019373[\ln(Ntu_{o,e})]^5$

$n = 1.881 + 0.0331197[\ln(Ntu_{o,e})]^2$

Eq. (7) agrees with numerical results within 0.25 % for the range $C_{r,e}^* > 1.5$ and $0.5 < Ntu_{o,e} < 500$ and within 0.7 % for $C_{r,e}^* > 1$ and $0.5 < Ntu_{o,e} < 500$. The largest error is 0.69 % and occurs at $C_{r,e}^* = 1.5$ and $Ntu_{o,e} = 80$.

The effectiveness of the unbalanced regenerator is calculated from:

$$\varepsilon = \frac{1-e^n}{1-C^*e^n} \text{ where } n = \varepsilon_e \left(C^{*2}-1\right) / \left(2C^*\left(1-\varepsilon_e\right)\right) \quad (8)$$

Baclic[7] obtained an expression for the effectiveness of a balanced regenerator. He employed the Galerkin method to obtain a closed form power series solution. His results are valid for $C^* = (hA)^* = 1$. The Baclic equation[7] is in excellent agreement with numerical results. The largest errors occur at high Ntu_o numbers and $C^* = 1$. At $Ntu_o = 500$ the error is 0.3 %. The method can however be extended to produce answers to the desired accuracy. The method is not for hand calculation and the computing time needed is similar to that of finite difference methods. It is therefore not well suited if one is interested in optimisation. The solution is a long one and will not be presented here.

The method by Razelos[6] was evaluated over a wide range of parameters. It was used with numerical results for $C^* = 1$ to extend the results to $C^* < 1$ for $(hA)^* = 1$. This was done in order to establish how well the method of Razelos[6] accounts for the influence of C^*. Predictions agree with numerical results within 1 % in the following ranges:

1. $1 < Ntu_o < 500$, $C_r^* > 1$, $0.7 < C^* < 1$
2. $1 < Ntu_o < 500$, $C_r^* > 1.5$, $0.5 < C^* < 1$
3. $1 < Ntu_o < 500$, $C_r^* > 2.5$, $0.3 < C^* < 1$

The influence of $(hA)^*$ has not been taken into account by any of the approximate solutions thus far. Usually it is quoted that the influence of $(hA)^*$ is small in the range $0.25 < (hA)^* < 4$ and the results are then given for $(hA)^* = 1$. Numerical results were calculated by the author for $(hA)^* = 0.25$ and 4 and it was found that the deviation in the effectiveness $(\varepsilon_{(hA)^*=1} - \varepsilon_{(hA)^*=0.25;4})$ is less than 1 % in the following ranges:

1. $1.5 < Ntu_o < 500$, $C_r^* > 2.5$, $0.5 < C^* < 1$
2. $1.5 < Ntu_o < 500$, $C_r^* > 1.5$, $0.7 < C^* < 1$
3. $1.5 < Ntu_o < 500$, $C_r^* > 1$, $0.85 < C^* < 1$

Effect of longitudinal heat conduction

The effect of longitudinal heat conduction is to reduce the regenerator effectiveness for a specified Ntu_o. Two additional dimensionless parameters, λ and A_k^* now play a role. The following method was recommended by Shah[8] to account for the longitudinal heat conduction effect.

$$\varepsilon_{\lambda\neq 0} = C_\lambda \varepsilon_{\lambda=0}$$
$$\text{where } C_\lambda = \left(\frac{\varepsilon_{\infty,\lambda\neq 0}}{\varepsilon_{\infty,\lambda=0}}\right) \quad (9)$$

The effectiveness ratio in brackets is for a balanced recuperator. It is therefore postulated that the same ratio can be applied to unbalanced regenerators. The solution for the effectiveness of an unbalanced recuperator, including the effect of longitudinal heat conduction was first given by Hahnemann[9] and is reported again by Bahnke & Howard.[4]

Romie[10] suggested the following method to take into account the influence of longitudinal heat conduction. Ntu_o is replaced by $G_L Ntu_o$ in the calculations for zero longitudinal heat conduction. The factor G_L is derived from recuperator analysis and is given by:

$$G_L = \frac{1}{Ntu_o(1-C^*)} \ln\left(\frac{1-C^*\varepsilon_{\infty,\lambda\neq 0}}{1-C^*}\right) \quad (10)$$

For $C^* = (hA)^* = 1$ it reduces to $G_L = \varepsilon_{\infty,\lambda\neq 0}/(Ntu_o(1-\varepsilon_{\infty,\lambda\neq 0}))$. Good agreement was noted when the above method was compared to numerical results in the following range of parameters: $1 \leq Ntu_o \leq 500$, $0.7 \leq C^* \leq 1$, $1 \leq C_r^* \leq \infty$, $0.25 \leq ((hA)^* = A_k^*) \leq 4$, $0 \leq \lambda \leq 0.16$.

Effect of transverse heat conduction

An equivalent thermal resistance is added to the convective resistances in the expression for Ntu_o to account for the effect of wall resistance. Hausen[11] showed that this resistance can be expressed as:

$$R = \left(\frac{\delta}{6k_w}\right)\left(\frac{1}{A_h}+\frac{1}{A_c}\right)\phi \qquad (11)$$

Hausen[11] gives the following expressions for ϕ:

$$\phi = 1-(1/15)Z \qquad \text{for } Z \leq 5 \qquad (12)$$

$$\phi = 2.142(0.3+2Z)^{-0.5} \qquad \text{for } Z \geq 5 \qquad (13)$$

where

$$Z = \frac{\delta C_r}{2k_w}(1/A_h + 1/A_c)$$

Approximate solution method for an ideal regenerator

The following procedure is adopted and the predictions are in good agreement with numerical results.

1. The six dimensionless parameters

   $Ntu_o, C^*, C_r^*, (hA)^*, \lambda$ and A_k^*

   are calculated from the known design parameters and operating conditions. Ntu_o already includes the effect of transverse heat conduction.

2. The longitudinal heat conduction correction factor, G_L is calculated from eq. (10). Ntu_o is corrected by replacing it with $G_L Ntu_o$, Bahnke & Howard[4] presents the solution for $\varepsilon_{\infty,\lambda\neq 0}$.

3. The parameters $Ntu_{o,e}$ and $C_{r,e}^*$ for an equivalent balanced regenerator are evaluated using eq. (5).

4. The effectiveness of the equivalent balanced regenerator is calculated using eq. (7).

5. The effectiveness of the original unbalanced regenerator is finally calculated from eq. (8).

We have now established an approximate method that can be compared to numerical results. The comparison was done with good agreement noted over the following range of parameters:

$1 \leq Ntu_o \leq 500$, $0.7 \geq C^* \leq 1$, $1 \leq C_r^* \leq \infty$, $0.25 \leq ((hA)^* = A_k^*) \leq 4$, $0 \leq \lambda \leq 0.16$. The ranges are sufficient to cover most applications. The error was less than 1% within the following ranges:

1. $1 \leq Ntu_o \leq 500$, $0.7 \leq C^* \leq 1$, $2 \leq C_r^* \leq \infty$, $0.25 \leq (hA)^* \leq 4$, $0 \leq \lambda \leq 0.16$
2. $1 \leq Ntu_o \leq 500$, $0.8 \leq C^* \leq 1$, $1.5 \leq C_r^* \leq \infty$, $0.25 \leq (hA)^* \leq 4$, $0 \leq \lambda \leq 0.16$
3. $1 \leq Ntu_o \leq 500$, $0.7 \leq C^* \leq 1$, $1.5 \leq C_r^* \leq \infty$, $0.25 \leq (hA)^* \leq 1$, $0 \leq \lambda \leq 0.04$
4. $1 \leq Ntu_o \leq 15$, $0.9 \leq C^* \leq 1$, $1 \leq C_r^* \leq \infty$, $0.25 \leq (hA)^* \leq 4$, $0 \leq \lambda \leq 0.16$

Influence of pressure- and carryover leakage

Pressure leakage is the flow of gas past the seals from the cold side to the hot side. The driving force is the pressure difference between the cold gas stream and hot gas stream. The carry-over leakage refers to the transportation of gas trapped in the matrix void volume from the one side into the opposite gas stream owing to matrix rotation. Shah[12] presents a model to account for the pressure- and carry-over leakages simultaneously. The most important assumption is that pressure- and carry-over leakage takes place outside the regenerator disk. The actual outlet temperatures and heat transfer rate are then found by doing energy balances on the fluid streams entering and leaving the ideal regenerator.

Pressure drop evaluation

The expression for evaluating the pressure drop across a plate fin heat exchanger is given by Kays & London.[5]

$$\Delta p = \frac{G^2}{2}\left[\begin{array}{l} 4f\frac{L}{D_h}\left(\frac{1}{\rho_m}\right)+2\left(\frac{1}{\rho_o}-\frac{1}{\rho_i}\right) \\ +(1+K_c-\sigma^2)\frac{1}{\rho_i}-(1-K_e-\sigma^2)\frac{1}{\rho_o}\end{array}\right] \qquad (14)$$

Often in heat exchanger pressure drop analysis, the exit and entrance losses are accounted for in the friction factor. Eq. (14) then becomes:

$$\Delta p = \frac{G^2}{2}\left[4f_e\frac{L}{D_h}\left(\frac{1}{\rho_m}\right)+(1+\sigma^2)\left(\frac{1}{\rho_o}-\frac{1}{\rho_i}\right)\right] \qquad (15)$$

Here f_e is an effective friction factor.

Design optimisation

In this section it will be shown how the operating conditions can be optimised in a specific application using a particular heating surface. Afterwards five heating surfaces will be compared to each other at their respective optimum operating points. The heating surfaces were experimentally tested for pressure drop and heat transfer characteristics. The surfaces were tested at different spacings. The comparison will be done at a single spacing.

A PC software package was written for the thermal evaluation and design of a counterflow rotary regenerator. The program consists of two parts, namely, (1) Evaluation or Rating and (2) Optimisation of the Design. In order to rate a regenerator, it is required that the designer specifies the operating conditions (mass flow rates, inlet temperatures, inlet pressures, disk rotational speed), rotating disk dimensions (outer diameter, hub diameter, disk height, ratio of area split between hot and cold fluid) and the matrix properties (plate spacing, wall thickness, material thermophysical properties, heat transfer and pressure drop correlations for the matrix). The program then calculates the outlet gas temperatures, the total heat transfer rate, and the pressure drop across the regenerator.

In order to optimise the design the overall effectiveness or equivalently the total heat transfer rate desired is specified. The disk rotational speed, the hot to cold fluid flow split ratio and the matrix wall thickness are then varied. At each combination of the varied parameters the disk height is calculated that will deliver the required heat transfer rate.

Note: The disk outer diameter was not taken as an optimisation variable, because it was found that the largest possible frontal area always produces the most cost effective answer. It is therefore up to the designer to specify the largest possible outer diameter for the rotating disk which can be practically implemented in a system.

For illustrative purposes a single regenerator application was investigated. The regenerator data and operating conditions were obtained from Leong *et al.*[13] The pressure drop and heat transfer correlations were based on experiments conducted for this investigation.

Input data

Operating conditions

Cold gas inlet mass flow rate, $W_c = 109$ kg/s
Hot gas inlet mass flow rate, $W_h = 109$ kg/s
Cold gas inlet temperature, $T_{c,i} = 337$ K
Cold gas inlet pressure, $p_{c,i} = 106.21$ kPa
Hot gas inlet temperature, $T_{h,i} = 580.5$ K
Hot gas inlet pressure, $P_{h,i} = 102.28$ kPa
Pressure leakage fraction, $x_p = 0.08$ of cold gas flow rate

Disk dimensions

Disk outer diameter, $D = 7.863$ m
Hub diameter, $D_{hub} = 2.237$ m
Seal face coverage fraction, $x_s = 0.1$ of total frontal area

Matrix properties

Plate spacing, $s = 5$ mm
Matrix wall thickness, $\delta = 0.5$ mm
Matrix wall thermal conductivity, $k_w = 51.9$ W/mK
Matrix wall density, $\rho_w = 7\,817$ kg/m^3
Matrix wall specific heat, $c_w = 434$ J/kgK
Ratio of actual surface area to projected area, $\alpha = 1.2$

Other input parameters like the porosity, σ, the surface area density, β, and the effective mass of the matrix, m, can be calculated if the plate spacing, the plate thickness, the matrix wall density and α are known.

Optimisation of operating conditions

The disk rotational speed, the hot to cold gas flow split ratio and the plate thickness were varied in the optimisation process to arrive at their optimum values for a particular type of heating surface. The required effectiveness was specified as 75 %. The heat transfer and pressure drop correlations for the matrix used were:

$$Ny = \frac{hA}{k\ \mathrm{Pr}^{0.333} A_{fr}} = 45.165\, Ry^{0.8068} \sigma^{-1.8068} L \quad (16)$$

$$K = \frac{\Delta p \rho}{0.5 \times G^2} = 424.715\, Ry^{-0.2903} \sigma^{-1.7097} L \quad (17)$$

where $Ry = \frac{G}{\mu}$.

The correlations are only valid for a plate spacing (not the pitch) of 5 mm, but for any porosity, σ, and disk height, L.

Discussion

(i) Effect of disk rotational speed and matrix wall thickness

The effectiveness of an ideal regenerator (no pressure- or carry-over leakage) increases with an increase in rotational speed and asymptotically approaches that of a recuperator as predicted by eq. (4). This is also evident from numerical results. However, in an actual regenerator, an increase in rotational speed increases the gas

carry-over which in turn reduces the regenerator effectiveness. These two effects oppose each other. The effectiveness first increases with an increase in rotational speed and then decreases indicating an optimum value at some rotational speed. If the effectiveness is held constant as was done in this case the required surface area or disk height will show a minimum at some rotational speed. In Figure 1 it can be seen that there is a minimum disk height for every plate thickness as the disk rotational speed is increased. Three matrix wall thicknesses, namely 0.3, 0.5 and 0.7 mm were investigated. The optimum disk rotational speed is defined as the speed that requires the minimum disk height for a specified effectiveness.

In the literature it is indicated that regenerators usually operate at rotational speeds of 1 to 3 r.p.m.[14,15,16] For this example the optimum speeds are in the same range. The disk rotational speed is the most important optimisation variable if the regenerator matrix is fixed. The disk height increases drastically if the rotational speed is lowered beyond a certain value. The designer must be certain that the rotational speed is high enough to avoid this situation, but within limits to avoid excessive fluid intermixing.

(ii) Effect of the flow split ratio

A flow split ratio of 1.14 was used in Figure 1, being the optimum value obtained in the optimisation process. It was found that the flow split ratio does not have a significant influence on the minimum disk height or the rotational speed where this height occurs. The effect of a change in the flow split ratio on the regenerator effectiveness is mainly reflected by changes in the dimensionless groups $(hA)^*$ and A_k^*. It was noted earlier that the two above-named dimensionless groups do not have a significant influence on the regenerator effectiveness and can practically be eliminated from the solution process. The optimum disk height increased from 3.68 m to 3.69 m as the flow split ratio was increased from 1 to 1.2. For practical purposes this is insignificant.

A more important role that the flow split ratio plays in the optimisation process is to balance the pressure drop for the hot and cold sides. Figure 2 shows how the hot and cold side pressure drop are influenced by the flow split ratio. It can be seen that the pressure drops are balanced near the optimum ratio of 1.14. This has the advantage of reducing the overall fan sizes needed to supply the gas.

Heating surface selection

Five heating surfaces that were experimentally tested for their pressure drop and heat transfer characteristics will be compared at their respective optimum operating conditions. The heating surfaces are numbered from 1 to 5. Surface number 1 is a plain flat surface. Surfaces $2 - 5$ have wavy patterns to enhance the heat transfer. They are numbered in order of increasing heat transfer coefficient. Number 5 thus has the highest heat transfer coefficient. Surface number 2 was used to illustrate the optimisation of the operating conditions. For this range of surfaces it was found that the ratio h/f decreased as the heat transfer coefficient was increased. If this ratio is taken as the only performance criteria, the flat surface seems to be the best choice. The physical size and operating costs are two additional factors that the designer will have to consider.

Figure 3 shows a comparison of the required disk height for the five surfaces. The results are shown at the optimum rotational speed for each surface. The heat transfer rate and operating conditions are the same for all the surfaces. The size of a regenerator fitted with the flat surface, will exceed that of any of the enhanced surfaces by far. The plate thickness does not have a significant influence on the required disk height.

Figure 4 shows the average pressure drop for the five different matrixes. It can be seen that although surface number 1 (the flat surface) requires the largest disk dimensions, its pressure drop is the lowest. Surface number 5 which requires the least matrix material, has the highest pressure drop. The pressure drop increases slightly with an increase in plate thickness.

Figure 5 shows the disk rotational speed at the minimum disk height for each matrix. One of the major disadvantages of a rotary regenerator is the intermixing of the two gas streams. In applications where this is not allowable, a rotary regenerator can not be used. A large disk height and a high rotational speed contribute to intermixing. It is interesting to note that in the cases where the disk height is the largest, the rotational speed is the lowest, This is an advantage, since it helps to limit gas intermixing.

Concluding remarks

It was shown how the some of the operating conditions of a rotary regenerator can be optimised for a particular choice of matrix. To choose a matrix that is best suitable for an application is perhaps more difficult. The regenerator dimensions and pressure drop vary drastically with choice of matrix. The final selection in most cases will have to be made after an economic evaluation. A detailed cost analysis will have to be carried out, taking into account the capital and operating costs of the regenerator over its life span.

Such an analysis falls beyond the scope of this work. The spacing between the sheets is another important design parameter. Although the results were presented for

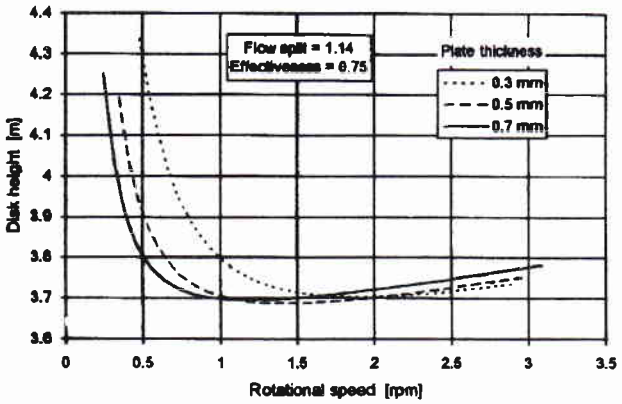


Figure 1 Required disk height at different rotational speeds

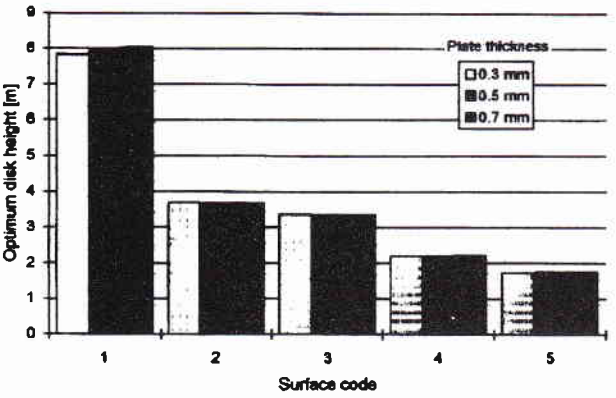


Figure 3 Optimum disk height for the various surfaces

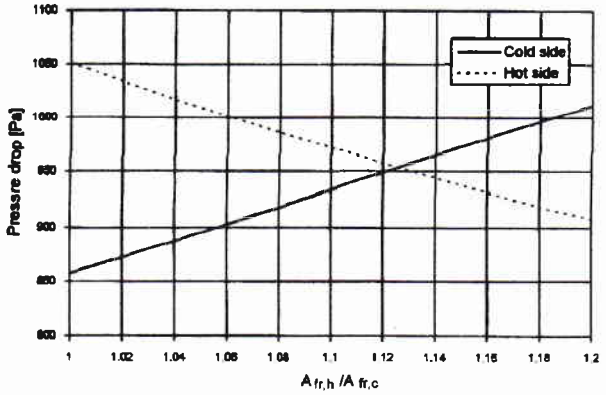


Figure 2 Hot and cold side pressure drop at different flow split ratios

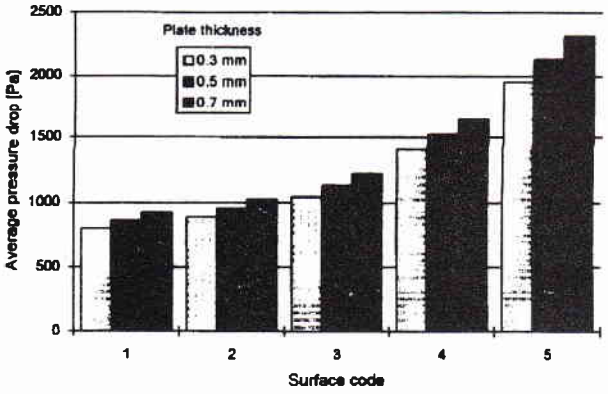


Figure 4 Pressure drop at the optimum disk height for the different surfaces

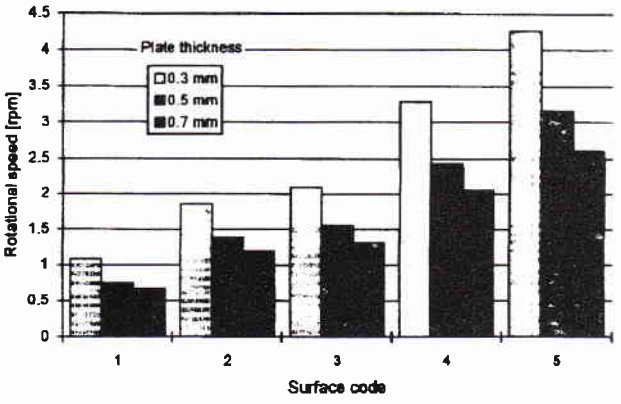


Figure 5 Optimal rotational speed for the different surfaces

a spacing of 5 mm, the surfaces were tested at spacings of 3.5 and 3 mm as well.

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