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Theory and Design of Rotary Regenerators for Air Conditioning

BY

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Theory and Design of Rotary Regenerators for Air Conditioning

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mary.—The heat transfer, mass transfer and pressure drop charac- of rotary regenerative sensible heat exchangers or regenerators are 1. The literature on the theory of heat exchange by regenerators 'ed briefly. The dimensionless parameters describing regenerator ince are expressed in terms of the important design variables to the comparison of alternative designs. Effectivity charts are d for balanced and unbalanced flow.

: advantages of the parallel plate regenerator developed by the authors ssed and a simple design procedure is given.

t recovery from exhaust air in air conditioning systems using generators is common in the United States of America and Canada shown that installations in Australia are well justified economically.

LIST OF SYMBOLS

Thickness of plate in parallel plate matrix (m.).

Area of matrix heat transfer surface exposed to a gas stream (m.^{2}).

Frontal area of matrix through which gas flow is prevented by seals (m.^{2}).

$= \varepsilon A_{fr}$ free flow area of matrix through which a gas stream flows (m.^{2}).

Frontal area of matrix through which a gas stream flows (m.^{2}).

Total frontal area of matrix ($A_{ft} = A_{fr,1} + A_{fr,2} + A_{bs}$), (m.^{2}).

Area of gap in clearance seals (m.^{2}).

Spacing between plates in a parallel plate matrix (m.)

Specific heat of matrix heat storage material (J kg.$^{-1}$ K^{-1}).

Specific heat at constant pressure of gas stream (J kg.$^{-1}$ K^{-1}).

$\rho c_p Q$ gas stream capacity rate (W K^{-1}).

Clearance seal coefficient of discharge (dimensionless).

$= c_m Mn$ matrix capacity rate (W K^{-1}).

$= 4A_c L/A$ hydraulic diameter of matrix passages (m.).

Friction factor based on mean shear stress (dimensionless).

Friction factor for fully developed flow (dimensionless).

Heat transfer coefficient from gas stream to matrix (W m.$^{-2}$ K^{-1}).

Pressure drop coefficient for entrance effect in passage (dimensionless).

Depth of matrix in direction of flow (m.^{2}).

Total mass of matrix (kgm.).

Rotational speed of matrix (Hz)

$1/\{C_1 [1/(h_1 A_1) + 1/(h_2 A_2)]\}$ number of heat transfer units of regenerator (dimensionless).

uA_c volume flow rate of gas stream (m.^{3} sec.$^{-1}$).

Minimum volume flow rate of gas streams (m.^{3} sec.$^{-1}$).

Seal leakage volume flow rate (m.^{3} sec.$^{-1}$).

$= uD_h/\nu$ Reynolds number of flow through matrix (dimensionless).

$= h/\rho c_p u$ Stanton number of flow through matrix (dimensionless).

Bulk mean temperature of flow stream (K).

Mean velocity in matrix passages (m. sec.$^{-1}$).

$n\varepsilon A_{ft} L/Q_a$ carry-over ratio (dimensionless).

Pressure drop across matrix passages (N m.$^{-2}$).

Pressure drop across clearance seal (N m.$^{-2}$).

Porosity or void fraction of matrix (for parallel plates $\varepsilon = b/(a + b)$) (dimensionless).

$(t_{1,out} - t_{1,in})/(t_{2,in} - t_{1,in})$ heat transfer effectivity (effectiveness or efficiency) (dimensionless).

Kinematic viscosity (m.^{2} sec.$^{-1}$).

Density of gas stream (kg. m.$^{-3}$).

Density of matrix material (kg. m.$^{-3}$).

A_{fr}/A_{ft} matrix flow fraction ($\phi_1 + \phi_2 = 1 - A_{bs}/A_{ft}$) (dimensionless).

lthough S.I. units are given above, the dimensions of quantities the equations may be in any consistent system of units except Eqs. (11), (12), (13), (14) and (15) where the units are given in ie text.

2. The subscript 1 denotes a quantity of the gas stream with the lesser value of C. These subscripts are applied to the quantities A, A_c, A_{fr}, C, h, Q, t, u, Δp and φ when they differ for the two gas streams.
3. In the definition of the areas A_{bs}, A_{fr}, A_{ft} and the porosity ε, the matrix consists of the porous material intended to store heat only and does not include the hub, spokes, rim or other components of the frame in which the matrix is mounted.

1.—INTRODUCTION

In fresh-air heat recovery or regenerative evaporating cooling systems for air conditioning, conditions are less severe and temperature differences are smaller than for other applications of rotary regenerative heat exchangers or regenerators. Smaller temperature differences result in less energy saved to offset the capital cost. In the United States of America and Canada the Ljungström rotary regenerator with wire mesh matrix has been used for air conditioning applications (Refs. 1, 2, 3 and 4) but smaller and cheaper designs seem desirable.

The Division of Mechanical Engineering of C.S.I.R.O. (D.M.E.) has been investigating the application of three types of rotary heat and moisture regenerators to air conditioning, and attempting to improve their design for some years (Refs. 5, 6, 7 and 8). The three types are; the rotary sensible heat regenerator with a high heat and low moisture storage capacity, the rotary total heat regenerator with high heat and high moisture storage capacity and the rotary de-humidifier with low heat and high moisture storage capacity. Only rotary sensible heat regenerators will be considered below and they will be referred to as regenerators. D.M.E. has experimented with a variety of regenerators. Of these, the parallel plate laminar counter flow type is the most promising. This regenerator is made by winding a reel of polyester film spirally over aluminium spacers which are retained in spokes. A parallel plate regenerator of this type is shown in Fig. 1.

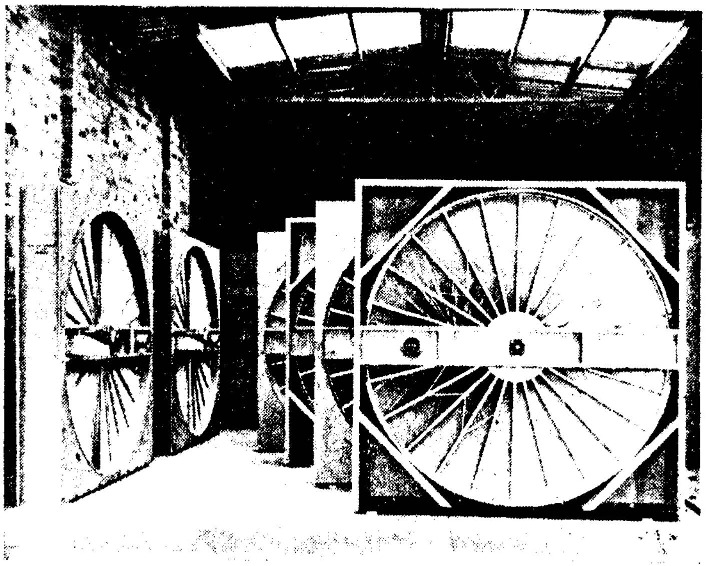

Fig. 1.—78-in. dia. Parallel Plate Rotary Regenerator made by Rotary Heat Exchangers Pty. Ltd., Vermont, Victoria.

This design was selected for study because of several advantages over other systems, namely: Compactness, light weight, simplicity of design, construction and installation, low cost and availability of materials, smooth flow passages with low dust deposition and low water vapour transfer between air streams. Parallel plate matrix passages have the highest Stanton number friction factor ratio which is the most important measure of the heat transfer and pressure drop performance of passages (Section 4).

2.—REVIEW OF REGENERATOR THEORY

Heat transfer in regenerators takes place by the matrix of heat storage material absorbing heat from the hot gas stream and subsequently transferring it to the cold gas stream. The matrix is mounted in a frame which is rotated in a housing. The housing has seals to separate the gas streams

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nd ensure they pass through the matrix. Heat transfer also takes place y leakage of fluid through the seals but this is considered in Section 6. 'he matrix and frame usually form a disc or wheel (Fig. 1) with axial flow hrough the matrix or a drum with radial flow.

The history of the theory up to 1950 is discussed in the books by Iausen (Ref. 9) and Jakob (Ref. 10). The primary problem of the theory ; the calculation of heat transfer effectivity given the heat transfer coefficient rom gas to matrix, the properties of the gas and matrix, and the gas flow ates. For parallel or co-current flow Hausen (Ref. 9) and Kardas (Ref. 11) ave obtained analytical solutions which allow relatively easy calculation of ffectivity. For counter flow, which is usual for regenerators, the problem ; more difficult. There are two common approaches to the solution of ie counter flow problem.

In the first approach the problem is reduced to the simultaneous olution of two integral equations, which is readily done numerically on a igital computer. This method has been used by Nahavandi and Weinstein Ref. 12) and Kohlmayr (Ref. 13) has suggested some improvements to the iethod. This method requires fewer calculations (Ref. 12) than the ollowing method.

Alternatively, the partial differential equations of the regenerator may e solved by finite difference methods using an electronic computer. This :cond approach has been used by Lambertson (Ref. 14), Bahnke and Howard Ref. 15), Willmott (Ref. 16) and others. This method is simpler to program r an electronic computer, which is important if the designer is writing his wn program and he needs to know the temperature distributions (Ref. 17) hich are not usually given in the literature.

Most authors have made the same assumptions in the analysis of :generators, but there have been some differences in the treatment of iermal conduction in the matrix. The simpler treatments assume the onductivity into the matrix is infinite and the conductivity in the direction f fluid flow is zero. These assumptions are satisfactory for wire matrices ut the first is invalid for brick chequerwork in Cowper stoves and the second r some metal matrices. A solution without the first assumption has been ven by Kardas (Ref. 11) for parallel flow and approximately by Hausen Ref. 9) for counter flow. Bahnke and Howard (Ref. 15) have given umerical solutions without the second assumption for counter flow.

Granville et al (Ref. 18) have compared the above theory with exper ments on fixed bed regenerators. No similar comparisons are availabl in the literature for rotary regenerators.

Dimensionless parameters are used to present the performance c regenerators. The number of parameters required increases with th complexity of the mathematical model. The parameters used by Kay and London (Ref. 19), η, N, C_1/C_2, C_r/C_1, and $h_1 A_1/h_2 A_2$, are use below. They have the advantage that when $0.25 < h_1 A_1/h_2 A_2 <$ 4 $h_1 A_1/h_2 A_2$ may be neglected so the heat transfer coefficients will onl appear in N. N is the most important parameter determining the effec tivity, η. C_1/C_2 gives the effect of unbalanced flow on η while C_r/C_1 give the effect of speed of rotation. If $C_1/C_2 = 1$ the regenerator is said to hav balanced flow and if $A_1/A_2 = 1$ it is said to be symmetric.

The book by Kays and London (Ref. 19) gives tables and charts o effectivity and also transient response data, physical properties, frictio factors and Stanton numbers. Jakob (Ref. 10) gives a chart (page 299 of the effectivity of balanced symmetric parallel flow regenerators in term of the parameters $\Lambda = 2N$ and $\Pi = 2N/(C_r/C_1)$. Bahnke and Howar (Ref. 15) gives tables and charts for counter flow regenerators with conduc tion in the flow direction. Peiser and Lehner (Ref. 20) gives a char of the temperature swing or variation across the outlet of a balanced sym metric counter flow regenerator. They use the parameters $S = 2N$ and $P = 2N/(C_r/C_1)$. Near $C_r/C_1 = 1$ they used an approximate metho which gives values of effectivity 5% lower and 1% higher than more accurat finite difference methods (Refs. 14 and 15). Figs. 2 and 3 were prepare from (Refs. 14, 15 and 19) and give effectivity for balanced and unbalance counter flow.

The following discussion, except where stated to the contrary, wil be for balanced flow symmetric regenerators with the same gas for bot streams. The generalization to unbalanced flow unsymmetrical regenerator with different gases for each stream is straight forward.

3.—PRESSURE DROP

The pressure drop Δp across the regenerator passages may be deter mined from:

$$\Delta p = \frac{f \rho u^2 A}{2 A_c} \qquad \text{.................................(1}$$

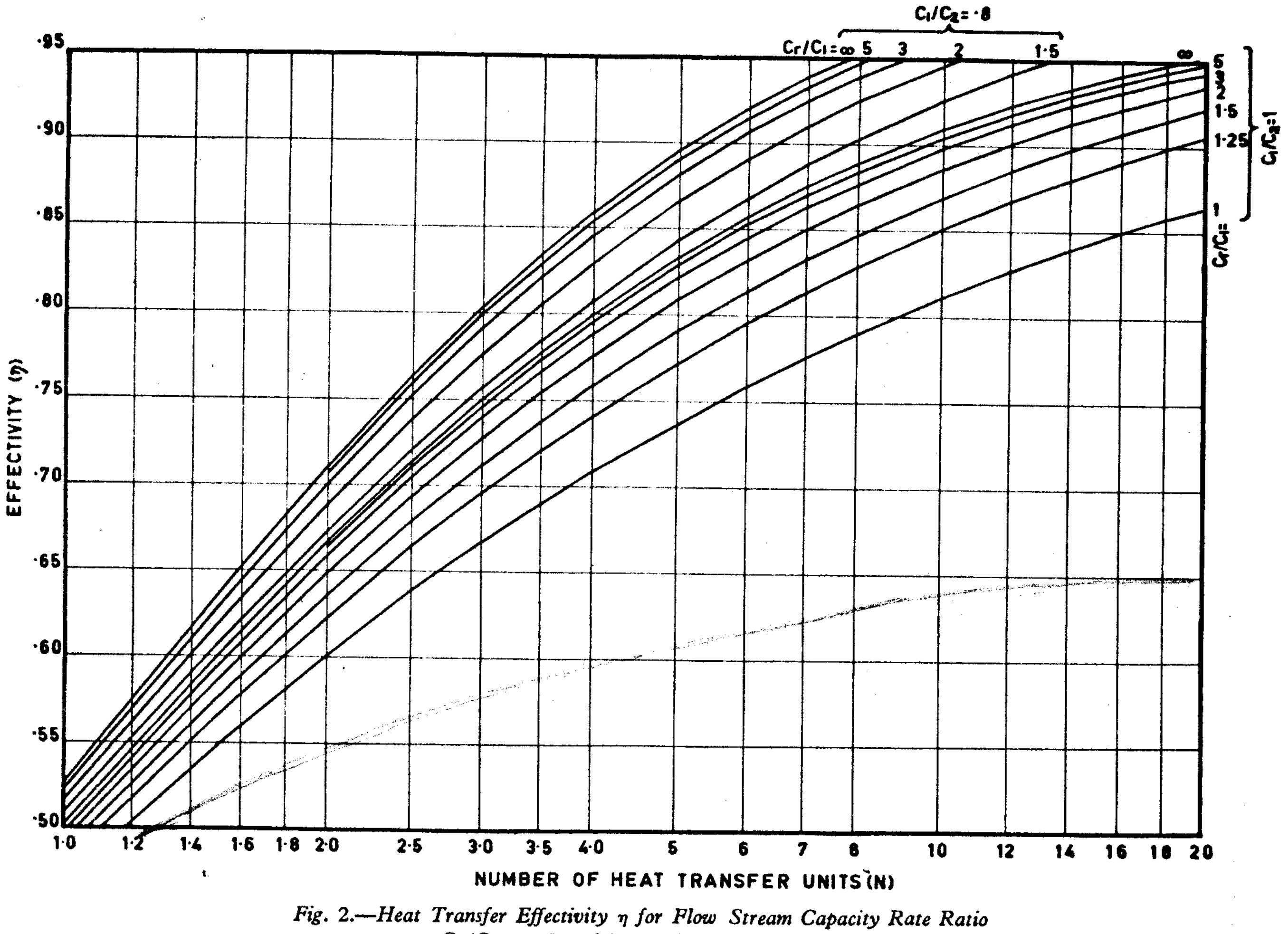


Fig. 2.—Heat Transfer Effectivity η for Flow Stream Capacity Rate Ratio $C_1/C_2 = 1.0$ and 0.8, and $h_1 A_1/h_2 A_2 = 1$.

For laminar flow it is convenient to separate the entrance effect from the rest of the equation so that Eq. (1) becomes:

$$\Delta p = \frac{f_\infty \rho u^2 A}{2A_c} + \frac{K\rho u^2}{2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

To obtain the total pressure drop across the regenerator, the pressure losses outside the matrix must be added to the above formulae, but they are usually small. The effect of flow acceleration due to density changes has also been neglected since density changes are small in air conditioning. Values of f and f_∞ are given in (Ref. 19) and values of K in (Ref. 21). For parallel plates $f_\infty Re = 24$ and $K = 0.686$.

4.—HEAT TRANSFER

The definition of N given in the list of symbols is not a convenient form for design work and it may be transformed as follows. It has been assumed above that $C_1 = C_2 = C$, $A_1 = A_2$ and the same gasses are used for each stream. Thus $h_1 = h_2 = h$. Substituting these in the definition of N gives:

$$N = \frac{hA}{2C} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

Substituting for C and introducing St from the list of symbols and A/A_c from Eq. (1) gives:

$$N = \frac{St\,\Delta p}{f\rho u^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

Substituting for u from the list of symbols this becomes

$$N = \frac{St\varepsilon^2 \phi^2 A_{ft}^2 \Delta p}{f\rho Q^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

If two regenerators, with different values of Stanton number friction factor ratio St/f but the same values of N, Q, A_{ft}, ϕ, ε and ρ, are compared, Eq. (5) shows that the regenerator with the highest value of St/f has the lowest pressure drop Δp. St/f depends on the Prandtl and Reynolds numbers and the shape of the passages. It is usually independent of Reynolds number in the laminar and turbulent regimes and changes on slightly at transition. Entrance effects are usually smaller for St/f tha for St or f alone. St/f is thus a useful measure of the performance of tl passages of a matrix. Table I gives values of St/f for some commo matrices. For other matrices values may be obtained from Ref. (19).

If two regenerators, with different values of St/f and porosity ε b the same values of N, Q, A_{ft}, ϕ and ρ, are compared, Eq. (5) shows th the regenerator with the highest value of $\varepsilon^2 St/f$ has the lowest Δp. $\varepsilon^2 St$ depends on the porosity of the matrix, Prandtl and Reynolds numbers an the shape of the passages. $\varepsilon^2 St/f$ is thus a measure of the performance the matrix. A high value of porosity is desirable. The low fixed valu of porosity for crushed rock and randomly packed spheres (Table I) are further disadvantage of these matrices.

TABLE I

Stanton Number Friction Factor Ratio (St/f) Prandtl Number 0.70

Matrix	ε	Re	St/f	Refs.
Parallel plates	—	—	0.484	(19)
Circular duct	—	—	0.386	(19)
Equilateral triangular duct	—	—	0.318	(19)
Randomly stacked wire mesh	0.832	300	0.19	(19)
Randomly packed spheres	0.38	100	0.064	(19)
Crushed rock	0.47	100	0.015	(22)

5.—MASS TRANSFER

For air conditioning regenerators, mass transfer refers to the transfe from stream to stream of contaminants or water vapour or both. Mas transfer takes place due to absorption or adsorption in the heat storag material of the matrix or in dust deposited on the matrix, carry-over c fluid in the voids of the matrix from one stream to the other, seal leakag and in some cases by condensation and evaporation in the matrix.

Rotary sensible heat regenerators are made of material such as alu minium wire and polyester film with negligible contaminant and moistur adsorption. Experiments conducted by D.M.E. have shown that nylo

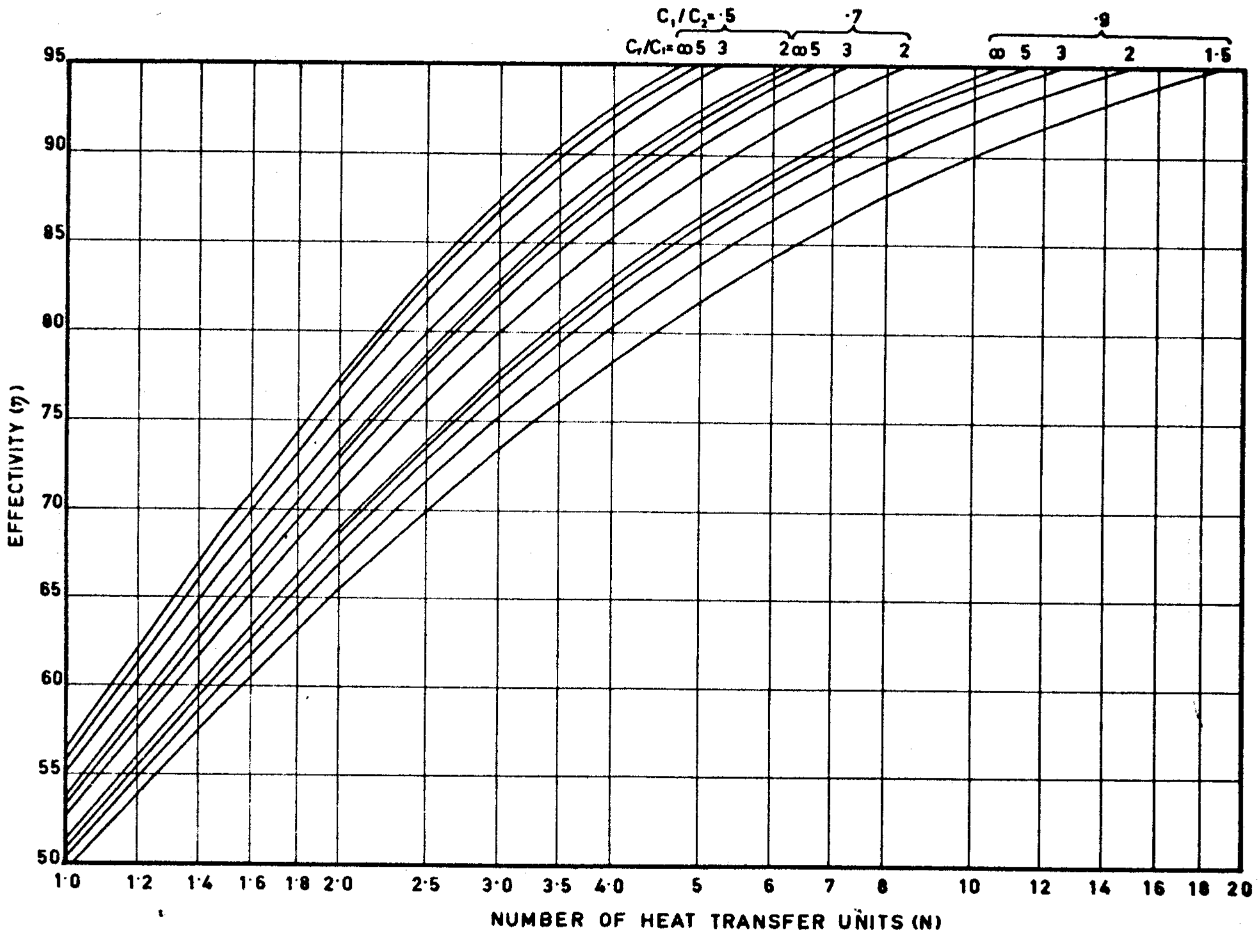
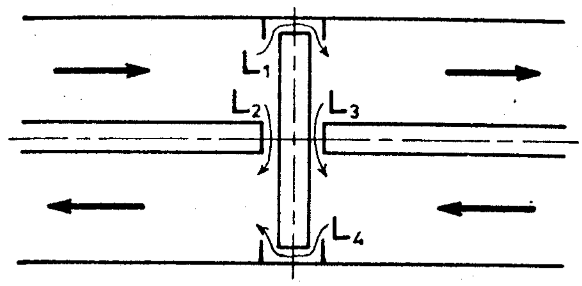
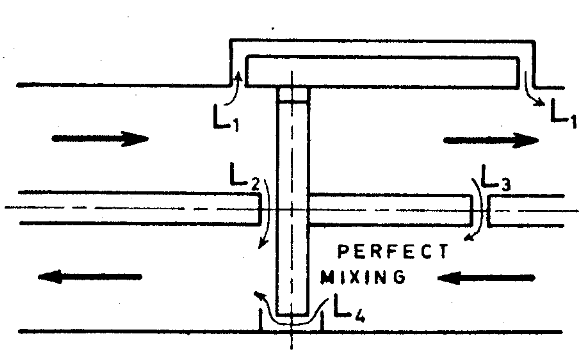


Fig. 3.—Heat Transfer Effectivity η for Flow Stream Capacity Rate Ratio $C_1/C_2 = 0.9$, 0.7 and 0.5, and $h_1A_1/h_2A_2 = 1$.



(a) ACTUAL LEAKAGES THROUGH SEALS



(b) ASSUMED LEAKAGES THROUGH SEALS

ig. 4.—Assumptions for Calculating Effect of Seal Leakages on Heat and Mass Transfer.

ınd polypropylene felts when used as a heat transfer matrix tend to pick ıp dust which has appreciable moisture adsorption but that parallel plates ıave negligible dust pick up. Dust picked up by the matrix may be con- idered as part of the matrix for heat and mass transfer purposes. An ıdequate treatment of coupled heat and mass transfer in regenerators with ıdsorption in the matrix has not yet appeared in the literature, however he authors and their colleagues are working on this problem.

If seal leakage and adsorption in the matrix are negligible the carry- ›ver ratio γ is equal to the mass transfer effectivity. The definition of nass transfer effectivity is similar to that of heat transfer effectivity which ıppears in the list of symbols. The carry-over ratio may be expressed 'or balanced symmetric regenerators as

$$\gamma = \frac{nL}{\phi u} \qquad \text{(6)}$$

6.—SEAL LEAKAGE

In the above discussion of heat and mass transfer in regenerators he effect of seal leakage has been neglected. The effect of seal leakage nay be taken into account by considering the leakage flows to mix perfectly with the flow streams at the inlet to the matrix before passing through, ınd at the outlet (Fig. 4). Since the leakage flows are small this is a ıatisfactory procedure for design.

Clearance seals have been found by the D.M.E. to be satisfactory and nexpensive if properly designed. The leakage flow Q_s through them may ɔe calculated from

$$Q_s = C_D A_s \left(\frac{2\Delta p_s}{\rho}\right)^{\frac{1}{2}} \qquad \text{(7)}$$

Harper (Ref. 23) gives the value 0.65 for C_D.

7.—HEAT STORAGE

The desirable properties of the heat storage material in parallel plate :egenerators are high specific heat, high density, low cost, fire resistance, ıatisfactory mechanical properties and a thermal conductivity that is not so ıigh the conduction in the flow direction is a problem or so low that con- luction into the matrix is a problem. The high strength plastic films ıre the best class of material at air conditioning temperatures and polyethy- ene terephthalate polyester film has been used by the D.M.E.

The heat storage properties of the matrix appear in the matrix capacity :ate ratio C_r/C_1. This may be expressed for balanced symmetric regenera- :ors as

$$C_r/C_1 = \frac{c_m Mn}{\rho c_p Q} \qquad \text{(8)}$$

$$= \frac{\rho_m c_m (1 - \varepsilon)\gamma}{\rho c_p \varepsilon} \qquad \text{(9)}$$

For parallel plates $\varepsilon = b/(a + b)$ therefore

$$C_r/C_1 = \frac{\rho_m c_m a\gamma}{\rho c_p b} \qquad \text{(10)}$$

8.—DESIGN PROCEDURE FOR PARALLEL PLATE REGENERATORS

A thermal design procedure will be described below for lamina counter flow, unbalanced or balanced, symmetric, polyester film, parall plate, rotary sensible heat regenerators. The values of $St, f, K, \rho, c_p, \rho_m$ c_m, C_D for Type A polyester film (Ref. 24) and air at 75°F., 50% R.H and 14.7 lb./sq. in. have been substituted in Eqs. (2), (4), (6), (7) and (10 to give Eqs. (11), (12), (13), (14) and (15). The units of a, b and L a inches, A_s square inches, Q_s cubic feet per minute, Δp and Δp_s inche water gauge at 60°F., n revolutions per minute and u feet per minute.

$$N = 3.92 \times 10^6 \, \Delta p/u^2 \qquad \text{(11)}$$

$$C_r/C_1 = 1.36 \times 10^3 \, a\gamma/b \qquad \text{(12)}$$

$$\Delta p = 1.77 \times 10^{-7} \, uL/b^2 + 4.20 \times 10^{-8} \, u^2 \qquad \text{(13)}$$

$$\gamma = \frac{nL}{12\phi u} \qquad \text{(14)}$$

$$Q_s = A_s (332 \times \Delta p_s)^{\frac{1}{2}} \qquad \text{(15)}$$

The design procedure for given η, γ, Δp_1, L, Q_1, Q_2, ϕ, a/b an negligible seal leakage consists of the following steps.

(1) From Eq. (12) determine C_r/C_1.
(2) From Figs. 2 and 3, depending on $C_1/C_2 = Q_1/Q_2$, determine for the required value of η.
(3) From Eq. (11) determine u_1.
(4) From Eq. (13) determine b.
(5) From Eq. (14) determine n.

9.—ASYMMETRIC AND BYPASSED REGENERATORS FO UNBALANCED FLOW

There are two methods of reducing the total pressure or fan powe required by a regenerator, below that required by a symmetric regenerato for the same flows, effectivity, regenerator size, rotor type and passag shape when the flows are unbalanced. Firstly, the regenerator may b made asymmetric by increasing the frontal area for the large flow and decreas ing the frontal area for the smaller flow. Secondly, the larger flow ma be divided into two parts which are remixed after one has been passe through the regenerator using a fan and the other has bypassed th regenerator.

The reduction in total pressure power which can be achieved depend among other things, on the relation between pressure drop (Δp) and passag velocity (u). The authors have studied the reductions for linear ($\Delta p \propto$ and square ($\Delta p \propto u^2$) relations and make the following three recommenda tions.

(1) If the larger flow is less than or equal to twice the smaller flo neither asymmetric nor bypassed regenerators should be used a the reductions are too small to justify the extra capital cost.
(2) If the larger flow is greater than twice the smaller flow, bypasse regenerators should be used as the reductions are higher an extra capital cost lower than for asymmetric regenerators.
(3) That part of the larger flow which passes through the bypasse regenerator should be four-thirds of the smaller flow if $\eta > 0.8$ and equal to the smaller flow if $\eta < 0.85$. This gives reductior sufficiently close to the optimum for practical purposes.

10.—MECHANICAL DESIGN PROBLEMS

A detailed consideration of the mechanical design problems is outsid the scope of this paper. However, two important points will be mentione here which impinge directly on the thermal design.

The frame and housing of the regenerator must be sufficiently rigi so that the seals are maintained in alignment.

The heat exchanger passages must be of uniform size or the perfor mance is reduced. To ensure this for parallel plate regenerators of th spirally wound type, the tension must be kept uniform during winding an the spacers must be sufficiently rigid.

11.—TEST RESULTS ON A PARALLEL PLATE REGENERATOR

The authors were present during tests made by Whitaker, Renehan and Robson, Consulting Engineers on the 78 in. dia. parallel plate regenerator in Fig. 1, and were given permission to quote some test results. The matrix had depth $L = 4$ in., film thickness $a = 0.003$ in. and passage spacing between plates $b = 0.036$ in. The predicted balanced flow performance using Eqs. (11), (12), (13), (14) and (15) and Fig. 2 was effectivity $\eta = 0.72$, pressure drop $\Delta p = 0.40$ in. H_2O, carry-over $\gamma = 0.015$ at a flow of 8,700 c.f.m. per side. The measured performance was effectivity $\eta = 0.69$, pressure drop $\Delta p = 0.48$ in. H_2O, carry-over $\gamma < 0.02$ at the same flow rate.

There were two suspected deficiencies in the test rig, leakage of air through the wooden ducts and non-uniform flow through the regenerator due to the presence of bends without turning vanes upstream in the ducts. The assumed value of ϕ may have been optimistic but this cannot be inferred from the results because of the deficiencies in the test rig. The second deficiency would be present in many air conditioning installations and it is suggested that turning vanes be used to reduce separation near the inlets to regenerators.

12.—ECONOMICS OF HEAT RECOVERY FROM EXHAUST AIR IN AIR CONDITIONING

In considering whether to install a regenerator for heat recovery in an air conditioning system it is appropriate to compare the installed cost of the regenerator with the present worth or value (Ref. 25) of the savings to be gained from its installation. The present value of savings consists of two components, the initial capital cost savings due to reduction of the peak capacities required of the heating and cooling plant, and the annual running cost savings multiplied by a present value factor (Ref. 25). It will vary depending on climatic factors, type and size of air conditioning plant and amount of fresh air required. It is calculated in Table II on a per unit exhaust airflow basis for 12 hr. and 24 hr. air conditioning with balanced symmetric regenerators at eight Australian locations.

The present values of savings in Table II lie between 43 and 77 cents/c.f.m. for 12 hr. air conditioning and 86 and 156 for 24 hr. These values are less than those at Canadian locations (Ref. 4) but are large enough to justify Australian installations since they are larger than the installed costs. The installed costs calculated for two American manufactured woven wire regenerators of 20,000 c.f.m. were 63 and 78 cents/c.f.m. and for an Au[s]ralian manufactured parallel plate regenerator of 13,000 c.f.m. 27 cents/c.f.[m.] In calculating these costs allowances for freight, installation, extra ducti[ng] and increased fan capacity have been added to the prices quoted by t[he] manufacturers.

In air conditioning practice the inflow of fresh air at the plant room [is] usually greater than the outflow or exhaust air. The effect of unbalanc[ed] flow, through a symmetric regenerator on the present values of savings [in] Table II is therefore of interest. Table III gives values of the chang[e] in the present value of savings for unbalanced flow. If $Q_1/Q_2 >$ [0] these changes lie between −14 and +1 cents/c.f.m. for 12 hr. air conditio[n]ing and −40 and +4 cents/c.f.m. for 24 hr. They are not large enou[gh] to alter the conclusion in the previous paragraph.

Table III shows that slight unbalanced flow gives an increase in t[he] present value of savings per unit exhaust airflow. Overall savings will [be] increased, however, if the system is designed so that as much as possible [of] the exhaust air is passed through the regenerator. Table III also sho[ws] that savings are higher for the linear relation in unbalanced flow due [to] lower pressure power.

The pressure drops and effectivity used for Table II and III are n[ot] necessarily the optimum for a given installation. As a rule it is bett[er]

TABLE III

Corrections to Present Values of Savings for Unbalanced Fl[ow] $(C_1/C_2 < 1)$

Period of air conditioning (hr.)		24	24	12	1
Cost of electricity (cents/kWh.)		2	3	2	
Linear Relation ($\Delta p \propto u$)	$C_1/C_2 = 0.75$	+ 3	+ 4	+ 1	+
Corrections (cents/c.f.m.)	$C_1/C_2 = 0.50$	−11	−16	− 4	−
Square Relation ($\Delta p \propto u^2$)	$C_1/C_2 = 0.75$	+ 1	+ 1	0	
Corrections (cents/c.f.m.)	$C_1/C_2 = 0.50$	−27	−40	− 9	−1

1. u_1 same as for balanced flow and $\eta = 0.80$.
2. $C_r/C_1 = 2$ for balanced and unbalanced flow.
3. Correction is due to change in running cost of regenerator.
4. Corrections add to present values of savings in Table II.

TABLE II

Calculation of Present Values of Heat Recovery Savings per Unit Exhaust Airflow

Quantity, Units and Equation		Numbers of Assumptions	Symbol for Quantity	Adelaide Airport	Alice Springs Airport	Canberra Airport	Hobart Airport	Kalgoorlie Airport	Melbourne	Perth	Sydney
24 hr. Climatic Data											
degree hours (°F.hr. × 10^3) calculated from (Ref. 26) with base temperature 75°F.	cooling	2, 3, 4, 5	l_1	6.5	44.7	4.4	0.9	20.4	4.9	10.7	3.1
	heating	2, 3, 4, 5	l_2	130.0	71.6	175.5	183.5	104.5	146.6	100.3	101.7
design Δt (°F.) exceeded between 10 and 20 hr. per year calculated from (Ref. 26).	cooling	2, 4, 5	l_3	24	30	18	12	30	24	26	18
	heating	2, 4, 5	l_4	40	42	50	42	42	40	34	32
24 hr. Capital Cost Savings											
cooling (cents/c.f.m.)	$l_5 = 1.43 \times l_3$	1, 6	l_5	34	43	26	17	43	34	37	26
heating (cents/c.f.m.)	$l_6 = 0.17 \times l_4$	1, 6, 7	l_6	7	—	9	7	—	7	6	5
24 hr. Running Cost Savings											
cost electricity (cents/kWh.)			l_7	2	3	2	2	3	2	2	2
cost heat (cents/kWh.)		7, 8, 9	l_8	0.40	0.60	0.40	0.40	0.60	0.40	0.40	0.4
cooling saving (cents/c.f.m.yr.) $l_9 = 0.0628 \times l_1 \times l_7$		9	l_9	0.8	8.4	0.6	0.1	3.8	0.6	1.3	0.4
heating saving (cents/c.f.m.yr.) $l_{10} = 0.251 \times l_2 \times l_8$			l_{10}	13.0	10.8	17.6	18.4	15.7	14.7	10.1	10.2
running cost of regenerator (cents/c.f.m.yr.) $l_{11} = 2.06 \times l_7$		1, 10	l_{11}	4.1	6.2	4.1	4.1	6.2	4.1	4.1	4.1
nett running cost saving (cents/c.f.m.yr.) $l_{12} = l_9 + l_{10} - l_{11}$			l_{12}	9.7	13.0	14.1	14.4	13.3	11.2	7.3	6.5
24 hr. present values of saving (cents/c.f.m.) $l_{13} = l_5 + l_6 + 8.51 \times l_{12}$		11	l_{13}	124	154	155	147	156	136	105	86
12 hr. present values of savings (cents/c.f.m.)		3, 4, 5	l_{14}	60	77	62	51	71	72	57	43

Assumptions :
1. Balanced flow regenerator with $\Delta p = 0.50$ in. H_2O and $\eta = 0.80$.
2. Room design condition 75°F., 50% R.H. at 14.7 lb./sq. in.
3. 24 hr. (12 hr.) air conditioning is for 8,760 (3,000) hours per year.
4. 12 hr. degree hours cooling (heating) = 0.40 (0.25) × 24 hr. degree hours cooling (heating).
5. 12 hr. design Δt = 24 hr. design Δt.
6. Capital cost saving cooling plant \$200/ton and boiler plant \$2 per 1000 B.Th.U./h[r.]
7. Reverse cycle plant in Alice Springs and Kalgoorlie with no capital cost savi[ng] heating.
8. Fuel oil \$35/ton with heating value 19,000 B.Th.U./lb.
9. C.O.P. cooling plant 4, C.O.P. reverse cycle 5 and boiler efficiency 0.70.
10. Running cost of regenerator calculated with combined fan motor efficiency 0.50.
11. Present value factor for 20 years with 10% interest compounded annually 8.51.

use lower pressure drops or higher effectivities or both when the present ue of savings is high and the installed cost of the regenerator low.

The eight Australian locations in Table II contain none in the humid pics. It is recommended that rotary total heat regenerators be used tead of rotary sensible heat regenerators in the humid tropics as possible ings are greater. Rotary total heat regenerators have been available Sweden and the United States of America for many years.

CONCLUSIONS

The parallel plate regenerator developed by the Division of Mechanical gineering, C.S.I.R.O. has significant advantages over the other types.

The use of rotary sensible heat regenerators or rotary total heat regenera-s for heat recovery from exhaust air in air conditioning systems is well tified economically for installations in Australia.

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