A HEAT EXCHANGER FOR ENERGY SAVING IN AN AIR-CONDITIONING PLANT

DONALD PESCOD

ABSTRACT

Savings in energy requirements of an air-conditioning plant may be obtained with heat exchangers. A plate heat exchanger can provide heat recovery in a ventilation system with complete separation of the air streams, or it can cool the incoming air without changing the moisture content by evaporative cooling of the plates from the exhaust side.

A novel design of a plate heat exchanger suitable for both uses is described, performance data are given and design studies are presented.

INTRODUCTION

A conventional air-conditioning plant is a large consumer of energy; the typical pattern in industrialized countries is for the energy demand by air-conditioning equipment to be a significant proportion of the total demand, and to be increasing.

When air is cooled by refrigeration, most of the energy is used in re-cycling the refrigerant, either by compressing the vapor for condensation and re-use, or by absorption processes involving large input and rejection of heat.

When buildings are heated, between 5 and 25% of the energy may be lost in the form of heat in the air exhausted for ventilation purposes.

Much of the wasted energy could be conserved by appropriate use of heat exchangers. Dunkle et al. (1), (2), discussed the theory and design of rotary regenerators for air conditioning, and Morse (3) and Dunkle and Ellul (4) discussed the design of particulate bed regenerators for heat recovery and air cooling in buildings. Rotary regenerators are being produced commercially in Australia by Rotary Heat Exchangers Pty. Ltd., Melbourne, as reported by Robson (5). Particulate bed regenerators have been manufactured commercially by S.I.D. Pty. Ltd., Adelaide, and Read et al. (6) have reported on the use of such systems in South Australian schools.

The author (7) developed a cross-flow heat exchanger for an air-conditioning application, using copper wires for heat transfer between the two air streams. Tests showed a performance superior to the predicted performance, and this was attributed to a very high heat transfer rate between the air streams, derived from the turbulent wake from the wires. This suggested a less expensive type of heat exchanger in which the copper wires would be eliminated. It would be necessary to have the dividing plates closer together and to provide turbulence promoters in the air streams, but an inexpensive material such as thin rigid polyvinyl chloride could be used.

The author (8) confirmed experimentally the feasibility of this design and also showed that it could be used as an effective air cooler by wetting the surfaces of the plates in contact with the exhaust air stream.

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Further experimental work was carried out by the author (9), (10) to establish design parameters for dry heat exchangers, and by a colleague, Chan (11), to establish design parameters for heat exchangers with wetted plates.

Experimental units have been installed in various localities in Australia to obtain information on reliability, and preparations for commercial manufacture are well advanced.

The main advantages of plate type-heat exchangers, often referred to as "recuperators", over regenerators in which the heat is stored temporarily while the storage medium is subjected alternatively to the passage of each air stream, are that recuperators have no moving parts and associated sealing problems, air streams may be completely isolated, and evaporative cooling can be obtained directly within the heat exchanger, but without adding water vapor to the incoming air. In this form, the cooling unit may be regarded as an evaporator in which the refrigerant is very cheap, absolutely non-polluting and safe to discard with the exhaust air, thus saving the expenditure of energy in compression. The quality of the water need not be high, but adequate bleed-off is essential. Equivalent cooling systems can be made using regenerators, but a separate evaporative cooler must be provided. The ducting with a plate heat exchanger is simpler than with a switched bed type of regenerator, but more complicated than the ducting required for a rotary regenerator. Wooldridge et al. (12) reported on all three cooling systems.

DESIGN

Theory

Dry Heat Exchangers. For economy of material and space, a high heat transfer coefficient is required. With plain flat plates, the air flow would be laminar, and a plate spacing of about 1 mm would be required to obtain a reasonable heat transfer coefficient. A passage length of about 100 mm would allow a reasonable effectiveness.

This arrangement is very suitable for rotary regenerators, but with cross-flow plate heat exchangers, the passage breadth would be limited to the length of the adjacent passage; that is, about 100 mm, a size quite unsuitable for the volume flows required. A more desirable breadth of passage has been found to be between 300 and 500 mm, with a plate spacing of about 3 mm.

Patterns of projections or corrugations on heat exchanger plates can create turbulence in the air stream and greatly increase the heat transfer coefficient, at the expense of increased pressure drop. Fig. 1 shows examples of vacuum formed plastic plates used in experimental heat exchangers. Fig. 2 shows the variation in heat transfer coefficient and pressure gradient with plate spacing, for plain plates, and for plates with small turbulence promoters, at a constant air velocity of 2 m/s at entry. In Fig. 2, the pressure gradient appears to be independent of the plate spacing, as was assumed in earlier work (10). However, recent tests have shown that the resistance of the projections is approximately proportional to the diameter. For practical reasons, the projections were tapered, with the result that the mean diameter was reduced as the plate spacing was reduced, and the effect of variation of passage thickness was obscured. As the resistance of the projections varies with the diameter, it is desirable to use the smallest diameter which will provide sufficient turbulence.

A series of tests was conducted by the author with the spacing of the plates and the projections varied to obtain empirical data for design optimization. These data are presented with dimensionless parameters in Fig. 3 and 4, as in (10). The key to the symbols used in these and subsequent graphs is given in Table 1. The Stanton Numbers used in Fig. 3 were based on the projected area of the plates, that is, the same area as used for plain plates, because the extended surface area was too difficult to determine precisely. If the complete surface area of the corrugated plates had been used for the calculations, the points on the graph for the corrugated plates would have been more closely aligned with the other points. Fig. 4 shows a marked difference in the friction factor for the corrugated plates between air flow along the corrugations and transverse air flow.

The effectiveness of a heat exchanger may be expressed as

$$\varepsilon = \frac{(t_{hi} - t_{ho}) + (t_{co} - t_{ci})}{2 (t_{hi} - t_{ci})}$$ (1)

The general equation for the heat transfer coefficient between a fluid in motion and an
adjacent surface may be written

\[ h = \rho V^b \]  

(2)

for constant fluid properties. In Fig. 5, the heat transfer coefficients derived from the test results are plotted against the values of approach velocity, and curves of the type given in Eq 2 have been fitted to the test results.

Consider a dry counterflow heat exchanger with balanced air streams and negligible differences of air density and other properties, and negligible temperature difference across the plates.

The mass flow per unit of frontal area of the heat exchanger is \(0.5 \rho V_a\). The factor 0.5 is used because only half of the passages are open to the air stream being considered.

The heat flow

\[ H = (t_{hi} - t_{ho}) \text{ or } (t_{co} - t_{ci}) \times 1000 \rho \times (0.5 \rho V_a) \text{ W} \]  

(3)

Hence

\[ (t_{hi} - t_{ho}) = (t_{co} - t_{ci}) = H/(500 \text{ Cp} \rho V_a) \text{ °C} \]  

(4)

There will be a constant temperature difference between the two air streams:

\[ (t_{hi} - t_{ci}) = (t_{ho} - t_{ci}) = 2H/(hA) \text{ °C} \]  

(5)

From Eqs 4 and 5,

\[ (t_{hi} - t_{ci}) = (t_{hi} - t_{co}) + (t_{co} - t_{ci}) = 2H/(hA) + H/(500 \text{ Cp} \rho V_a) \]

Substituting in Eq 1 and transposing,

\[ \epsilon_d = 1/[1 + (1000 \text{ Cp} \rho V_a)/(hA)] \]  

(6)

The plate surface area exposed to one air stream in unit frontal area of the heat exchanger = \((L/s)_a\) m².

Substituting \((L/s)_a\) for \(A\) and \((F V_a)^b\) for \(h\) in Eq 6,

\[ \epsilon_d = 1/[1 + (1000 \text{ Cp} \rho V_a)/(F V_a^b L/s_a)] \]  

\[ = 1/[1 + (1000 \text{ Cp} \rho V_a (1-b))/(FL)] \]  

(7)

With a significant temperature difference across the heat exchanger plates, a correction factor \(J\) must be applied to Eq 7:

\[ \epsilon_d = 1/[1 + (1000 J s^\prime \text{ Cp} \rho V_a (1-b))/(FL)] \]  

(8)

A further correction factor \(Q\) must be applied for cross flow heat exchangers:

\[ \epsilon_d = Q/[1 + (1000 J s^\prime \text{ Cp} \rho V_a (1-b))/(FL)] \]  

(9)

Eq 9 may be re-written:

\[ \epsilon_d = Q/[1 + s^\prime \text{ M V}^c/L] \]  

(10)

where

\[ M = 1000 J \text{ Cp} /F \]

For unbalanced air flows, further correction would be necessary.

With plastic plates, a typical value for \(J\) is 1.07, calculated from the known thermal resistance of the plates.

From information given by Kays and London (13), values of \(Q\) may be calculated. With most designs of heat exchangers used in air-conditioning applications, \(Q\) varies from 0.915 to 0.902, corresponding to values of \(AU/(W \text{ Cp})\) of 2 to 5. A close estimate of the effectiveness of a dry
plastic plate heat exchanger may be obtained by using Eq 10, and values for Q, M and e given in Table 2; derived from the experimental results as given in Fig. 5.

The resistance to air flow through the passages may be divided into two components - shear of the boundary layer at the walls, and velocity pressure loss at projections. This is evident in Fig. 2. With these types of plates, change of air velocity has a different effect on the coefficient of friction than change of place spacing, so results converted to Reynolds Numbers tend to be scattered, as seen in Fig. 4. More consistent results are obtained with pressure gradient plotted against velocity, as in Fig. 6.

Using the "two-component" approach to pressure gradient, it may be expressed mathematically as:

\[ \frac{P}{L} = 2 \rho V_m^2 \frac{f}{D} + 0.5 \rho V_m^2 \frac{d}{D} \frac{d}{L} \]  \hspace{1cm} (11)

Values of \( f \) may be obtained from the data presented by Kays and London (13). For the physical sizes and range of air flows used in the experiments,

\[ f \cdot Re = 21.5, \text{ approximately} \]

and

\[ D = 2 \text{s, approximately} \]

The drag coefficient may be taken as approximately 1.0, and the spacing of the projections is such that interference effects may be ignored.

Using a value for \( \rho \) of 1.2 Kg/m\(^3\) and a value for \( \mu \) of 18.1 \times 10^{-6} \text{ Pa s}, Eq 11 may be written

\[ \frac{P}{L} = 0.000195 \frac{V_a}{s^2} + 0.6 \frac{V_m^2}{s} \frac{q}{R/L} \]  \hspace{1cm} (12)

\[ V_m = V_a/(1-q) \]  \hspace{1cm} (13)

In Fig. 6, theoretical curves of pressure gradient vs approach velocity, using Eq 12, are drawn for three different heat exchangers to show the agreement with the experimental results. The effect of change of number of rows of projections is evident; the effect of variation of passage thickness is obscured by the effect of change of mean diameter of projections.

The corrugated plates cannot be analysed in the same way, but the equation

\[ \frac{P}{L} = K V_a^{1.75} \]  \hspace{1cm} (14)

agrees well with the experimental results, with \( K = 50 \) for flow parallel to corrugations and \( K = 220 \) for flow normal to corrugations.

Heat Exchangers with Wet Surfaces. The effect of wetting the surfaces of the plates in contact with the exhaust air stream on the temperature and humidity of the air is shown in Fig. 7. The psychrometric chart compares the recorded performance of a plate heat exchanger cooler with that of a simple, direct evaporative cooler for a typical summer day. The improvement in room conditions is very evident.

The process of heat transfer with evaporation is complex, and a theoretical study of internal conditions within a cross-flow heat exchanger would require the use of a computer. However, the overall effectiveness can be related to the performance of the heat exchanger when dry.

The overall effectiveness of a heat exchanger when operating as an indirect evaporative cooler may be expressed as

\[ \varepsilon_w = \frac{t_{hi} - t_{ho}}{t_{hi} - t_{ciw}} \]  \hspace{1cm} (15)

where \( t_{ciw} \) is the wet-bulb temperature of the air entering the wet exhaust passages from the room, and corresponds to the theoretical minimum possible temperature of room inlet air (\( t_{ho} \)).

The value of \( \varepsilon_w \) has been found to remain fairly constant over the normal range of working conditions.

There is a similarity in behavior between sensible heat transfer and latent heat transfer.
The driving force in the former case is the temperature difference, and in the latter case it is the vapor pressure difference. It may be seen from steam tables that the relationship between these quantities is not linear, but the deviation from linearity is not significant over the range of conditions experienced with this type of cooler. Consequently, Eq 10 may be used also to calculate the effectiveness when the heat exchanger is used as a cooler. In practice, care must be taken to wet the surfaces of the plates as completely as possible, and it has been customary when using low pressure sprays to spray water on to the top of the plates at a rate 20 to 30 times that required for evaporation. The movement of water down the plates causes sensible heat transfer which modifies the performance slightly. Chan (11), using the heat exchangers previously tested in the dry condition, measured the performance of each type while water was sprayed into the "exhaust" passages. A fine atomizing spray was used to minimize effects of sensible heat transfer via the water. From these results, new values of $M$ in Eq 10 could be determined by fitting curves to the experimental results. A scatter of the points was probably due to variation in the area of wetted surface. The same values of $Q$ and $e$ have been used as in Table 2, but the values of $M$ are quite different. All these values are given in Table 3.

It may be seen that the wet effectiveness of a plate heat exchanger is substantially better than the dry effectiveness; in all cases the value of $M$ in the wet condition is less than half the value for the dry condition. The difference in effectiveness is illustrated in Fig. 8. The curves for effectiveness were calculated using Eq 10; the curve for pressure drop was measured by the author on a sample heat exchanger and includes entry and exit losses.

The above values apply to heat exchangers with balanced air flows. The effect of reducing the flow on the wet (exhaust) side was investigated by the author at an early stage of the research project, for two different designs of heat exchanger. The results, given in Fig. 9, had not been published previously, and although they are less accurate than later measurements, the trend is clearly shown. In each case the exhaust flow could be reduced by 20 to 30% before there was a significant reduction in effectiveness. In rooms having a high latent heat load, such as washrooms and kitchens, the exhaust air from the room need not be used and the cooled air from the heat exchanger may be divided, with about half entering the room, and the remainder returning through the wet passages of the heat exchanger. This system is shown diagrammatically in Fig. 10, and is compared with the normal system of cooling. Although the effectiveness would be about 12% lower than with the normal system, the wet-bulb temperature of the air entering the wet passages would be typically 1.7 deg C lower than normally, with the result that the air entering the room would be only about 0.5 deg higher than with the normal system. More than twice the air flow must pass through the primary passages of the heat exchanger, but only one fan is required.

The resistance to air flow through wet passages was measured by Chan (11) with various designs of plates and minimal water flow rate. With a downwards air flow it was found to be approximately 15% higher than through the same passages when dry. The author had tests carried out on a different design of heat exchanger and obtained an increase in resistance of 12% with wet passages and a downwards air flow, and an increase of 40% with wet passages and an upwards air flow. The water flow rate was not accurately controlled but it did not appear to have a significant effect on the resistance.

Design Optimization. Because there are so many design parameters, several designs may have to be prepared for comparison. Heat exchangers with the lower capital costs usually have the higher air resistance, and hence the higher energy requirements.

Examples of design procedures are given in Appendix A. The heat exchanger may be designed firstly on the basis of effectiveness, using different center spacings of projections and an estimated suitable air velocity. Values of $s'/L$ may be calculated, and $s'$ and $L$ may be chosen. The pressure drop may then be calculated, and the velocity may be revised if necessary; change of velocity affects the pressure drop much more than it does the effectiveness. Final design may be selected on the basis of compactness, or economy of plates or material.

Energy Savings. The author (14) made estimates of energy savings which could be obtained using evaporative cooling instead of refrigeration. Australia was divided into eighteen climatic zones, according to Ashton (15), and annual savings ranging from 65 to 90% were shown to be possible. However, as evaporative cooling is limited by the partial pressure of water vapor in the air, the inside conditions would be above the comfort region when heat-exchanging evaporative coolers are used with an outside wet-bulb temperature above about 25°C.

The amount of heat which may be recovered with a sensible heat exchanger during cold weather depends upon many factors and should be calculated for each particular installation proposed.
For example, consider a building having a fresh air flow rate of 10 m³/s (12 kg/s) and an inside temperature of 21°C. If the outside temperature is 2°C, the heat to be added to the fresh air is

\[ H = W C(t_r - t_i) \]

\[ = 12 \times 1.01 (21 - 2) = 230 \text{ kW}. \]

Using a heat exchanger with an effectiveness of 0.74, the saving in heat would be

\[ 230 \times 0.74 = 170 \text{ kW}. \]

The capital cost of heating plant would also be reduced. Dunkle and Maclaine-cross (1) have given some estimates of savings in capital costs and running costs using sensible heat exchangers.

**Practice**

The manufacture of the heat exchangers requires special techniques. Plates have been formed satisfactorily by drape moulding and by a mechanical process. The plates may be jointed by electronic welding, heat sealing, gluing, or they may be adequately sealed by mechanical pressure, assisted by air pressure using specially formed edges. The basic design and the manufacturing methods are mostly covered by patents. Manufacture of heat exchangers and coolers under licence is being arranged.

Most plastics do not wet readily when first used, so the initial cooling performance may be poor. This may be improved by coating the surfaces with suitable wetting agents, or by adding some to the water (11). However, after 50 to 100 hr of operation, a thin film builds up on the plates and enables the surfaces to remain wet. Care must be taken to ensure that the thickness of film does not become excessive; a high rate of circulation of water, or regular flushing, can overcome this problem.

It is desirable to have a constant bleed-off of water to prevent it becoming saturated with dissolved salts; the rate of bleed-off would depend on the quality of the water supply. Water with 3000 parts per million of dissolved salts has been used satisfactorily in an actual installation for several years.

Maintenance is simple, and consists of inspection of the water spray jets and cleaning if necessary, inspection and cleaning of air filters (if any), and occasional cleaning of the sump. With a rate of circulation of water about 30 times the evaporation rate, and adequate bleed-off to prevent the water becoming saturated with dissolved salts, no cleaning of the wet passages of the heat exchanger has been necessary.

Cooling units have also been used for heating; they provide ventilation with good heat recovery, and heating by means of a finned hot water coil or an electric heater at the room inlet.

**APPLICATIONS**

Plate heat exchangers may be used for energy saving by heat recovery in the forced ventilation system used in any building, but they are especially valuable where a high ventilation rate is required, as in schools, hospitals, public buildings, shopping complexes, and some factories.

Heat exchangers used for cooling may be installed in individual rooms, or in homes, office blocks, hotels, schools, public buildings, commercial premises and in industrial situations for either complete space cooling or for spot cooling of personnel. They are applicable to all areas except where the ambient wet-bulb temperature frequently exceeds 25°C.

Room coolers can be mounted through a wall or a window; units can be made mobile for spot cooling, and larger coolers can be mounted on a roof, beside a building or in a plant room. In larger applications, the most economical installation would usually consist of several medium sized units (2 to 5 m³/s), distributed around the building to minimize the ducting requirements.

**CONCLUSIONS**

Heat exchangers for air-conditioning applications may be made from thin rigid plastic sheets,
using novel designs.

They can offer savings in energy requirements when used for heat recovery in ventilation systems, or when used for cooling by fitting with water sprays to wet the exhaust passages. They are satisfactory for cooling in temperate and hot dry areas, but not completely satisfactory when the ambient wet-bulb temperature exceeds about 25°C.

Sufficient design data and operational experience is now available to design and manufacture plastic heat exchangers for a variety of applications.

APPENDIX A

Examples of Design Procedure

Heat Recovery Unit

Specification:

<table>
<thead>
<tr>
<th>Air flow rate</th>
<th>3 m³/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effectiveness</td>
<td>0.75</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>200 Pa</td>
</tr>
</tbody>
</table>

Try center spacing of projections of 9.5 mm. Fig. 6 indicates an approach velocity between 3 and 4 m/s, for plates of length 0.8 m to 0.4 m.

Try \( V_a = 3.5 \), with a cross-flow arrangement.

From Eq 10 and Table 1,

\[
\varepsilon_d = 0.91/[1 + 24 \times s' \times v^{0.35}/L]
\]

Thus

\[
s'/L = (0.91/\varepsilon_d - 1)/(24 \times v^{0.35})
\]

\[
= 0.00573
\]

Try \( L = 0.6 \) m

\[
s' = 0.00573 \times 0.6 = 0.0034 \) m.

Let plate thickness = 0.2 mm

Then \( s = 0.0034 - 0.0002 = 0.0032 \) m.

Let mean diameter of projects = 1.8 mm

Then

\[
q = 1.8/(9.4 \times \sqrt{2}) = 0.134
\]

and

\[
V_m = V_a/(1-q) = 4.04 \) m/s
\]

\[
R = 0.6 \times \sqrt{2}/0.0095 = 90
\]

From Eq 12:

\[
P = 0.000195 L V_m/s^2 + 0.6 V_m^2 q R
\]

Pressure drop = 46 + 118 = 164 Pa

The usual design of heat exchanger tends to restrict the flow at entry and exit, so that the pressure drop is higher than the theoretical value. This calculated value would allow up to 36 Pa for entry and exit losses.

Face area: Only half the face area is taken up by passages in one direction.

Hence face area = 2 x volume flow/approach velocity

\[
= 2 \times 3/3.5 = 1.714 \) m²
\]

Length of heat exchanger = 1.714/0.6 = 1.86 m

Volume of heat exchanger = 1.714 x 0.6 = 1.03 m³

Similar calculations may be made with different center spacing of projections.
Summary of alternative designs

<table>
<thead>
<tr>
<th></th>
<th>13.4</th>
<th>9.5</th>
<th>6.7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Projections; center spacing, mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Plates; length and width, mm</td>
<td>800</td>
<td>600</td>
<td>500</td>
</tr>
<tr>
<td>Plate pitch, mm</td>
<td>3.6</td>
<td>3.4</td>
<td>3.3</td>
</tr>
<tr>
<td>Heat exchanger; breadth, mm</td>
<td>1850</td>
<td>2860</td>
<td>4530</td>
</tr>
<tr>
<td></td>
<td>1.46</td>
<td>1.71</td>
<td>2.26</td>
</tr>
<tr>
<td></td>
<td>1.17</td>
<td>1.03</td>
<td>1.13</td>
</tr>
<tr>
<td>Number of plates</td>
<td>510</td>
<td>840</td>
<td>1370</td>
</tr>
</tbody>
</table>

Conclusion: Most suitable spacing of projections: 13.4 mm as it is the most convenient shape and uses the least number of plates.

Space cooler

Specification:

- Air flow rate: 1 m³/s
- Effectiveness: 0.85
- Pressure drop: 100 Pa

Unit to be as compact as possible.

This design is difficult to make compact as the high effectiveness and low pressure drop as specified both require a low approach velocity, and hence, a large face area. Close plate spacing is an advantage, but not necessarily close projection spacing. Let \( s = 0.002 \) and \( s' = 0.0022 \) in all cases. Try center spacing of projections of 13.4 mm, with a cross-flow arrangement.

As the projections are short, they may also be small in diameter, say 1.3 mm.

Then \( q = 1.3/(13.4 \times \sqrt{2}) = 0.069 \)

Try \( V_a = 2.4; \quad V_m = 2.58 \)

From Eq 10 and Table 2,

\[
\epsilon_w = 0.91/[1 + 12 s' V_{0.4}/L]
\]

Thus \( L = 12 s' V_{0.4}/(0.91/\epsilon_w - 1) = 0.53 \) m

\( R = 0.53 \times \sqrt{2}/0.0134 = 56 \)

From Eq 12: \( F = 0.000195 L V_m/s^2 + 0.6 V_m^2 q R \)

Pressure drop = 67 + 16 = 83 Pa

Note: Extra restrictions at entry and exit would probably increase the pressure drop to about 100 Pa on the dry side. The resistance on the wet side could be 12 to 40% higher, but it is usual to reduce the flow by 10 to 20%. This does not affect the effectiveness significantly, but reduces the pressure drop through the heat exchanger, and tends to prevent infiltration of outside air into the room.

Similar calculations may be made with different center spacing of projections.

Summary of alternative designs

<table>
<thead>
<tr>
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<th>13.4</th>
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<tbody>
<tr>
<td>Projections; center spacing, mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Plates; length and width, mm</td>
<td>530</td>
<td>460</td>
<td>400</td>
</tr>
<tr>
<td>Plate pitch, mm</td>
<td>2.2</td>
<td>2.2</td>
<td>2.2</td>
</tr>
<tr>
<td>Heat exchanger; breadth, mm</td>
<td>1570</td>
<td>1890</td>
<td>2500</td>
</tr>
<tr>
<td></td>
<td>0.83</td>
<td>0.87</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>0.44</td>
<td>0.40</td>
<td>0.40</td>
</tr>
<tr>
<td>Number of plates</td>
<td>715</td>
<td>860</td>
<td>1140</td>
</tr>
</tbody>
</table>

Conclusion: Most suitable spacing of projections: either 9.5 mm, because of the smaller volume and economy of material, or 13.4 mm, because of the smaller breadth and fewer plates.
Cooling capacity

Specification:
- Air flow rate: 1 m³/s
- Effectiveness: 0.85
- Room condition: Dry-bulb temp.: 27°C
  Wet-bulb temp.: 19°C
- Cooler inlet condition, from outside: Dry-bulb temp.: 35°C

Eq 15: \[ \varepsilon_w = \frac{t_{hi} - t_{ho}}{t_{hi} - t_{tw}} \]

Thus \[ t_{ho} = t_{hi} - \varepsilon_w (t_{hi} - t_{tw}) \]

\[ = 35 - 0.85 (35 - 19) = 21.4°C \]

Temperature rise in room = \( t_{ro} - t_{ri} = 27 - 21.4 = 5.6°C \).

Density of air at inlet = 1.11 kg/m³

Hence mass flow = 1.11 kg/s.

Specific heat of air = 1.01 kJ/(kg °C) at room condition.

Sensible Cooling capacity = \( \rho c (t_{ro} - t_{ri}) = 1.11 \times 1.01 \times 5.6 \)

\[ = 6.28 \text{ kW} \]

APPENDIX B

Accuracy of Measurements

The estimated accuracy of measurements are as follows:

Temperature: ± 0.2 °C

Electrical: ± 1%

Air flow: (a) Calculated from heat input and temperature rise, with corrections for losses. Probable accuracy: ± 4%

(b) Orifice meter. Probable accuracy ± 3%

(c) Vane anemometer. (Used for relative flow rate only, as in Fig. 9.) Probable accuracy ± 5%

Air pressure differences: ± 2Pa.

NOMENCLATURE

A Surface area of heat exchanger plates exposed to one air stream m²
B Breadth of heat exchanger m
b A constant relating to heat transfer, for each type of plate
C Specific heat kJ/(kg °C)
d Hydraulic diameter of passage m
e A constant, = (1-b)
F A constant relating to heat transfer
f Mean friction factor, = PD/(2dV²)
G Total projected area of all projections per unit area of all passages, = qR m²
H Heat flow W
h Heat transfer coefficient W/(m² °C)
J A factor allowing for thermal resistance of heat exchanger plates
K A constant relating to pressure drop
k Thermal conductivity W/(m °C)
L Length of flow path m
M A constant relating to heat exchanger design and function
Pressure drop
Prandtl Number, \( = \mu \sigma /k \)
A factor relating cross-flow to counter-flow performance
Proportional blockage per row of projections
Number of rows of projections
Reynolds Number, \( = \rho VD/\mu \)
Stanton Number, \( = h/\left( \rho V C_p \right) \)
Spacing of plates, = passage thickness
Plate pitch, = passage thickness + plate thickness
Temperature
Total thermal conductance
Air velocity
Mass flow rate

Subscripts
a approach
b cool side
w internal
o outlet
p constant pressure
r room
w wet

Greek letters
\( \varepsilon \) effectiveness
\( \mu \) viscosity
\( \rho \) density

REFERENCES


**TABLE 1**

Key to symbols in Figs. 3 to 6

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>SPACING - mm</th>
<th>L = 0.2 m</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>PLATES</td>
<td>PROJECTIONS</td>
</tr>
<tr>
<td>x</td>
<td>3.40</td>
<td>6.7</td>
</tr>
<tr>
<td>O</td>
<td>2.82</td>
<td>6.7</td>
</tr>
<tr>
<td>*</td>
<td>2.16</td>
<td>6.7</td>
</tr>
<tr>
<td>□</td>
<td>3.56</td>
<td>9.5</td>
</tr>
<tr>
<td>Δ</td>
<td>3.49</td>
<td>13.4</td>
</tr>
<tr>
<td>✶</td>
<td>3.56</td>
<td>[Corrugated - Fig. 1b]</td>
</tr>
</tbody>
</table>

**TABLE 2**

Values of Q, M and e in Eq 10, for estimation of dry effectiveness of a plastic cross-flow recuperator

<table>
<thead>
<tr>
<th>Turbulence Promotor</th>
<th>Q</th>
<th>M</th>
<th>e</th>
</tr>
</thead>
<tbody>
<tr>
<td>Projections; 6.7 mm center spacing</td>
<td>0.91</td>
<td>23</td>
<td>0.35</td>
</tr>
<tr>
<td>Projections; 9.5 mm center spacing</td>
<td>0.91</td>
<td>24</td>
<td>0.35</td>
</tr>
<tr>
<td>Projections; 13.4 mm center spacing</td>
<td>0.91</td>
<td>27</td>
<td>0.4</td>
</tr>
<tr>
<td>Corrugations as in Fig. 1(b)</td>
<td>0.91</td>
<td>15</td>
<td>0.1</td>
</tr>
</tbody>
</table>

**TABLE 3**

Values of Q, M and e in Eq 10, for estimation of wet effectiveness of a plastic cross-flow recuperator

<table>
<thead>
<tr>
<th>Turbulence Promotor</th>
<th>Q</th>
<th>M</th>
<th>e</th>
</tr>
</thead>
<tbody>
<tr>
<td>Projections; 6.7 mm center spacing</td>
<td>0.91</td>
<td>10</td>
<td>0.35</td>
</tr>
<tr>
<td>Projections; 9.5 mm center spacing</td>
<td>0.91</td>
<td>11</td>
<td>0.35</td>
</tr>
<tr>
<td>Projections; 13.4 mm center spacing</td>
<td>0.91</td>
<td>12</td>
<td>0.4</td>
</tr>
<tr>
<td>Corrugations as in Fig. 1(b)</td>
<td>0.91</td>
<td>6.3</td>
<td>0.1</td>
</tr>
</tbody>
</table>

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Fig. 1 Plastic heat exchanger plates

Fig. 2 Heat transfer and pressure gradient in typical plate heat exchangers

Fig. 3 Stanton No. $x \left( \text{Prandtl No.} \right)^{2/3}$ vs Reynolds No.

Fig. 4 Friction factor vs Reynolds No.
Fig. 5 Heat transfer coefficient vs approach velocity

Watch approach vel.

Passage (%A)

Fig. 6 Pressure gradient vs approach velocity

Fig. 7 Psychrometric chart showing performance of direct and indirect evaporative coolers
Fig. 6 Effectiveness and pressure drop of typical heat exchanger

Fig. 9 Wet effectiveness of heat exchangers with reduced exhaust flow

Fig. 10 Alternative cooling systems