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The Effect of Fresh Air Purging on the Effectiveness of Rotary Regenerators for Energy Recovery from Exhaust Air

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THE EFFECT OF FRESH AIR PURGING ON THE EFFECTIVENESS OF ROTARY REGENERATORS FOR ENERGY RECOVERY FROM EXHAUST AIR

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Abstract

In order to prevent carry-over of contaminated exhaust air to the fresh air in rotary heat regenerators, a small part of the fresh air is used for purging the heat transfer matrix during its transfer from the hot to the cold zone. This purging gives rise to a loss in the effectiveness of the regenerator.

A simple numerical scheme has been devised for computing the effectiveness of rotary regenerators, and the effect of fresh air purging has been computed over a wide parameter range. Besides in graphical representation the results are given in the form of approximate algebraic expressions.

INFLUENCE DE LA PURGE D'AIR NEUF SUR L'EFFICACITE DES REGENERATEURS ROTATIFS DANS LA RECUPERATION D'ENERGIE SUR L'AIR EXTRAIT

RESUME : Pour éviter l'entraînement d'air extrait contaminé avec l'air neuf des régénérateurs de chaleur rotatifs, on utilise une petite partie de l'air neuf pour purger la matrice de transfert de chaleur au cours de son passage de la zone chaude à la zone froide. Cette purge provoque une perte d'efficacité du régénérateur.

On a conçu un système numérique simple pour calculer l'efficacité des régénérateurs rotatifs et l'influence de la purge d'air neuf a été calculée dans une large gamme de paramètres. De plus, dans la représentation graphique, les résultats sont fournis sous forme d'expressions algébriques approchées.

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INTRODUCTION

For heat recovery from exhaust air rotary regenerators have certain advantages compared with ordinary heat exchangers (recuperators): an extremely compact and relatively inexpensive heat transfer surface can be used; and the amount of heat transferred can be regulated within rather wide limits simply by adjusting the speed of rotation of the heat transfer matrix. However, since there is no rigid separation between the two air streams some carry-over from one stream to the other will occur. In most space-heating applications contamination of the fresh air with exhaust air cannot be tolerated so, in order to avoid such contamination, a small section of the regenerator between the hot and the cold zone is purged with fresh air (see Figure 1). Because some of the fresh air is thus being discarded after being heated the purging entails an energy loss.

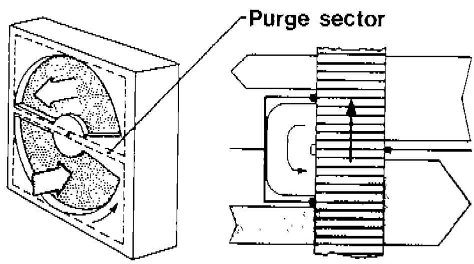

Fig. 1. Rotary regenerator with purging of the heat transfer matrix.

The effectiveness of periodic regenerators under conditions where the temperature gradient in the matrix material perpendicular to the surface as well as longitudinal heat conduction can be neglected was computed by Lambertson /1/. The results were presented in a wider context by Kays & London /2/. For the applications originally aimed at, preheating of combustion air for gas turbines, a small leakage from the hot to the cold side and vice versa is not particularly harmful. Hence, the question of purging the matrix did not enter into these computations. In the present paper a numerical method somewhat similar to the one employed by Lambertson is used for evaluating the effect on the heat transfer efficiency of using a small fraction of the fresh air for purging the matrix between the hot and the cold zone.

ANALYSIS

The heat transfer matrix of rotary regenerators for sensible heat recovery in HVAC applications is usually made of alternate layers of plain and corrugated thin aluminium plate. For such a matrix the temperature variation through the plate (perpendicular to the surface) is negligible and, since a very high effectiveness is not aimed at, the influence of axial conduction in the matrix may also safely be disregarded. Under these circumstances the regenerator may be viewed as two coupled cross-flow heat exchangers in each of which heat is exchanged between a gas (air)

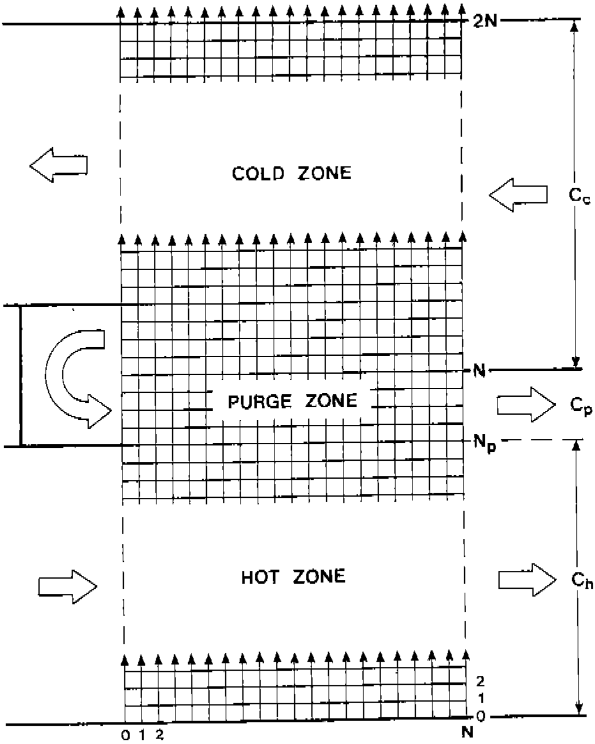

Fig. 2. Schematic representation of rotary regenerator showing hot and cold streams and matrix "stream" divided into flow channels.

stream and a "stream" of matrix material as shown in Figure 2. If the two gas streams and the matrix stream are each divided into N separate flow channels each of the two heat exchangers will be composed of N^2 elemental cross-flow heat exchangers. With the usual assumptions of constant specific heat and uniform heat transfer coefficient in the flow direction the temperature change across each of the elemental heat exchangers can be computed from the temperature effectiveness of a cross-flow heat exchanger. Hence, the temperature fields can be computed recursively from the following expressions in which non-dimensional temperatures have been introduced (see Figure 3).

Hot zone $(0 \leq n < N - N_p)$:

$$\Theta_{m+1,n-\frac{1}{2}} = \Theta_{m,n+\frac{1}{2}} - E_{hM}(\Theta_{m,1+\frac{1}{2}} - \Theta_{m+\frac{1}{2},n}) \tag{1a}$$

$$\Theta_{m+\frac{1}{2},n+1} = \Theta_{m+\frac{1}{2},n} + E_{Mh}(\Theta_{m,n+\frac{1}{2}} - \Theta_{m+\frac{1}{2},n}) \tag{1b}$$

$$\Theta_{0,n+\frac{1}{2}} = 1 \tag{1c}$$

$$\Theta_{m+\frac{1}{2},0} = \Theta_{m+\frac{1}{2},2N} \qquad (0 \leq m < N) \tag{1d}$$

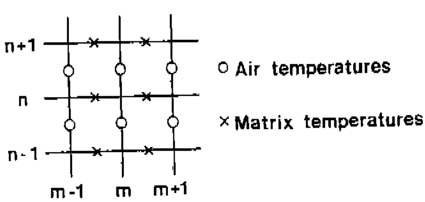

Fig. 3. Node points for numerical computation of temperature fields. Values at mid-points between node points represent mean values in the individual flow channels.

Purge zone $(N - N_p \leq n < N)$:

$$\Theta_{m+1,n+\frac{1}{2}} = \Theta_{m,n+\frac{1}{2}} - E_{cM}(\Theta_{m,n+\frac{1}{2}} - \Theta_{m+\frac{1}{2},n}) \tag{2a}$$

$$\Theta_{m+\frac{1}{2},n+1} = \Theta_{m+\frac{1}{2},n} + E_{Mc}(\Theta_{m,n+\frac{1}{2}} - \Theta_{m+\frac{1}{2},n}) \tag{2b}$$

$$\Theta_{m+\frac{1}{2},0} = \bar{\Theta}_p \tag{2c}$$

Cold zone $(N \leq n < 2N)$:

$$\Theta_{m-1,n+\frac{1}{2}} = \Theta_{m,n+\frac{1}{2}} - E_{cM}(\Theta_{m,n+\frac{1}{2}} - \Theta_{m-\frac{1}{2},n}) \tag{3a}$$

$$\Theta_{m-\frac{1}{2},n+1} = \Theta_{m-\frac{1}{2},n} + E_{Mc}(\Theta_{m,n+\frac{1}{2}} - \Theta_{m-\frac{1}{2},n}) \tag{3b}$$

$$\Theta_{N,n+\frac{1}{2}} = 0 \tag{3c}$$

The temperature effectiveness E was computed from the expressions for cross-flow heat exchangers with both streams unmixed derived by Stevens /3/ (reproduced in ref. 4). In determining E it was assumed that the ratio $(hA)_h/(hA)_c$ was equal to the ratio of thermal capacity rates C_h/C_c. In the purge zone the temperature effectiveness of the cold zone was used which implies the assumption that the mass flow rate and the heat transfer coefficient equalled those of the cold zone. When no specific information is available on how the pressure drop across the heat transfer matrix is overcome, this appears to be the most reasonable assumption.

At the start of the compution the temperature of the heat transfer matrix where it enters the hot zone is unknown and must be estimated. When the computation is completed the matrix temperature at the exit from the cold zone is fed back as starting temperature of the hot zone. This iterative procedure was found to converge satisfactorily.

For applications where the purpose is heat recovery from exhaust air the regenerator effectiveness is customarily defined as the ratio of the heat actually transferred to the fresh air to the maximum energy available for transfer /5/

$$\varepsilon = \frac{C_c(T_{co} - T_{ci})}{C_h(T_{hi} - T_{ci})} \tag{4}$$

Due to parasitic air losses in the ventilation system the exhaust flow rate is usually smaller than that of the fresh air. Under these circumstances the above definition is equivalent to the usual definition of the heat transfer effectiveness /2/.

When purging takes place the effectiveness relates to the part of the fresh air stream that is actually utilized

$$\varepsilon = \frac{C_c - C_p}{C_h}\bar{\Theta}_c = (1 - \alpha)\frac{C_c}{C_h}\bar{\Theta}_c \tag{5}$$

where $\bar{\Theta}_c$ is the mean exit temperature of the fresh air stream not used for purging.

NUMERICAL RESULTS

The effectiveness of rotary regenerators have been computed for the following values of the pertinent parameters.

N_{TU}: 0.2, 0.5, 1, 2, 5, 10

C_{min}/C_{max}: 1, 0.9, 0.8, 0.7, 0.6, 0.5

C_M/C_{min}: 1, 1.2, 1.5, 2, 3, 5

C_p/C_c : 0, 0.05, 0.10

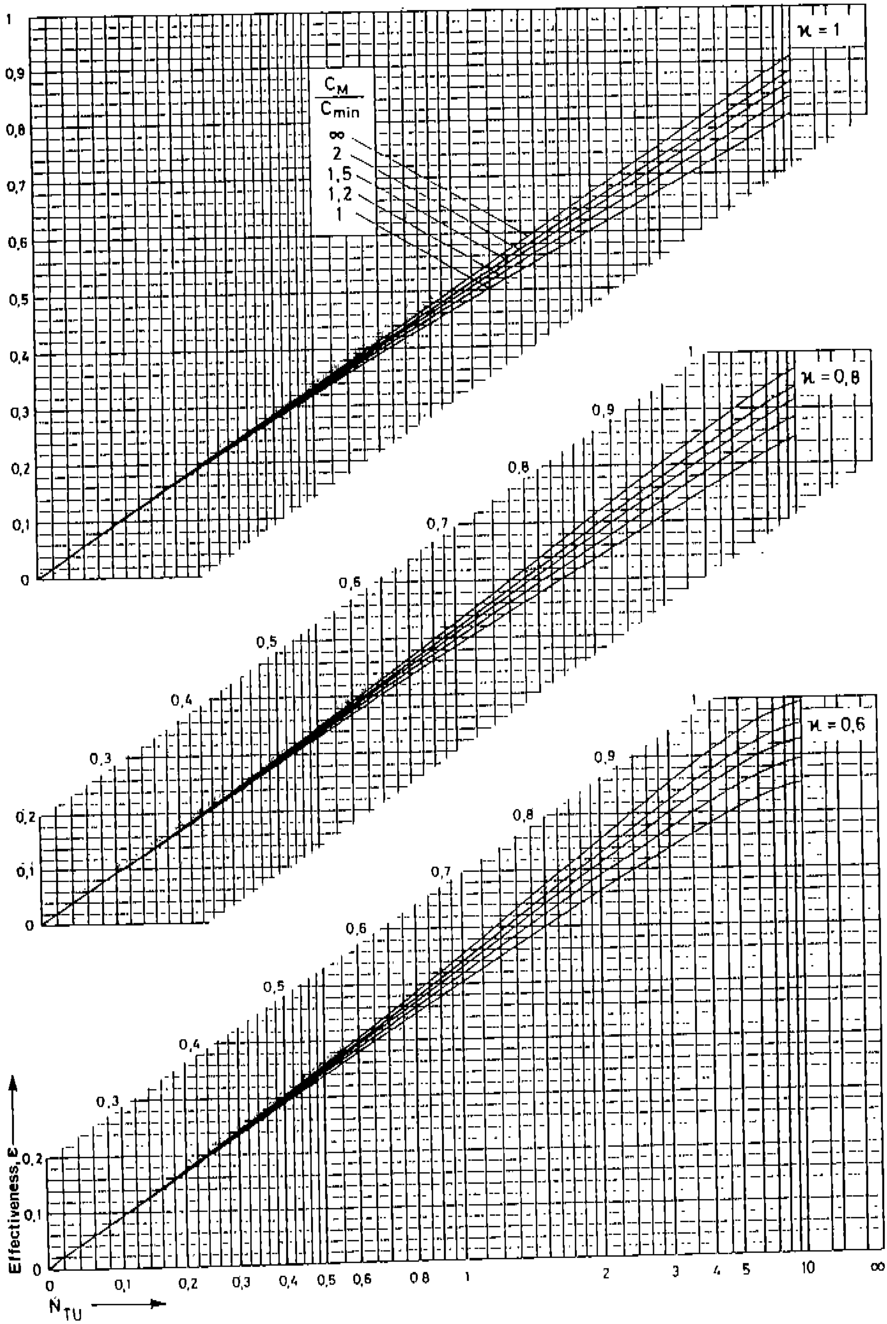

Fig. 4a-c. Heat transfer effectiveness of rotary regenerators computed from equation 6 for C_{min}/C_{max} (κ in the figure) = 1, 0.8, and 0.6.

The complete numerical results can be found in ref. 6. The effectiveness of rotary regenerators when no purging takes place are also presented by Kays & London /2/.

In Figures 4a-c the effectiveness of rotary regenerators when no purging or carry-over occurs is shown for selected values of C_{min}/C_{max} and C_M/C_{min}. The graphical representation is one that was proposed previously for direct-type heat exchangers /7/ in which the abscissa is $\alpha = N_{TU}/(1 + N_{TU})$. Since this is the expression for the effectiveness of a direct-type counter-flow heat exchanger with equal thermal capacity rates, the latter is represented by a diagonal line in the diagram. With α going from zero to unity the entire range of N_{TU} from zero to infinity is included without sacrificing the reading accuracy in the range of greatest interest, from 0.2 up to, say, 5.

The effectiveness of rotary regenerators can also be found from an empirical expression which relates to the effectiveness to that of a counter-flow heat exchanger of the direct type. Improving on a simple expression proposed by Kays & London /2/, the present author found that the following relation approximates the numerically computed values within 1-2 percent.

$$\varepsilon/\varepsilon_{cf} = 1 - \frac{0.114(1 - e^{-N_{TU}})}{(C_{min}/C_{max})^{0.44}(C_M/C_{min})^{1.93}} \qquad (6)$$

From the numerical results of the present work it appeared that for the range of N_{TU} which is relevant in this context, $0.5 \leq N_{TU} \leq 5$, the ratio of the effectiveness with purging to that without, $\varepsilon_\alpha/\varepsilon_0$, did not vary significantly with N_{TU}. Furthermore to a good approximation this ratio is given by the empirical relation

$$\varepsilon_\alpha/\varepsilon_0 = (1 - \alpha)\exp(- b\alpha)$$

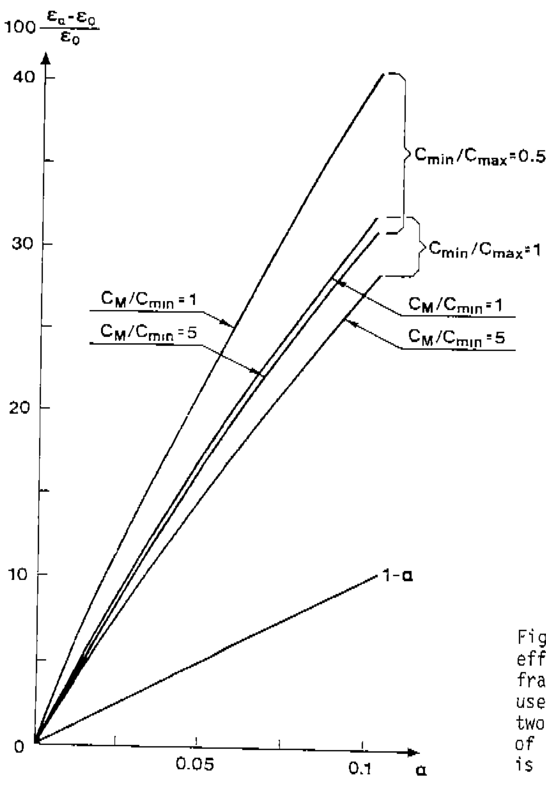

Fig. 5. Decrease in regenerator effectiveness as a function of the fraction of the fresh air stream used for purging. For each of the two values of C_{min}/C_{max} the range of values for $1 \leq C_M/C_{min} \leq 5$ is shown.

Here b is a function of C_{min}/C_{max} and C_M/C_{min} which is closely approximated by the following expression

$$b = 2.45 - 0.40C_{min}/C_{max} + 0.69\ (C_M/C_{min})^{-1}(C_{min}/C_{max})^{-4/3} \qquad (8)$$

DISCUSSION

The loss in effectiveness caused by purging of the heat transfer matrix is illustrated in Figure 5. From this figure it is evident that the loss is considerably larger than what corresponds to the rejection of a fraction of the fresh air if it were heated only to the mean exit temperature, the latter loss being $1 - \alpha$. The exponential function in equation 7 accounts for the extra loss that arises from the fact that it is precisely the air which has been heated to the highest temperature that is used for purging and, during the return flow through the heat transfer matrix, it extracts even more heat from the exhaust stream. It should be noted that the computations have been carried out under the assumption that the transfer area on both the hot and the cold side remains constant. If the sector used for the return flow had been taken from either the hot or the cold sector the loss in effectiveness would have been even greater.

From Figure 5 it also appears that the loss is reduced when the speed of rotation of the matrix is increased. The reason for this is that with increased matrix capacity flow the exit temperature on the cold side tends towards being more uniform. The effect, however, is rather modest, particularly when one considers that for $C_M/C_{min} = 5$ the effectiveness of a regenerator with no purging practically has reached the limiting value corresponding to that of a direct type counter flow heat exchanger. That the loss increases percentagewise when the exhaust flow rate is reduced relative to that of the fresh air is not surprising.

SUMMARY

By numerical computations it has been demonstrated that for rotary regenerators used for energy recovery from exhaust air the loss which is caused by the need for purging the heat transfer matrix to a good approximation can be expressed by a simple empirical expression. For $0.5 \leq N_{TU} \leq 5$ the reduction in effectiveness does not depend significantly on N_{TU}:

$$\varepsilon_\alpha/\varepsilon_0 = (1 - \alpha)\exp(-b\alpha)$$

where $b = 2.45 - 0.40\ C_{min}/C_{max} - 0.69(C_M/C_{min})^{-1}(C_{min}/C_{max})^{-4/3}$

The factor $1 - \alpha$ represents the loss that would be incurred if a fraction α of the fresh air was discarded after being heated to the mean exit temperature. The exponential factor accounts for the fact that the fresh air used for purging has been heated to a temperature above the average.

NOMENCLATURE

- A — heat transfer area
- b — defined by equations 6 and 7
- C — capacity flow rate
- C_{min} = min C_h, C_c
- C_{max} = max C_h, C_c
- h — heat transfer coefficient
- E — temperature effectiveness of elemental heat exchangers
- N — total number of flow channels in hot zone (including purge zone), cold zone and matrix "stream"
- N_p — number of flow channels in purge zone
- N_{TU} = $[1/(HA)_h + 1/(hA_c)]^{-1}/C_{min}$ number of transfer units
- T — temperature

Greek letters

α = C_p/C_c fraction of fresh air used for purging

ε regenerator effectiveness (see equation 4)

ε_α regenerator effectiveness with purge air fraction α

ε_0 regenerator effectiveness with no purging

Θ = $(T_a - T_{ci})/(T_{hi} - T_{ci})$ non-dimensional air temperature

$\bar{\Theta}$ = Average exit temperature

ϕ = $(T_M - T_{ci})/(T_{hi} - T_{ci})$ non-dimensional matrix temperature

Indices

a air (hot or cold stream)

c cold stream or cold zone

cf counter-flow

h hot stream or hot zone

i inlet

m,n see Figure 3

o outlet

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