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**Experimental Testing and Analytical
Analysis of a Plastic Panel Heat
Exchanger for Greenhouse Heating**

M. Olszewski
J. F. Thomas

**OPERATED BY
UNION CARBIDE CORPORATION
FOR THE UNITED STATES
DEPARTMENT OF ENERGY**

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PANEL HEAT EXCHANGER FOR GREENHOUSE HEATING

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FOREWORD

It has been recognized that, because of power cycle operating conditions, nuclear power plants reject more waste heat per unit of electricity produced than comparable fossil stations. Because the waste heat problem is more severe for nuclear power stations, research in the area of beneficial uses of waste heat has been sponsored for a number of years by the Advanced Systems and Materials Production Division of the Department of Energy (DOE) [formerly Nuclear Research and Applications Division of the Energy Research and Development Administration (ERDA)].

These studies have considered various uses for power plant reject heat and have centered on agricultural and aquacultural applications. Some of the agricultural work has been carried out in a cooperative program with the Tennessee Valley Authority (TVA). This work is continuing with a 0.2-ha (0.5-acre) demonstration greenhouse constructed and operated by TVA at the Browns Ferry Nuclear Station. This greenhouse is based on a DOE-supplied design, and a combined DOE-TVA report concerning this project will be issued when sufficient operating experience has been obtained.

During the course of the beneficial uses investigations, it has become evident that the cost to transfer heat from the condenser cooling water to the user's system was the major cost to utilize waste heat. One potential method of reducing this cost is to substitute cheaper materials (i.e. plastics) for the metals typically used in the heat exchangers. This study was initiated to make a preliminary assessment of a plastic panel heat exchanger for waste heat applications.

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ABSTRACT

The performance of a plastic panel-type heat exchanger, suitable for greenhouse heating using low-grade (25 to 60°C water) power plant reject heat, was investigated theoretically and experimentally. The theoretical analysis showed that a plastic panel heat exchanger would have an overall heat transfer coefficient, U_o , of about 18 $\text{W/m}^2\text{-}^\circ\text{C}$ compared to about 12 $\text{W/m}^2\text{-}^\circ\text{C}$ for a fin-tube heat exchanger, under typical greenhouse conditions. Furthermore the plastic heat exchanger would require less fan power due to reduced air pressure losses. The experimental data revealed a similar functional relationship for U_o and air flow when compared with the theoretical calculations, however the experimental values of U_o were consistently larger by 20 to 30%. It was concluded that a properly designed plastic heat exchanger can compete with metal fin tube type exchangers on a performance basis, but the plastic heat exchangers are 3 to 4 times larger by volume. However, because of the lower cost of plastic, a substantial cost reduction is expected. It appears that further study, examining heat exchanger lifetime, performance and costs, is warranted.

1. INTRODUCTION

1.1 Background

Because of temperature considerations, beneficial uses of power plant waste heat have focused primarily on greenhouse and aquaculture applications. In both of these uses, heat contained in the condenser cooling water is transferred to a second fluid which is then used for the given application. In the case of greenhouses, the heat is transferred to the greenhouse air to maintain the plants within their thermal comfort zone. In most aquaculture applications contaminants in the condenser cooling water (e.g., chlorine and biocides) preclude its direct use for aquaculture. Therefore, the heat is transferred to a secondary water stream that is used as the aquaculture system water supply.

Studies^{1,2} have shown that the costs associated with the heat exchanger system can represent a significant portion of the cost to use waste heat. This is due, in part, to the relatively large heat transfer surface areas required because of relatively low ΔT 's. Another consideration adding to the cost is that some aquaculture concepts use species that accumulate heavy metals. In these applications costly inert heat exchanger metals (e.g., titanium) must be used.³

The use of low-cost plastic materials in innovative heat exchanger designs for such applications offers the potential for significant cost reductions in the heat exchanger system. Since plastic is basically an inert material it could replace expensive inert metals used in aquaculture applications. A significant heat exchanger cost reduction could therefore be expected due to the greatly reduced material cost. However, in water-to-water exchangers the material thermal conductivity is a significant component of the overall heat transfer coefficient. Therefore, the plastic exchanger will require a larger surface area, to transfer the same amount of heat under the same conditions, than a metal one; and the cost for the additional surface area will partially offset the material cost savings. However, it appears that it may be possible to obtain an overall cost savings of 50% or more using plastics.

For greenhouse applications cost savings are also expected because the material costs for the plastic are substantially lower than those for copper and aluminum, which are currently in use for such systems. For these water-to-air applications heat transfer is governed primarily by the air-side heat transfer coefficient. Therefore, use of plastics, whose thermal conductivity is lower than metals, will not significantly affect the overall heat transfer coefficient. The major drawback of using a plastic material is that fins cannot be used effectively, while in metal heat exchanger designs fins are used extensively.

In the past several years inexpensive double-wall plastic panels have become commercially available. These panels are available in a variety of materials (polypropylene, polycarbonate, etc.) and appear to be applicable for use in low-temperature heat exchanger concepts. Polypropylene panels, for instance, are currently in use as solar collector panels⁴ with temperatures reaching 65.5°C (150°F). Thus, it appears that they would be

applicable for use in waste heat systems where temperatures rarely exceed 120°F.

1.2 Purpose

The purpose of this investigation was to perform a preliminary evaluation of heat exchanger performance of these plastic panels and assess their applicability for use in waste-heat systems. This effort included both theoretical modeling of an air to water heat exchanger configuration and performance testing of a prototype. Use of the plastic panels for water to water heat exchangers was not investigated because this application was found to be less promising according to simple theoretical analysis, and the existing test facility could only handle air to water heat exchangers. Since polypropylene panels are among the least expensive and panels similar to those required for the heat exchanger are commercially available, this investigation concentrated on this material.

1.3 Description of Heat Exchanger Concept

As shown in Figs. 1 and 2, the heat exchanger consists of appropriately headered plastic panels arranged in parallel. Water flows upward through the plastic panel while air flows over the exterior of the panel in a cross-flow configuration. The size of the air passage between the panels is variable and depends upon the height of the exchanger and the panel spacing.

The panels are an extruded double-wall board that consists of two exterior "skins" with integral ribs to provide rigidity. The interior channel (formed by the ribs and exterior "skins") shape and dimensions can be varied to suit individual needs.

Although a number of materials are available, a co-polymer polypropylene panel was chosen for this initial study because it was the lowest cost panel commercially available. These panels are available in standard 1.2-m (4-ft) widths and can be cut to any desired length. Significant properties and dimensions of the panel are given in Table 1.

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Fig. 1. Plastic panel design.

Fig. 2. Plastic panel heat exchanger unit.

Table 1. Panel properties and dimensions

Plastic skin thickness	0.060 cm
Channel width	0.357 cm
Channel depth	0.505 cm
Number of channels per m of panel	177
Plastic thermal conductivity	0.12 W/m-°C

2. THEORETICAL ANALYSIS OF THE SYSTEM

2.1 Overall Heat Transfer

An analytical model was developed by predicting the heat transfer through a single panel surface and extending this to the entire heat exchanger. The panel surface itself was treated as a flat plate with convective heat transfer occurring at either surface and conductive heat transfer through the plastic wall. The overall heat transfer coefficient is expressed by the familiar equation

$$U_o = \frac{1}{1/h_o + \Delta X/k_p + 1/h_i} \quad (1)$$

where h_i and h_o are the inner (water side) and outer (air side) convection coefficients, ΔX is the plastic wall thickness and k_p is the plastic thermal conductivity.

Preliminary order of magnitude analysis indicated that h_o was the controlling quantity in the expression. In this analysis it was assumed that the water flow was laminar, therefore $h_i \sim 600 \text{ W/m}^2\text{-}^\circ\text{C}$. From the values given in Table 1, $\Delta X/k_p \sim 300 \text{ W/m}^2\text{-}^\circ\text{C}$ and for air velocities near 5 m/sec, $h_o \sim 20 \text{ W/m}^2\text{-}^\circ\text{C}$. From these values it is clear that air-side heat transfer will be the controlling mechanism. The major emphasis of the analysis was therefore directed toward evaluating the air-side heat transfer coefficient.

2.2 Air-Side Heat Transfer

Because little information is available for developing turbulent flow between parallel plates, the problem was approached by using pipe flow correlations which were modified through the use of the hydraulic diameter. For turbulent flow in a long duct

$$Nu_{Dh} = 0.023 Re_{Dh}^{0.8} Pr^{1/3}$$

where Nu_{Dh} and Re_{Dh} are the Nusselt and Reynolds numbers based on the hydraulic diameter, Dh , and Pr is the Prandtl number. This expression can then be modified for relatively short tubes with a sharp entry⁵ giving

$$\overline{Nu_L} = 0.023 Re_{Dh}^{0.8} Pr^{1/3} \left[1 + \left(\frac{Dh}{L} \right)^{0.7} \right], \quad (2)$$

where $\overline{Nu_L}$ is the average Nusselt number over the entire length, L , of the tube. Equation (2) is valid for sharp edged entries, $0.7 < Pr < 120$, and $Re_{Dh} > 10,000$, and is applicable for the plastic heat exchanger (see Table 7, Appendix). The expression

$$Nu(x) = 0.023 Re_{Dh}^{0.8} Pr^{1/3} \left[1 + 0.3 \left(\frac{x}{Dh} \right)^{-0.7} \right] \quad (3)$$

gives the local Nusselt number at any distance x from the tube entrance. For computation purposes the average heat transfer coefficient between two points a and b , $\overline{h_{a-b}}$, is desired. Integrating equation (3) yields

$$\begin{aligned} \int_a^b Nu(x) dx &= 0.023 Re_{Dh}^{0.8} Pr^{1/3} \int_a^b [1 + 0.3(x/Dh)^{-0.7}] dx \\ &= 0.023 Re_{Dh}^{0.8} Pr^{1/3} [b - a + Dh^{0.7} (b^{0.3} - a^{0.3})] . \end{aligned}$$

Therefore,

$$Nu_{a-b} = 0.023 Re_{Dh}^{0.8} Pr^{1/3} \left[1 + \frac{Dh^{0.7} (b^{0.3} - a^{0.3})}{b - a} \right]$$

and

$$\overline{h_{a-b}} = 0.023 k/Dh Re_{Dh}^{0.8} Pr^{1/3} \left[1 + Dh^{0.7} \frac{(b^{0.3} - a^{0.3})}{b - a} \right] . \quad (4)$$

The general expression for hydraulic diameter is

$$Dh = 4A/P_w ,$$

where A is the duct area and P_w is the wetted perimeter. The panels are approximated by plates which have a length ℓ much larger than the distance between the panels a , so that

$$D_h = \frac{4 a \ell}{2(\ell + a)} \approx 2a .$$

2.3 Water-Side Heat Transfer

Given typical water flow rates through the panel and the dimension of the interior channels, water flow within the panel will always be laminar (see Table 7, Appendix). For fully developed flow the expression for Nusselt number is simply

$$Nu_{Dh} = 3.66$$

and

$$h_{Dh} = 3.66 (k_w/D_h) \quad (5)$$

where k_w is the water thermal conductivity. This expression does not include the entrance effects and will give values for h_1 which are somewhat low. However, since even large differences in h_1 will not significantly affect U_o , the error introduced by ignoring entrance effects will be small.

2.4 Conduction Through the Plastic

Simple conduction was considered for heat transfer through the plastic, from the water side to the air side. No conduction was considered between water ducts or in the direction of water or air flow. These modes of heat transfer can easily be shown to be negligible.

2.5 Calculation Methods

The heat transfer expressions in Eqs. (1), (4), and (5) were used to model the overall performance of the exchanger. Because the heat exchanger is of cross flow design it was necessary to use a computer

program for calculations. Simple once-through stepping methods were used by dividing the plastic panel into small rectangular sections and calculating the heat transfer through each section. Temperature variations of density, viscosity, Prandtl number, and thermal conductivity of both air and water were accounted for in all calculations. A more detailed explanation of the code is given in the Appendix.

3. PERFORMANCE TESTING OF PROTOTYPE

To verify the performance model described in the previous section a small prototype heat exchanger was fabricated and installed in an existing heat exchanger test facility.

3.1 Description of the Prototype Heat Exchanger

As shown in Figs. 1 and 2, the heat exchanger panels used in the prototype were fabricated by attaching a 38-mm (1.5-in.) polypropylene header pipe to the top and bottom of the 0.9 m high by 0.3 m (3 ft by 1 ft) deep (in the direction of air flow) panel and capping the ends of the header. This was accomplished by milling a 6-mm (0.25-in.) slot along the length of the header, inserting the panel in the slot, and welding the board to the pipe. Plastic welding, using a polypropylene weld stick was used because a satisfactory adhesive for joining polypropylene to other plastics is not available.

Nineteen-millimeter (0.75-in.) holes were then drilled and tapped in the header end-caps. Copper ell fittings were inserted into these holes and 6-mm (0.25-in.) aluminum tubing was used to connect the heat exchanger panel to the 38-mm (1.5-in.) main distribution header at the bottom of the exchanger. An identical system was used to connect the panels to the 38-mm (1.5-in.) main outflow header at the top of the exchanger. Two inlets, one at each end of the header, were used to insure proper distribution of the water flow within the panel. Small shutoff valves were installed in the inlet lines to permit isolation of any panel that developed a leak.

When placed into the test facility, the 10 panels were fitted into a hanger which maintained a nominal 43 mm (1.7 in.) spacing between the panels. Because the plastic panels were somewhat flexible, styrofoam spacers were placed intermittently between the panels to maintain the proper spacing.

As shown in Fig. 2 the test section opening for the 10-panel heat exchanger was 0.51 m by 0.81 m (20 in. by 32 in.). The height of the opening was somewhat smaller than the panel length because insulating covers

were fitted around the top and bottom header system to insure that heat transfer was confined to the panel area.

Since each panel had 0.252 m^2 (5.6 ft^2) of heat exchange area [two sides each 0.81 m by 0.30 m (32 in. by 12 in.)], the total heat transfer area of the installed heat exchanger was 5.04 m^2 (56 ft^2). With a 43 mm (1.7 in.) spacing between the panels and a 25 mm (1 in.) allowance between the end panels and the test section wall, the area for air flow was 0.116 m^2 (4.57 ft^2). Given the channel dimensions in Table 1, the area for water flow was $9.74 \times 10^{-4} \text{ m}^2/\text{panel}$ ($.0105 \text{ ft}^2/\text{panel}$) or $9.74 \times 10^{-3} \text{ m}^2$ (0.105 ft^2) for the exchanger.

3.2 Test Facility Description

The test facility is shown schematically in Fig. 3. It functions as follows: air drawn from the building through an adjustable 0.6-m by 2.4-m door flows through a bank of electric heaters and a baffle screen (serving as a flow straightener) before entering the heat exchanger. A 0.6-m (2-ft) diameter fan, powered by a $1/4 \text{ kW}$ motor, located downstream of the heat exchanger, discharges the air into a sheet metal duct approximately 3.4 m (10 ft) long. This allows the air to mix prior to data collection. Air flow through the test facility was controlled by throttling the door inlet.

Fig. 3. Test facility schematic.

The heat exchanger inlet air temperature was controlled using a bank of six tubular, finned, electric heaters, with a total rated capacity of 6 kW. Power input to the heaters was controlled by Variac auto-transformers.

The water flow system is shown schematically in Fig. 4. The building hot and cold water were mixed in a thermostatically-controlled mixing valve to obtain the desired inlet water temperature. Total water flow was manually controlled by a throttling valve in series with two flow

Fig. 4. Water flow system schematic.

meters* (one for flows above 0.25 kg/sec, the other for flows below this value). A pressure relief valve insured that total system pressure did not exceed 2.4×10^5 Pa absolute (20 psig).

The water was fed to a distribution header at the bottom of the heat exchanger where it was distributed to the heat exchanger panels. The water flowed upward through the exchanger and was collected in the outflow header at the top of the exchanger. From the outflow header the water was emptied to the building drain.

3.3 Test Procedure

The air-flow rate, water-flow rate, inlet air temperature and inlet water temperature were fixed and temperature measurements using thermocouples were taken at various locations. These locations included: ambient air inlet (position 3 in Fig. 3), heat exchanger inlet (position 4), heat exchanger outlet (position 5), and air temperature downstream of the fan (position 6). The heat exchanger air inlet and outlet temperature were measured using thermocouples spaced vertically (0.3 m) along the heat exchanger. The inlet and outlet water temperatures were recorded using thermocouples at points 1 and 2, respectively (see Fig. 4).

Air flow measurements were taken at the end of the outlet duct using a hot-wire anemometer. An equal area method was used in which the outlet duct was subdivided into five concentric annular zones. Air velocity readings were taken at two points for each annular zone and the average was used in the air-flow calculation.

The water-side pressure drop was measured by placing pressure taps in the inlet and outlet headers and connecting them to a mercury well-type manometer. The performance tests were run fixing the following parameters as indicated: water inlet temperature 35°C (95°F), water flow rates of 0.0787 to 0.1418 kg/sec (625 to 1125 lb/hr), and air flow rates of 1.26 to 2.52 kg/sec (10,000 to 20,000 lb/hr). The minimum air inlet temperature was governed by ambient air temperatures but remained fairly constant at about 18.3°C (65°F).

*Only one of the two flowmeters was actually used in the experimental tests. The larger flowmeter proved to be unnecessary.

The air flow rates used in the tests resulted in velocities of 2.5 to 5.0 m/sec (8.1 to 16.2 ft/sec) through the exchanger. Since these velocities are typical for greenhouse heat exchanger applications, the tests matched normal greenhouse operating conditions fairly well.

Because of thermal inertia (due to the low conductivity of the plastic and the relatively low water flow) a 15-min period was allowed before the readings were taken. Experimental observations indicated this was sufficient to allow the system to attain steady-state operation after air or water flow rate changes.

3.4 List of Equipment Used

The following equipment was used in performing the heat exchanger performance tests

1. Air flow measurements

Hot-wire anemometer — Alnor Model 8500

2. Pressure drop measurements

Mercury well-type manometer — MERIAM Type W, Model A-203, Serial No. K29261

3. Water flow measurements

Rotometer — Fisher & Porter, Model OA1735M, Serial No. 6409A1839B4

4. Temperatures

Thermocouple — Copper-Constantan

Recorder — Honeywell, Model 153X64V6-X-42, Serial No. 620880

5. Temperature regulator

Powers Regulator, Series 430 Hydroguard, Serial No. 431 065

4. DISCUSSION OF RESULTS

A series of tests were performed using the heat exchange facility to experimentally determine U_o values for the prototype heat exchanger. These measured U_o values were then compared to values generated by the analytical model using physical dimensions and input stream specifications matching the test values. The objective of the tests was to determine if the theoretical analysis gave reasonable results which could be extended to determine the feasibility and/or design of a greenhouse-scale plastic panel heat exchanger.

The experimental results are presented in Table 2 and are compared to the theoretical predictions in Fig. 5. The functional relationship of U_o with air flowrate agrees well, although the analytical model underpredicts the test results by about 30% for the range considered. This

Table 2. Experimental results

Air flowrate		Water flowrate		Overall heat transfer coefficient	
kg/sec	lb/hr	kg/sec	lb/hr	W/m ² -°C	Btu/hr-ft ² -°F
1.33	10,500	0.079	625	23.9	4.2
1.32	10,500	0.110	875	25.3	4.4
1.30	10,300	0.142	1125	25.9	4.6
1.49	11,800	0.079	625	26.3	4.6
1.52	12,100	0.110	875	28.0	4.9
1.53	12,200	0.142	1125	29.4	5.2
1.52	12,100	0.142	1125	30.9	5.4
1.80	14,300	0.079	625	26.5	4.7
1.72	13,700	0.110	875	27.2	4.8
1.80	14,300	0.142	1125	30.2	5.3
2.01	15,900	0.079	625	28.3	5.0
2.00	15,900	0.110	875	32.2	5.7
1.96	15,600	0.142	1125	33.8	5.9
2.11	16,700	0.079	625	29.2	5.1
2.12	16,800	0.110	875	30.9	5.4
2.13	16,900	0.142	1125	35.2	6.2

Fig. 5. Experimental results and theoretical predictions.

fairly large difference may be due to applying a pipe flow correlation to a parallel plate geometry, although adaptation through the use of the hydraulic diameter is mathematically correct. Also, the heat transfer is sensitive to the entrance effects, and the length to diameter ratio of the air passage $L/D_h \approx 3.5$ and therefore a discrepancy on the order of 30% could be expected.

According to theoretical analysis, U_o should be a strong function of air flowrate and be nearly independent of water flowrate for the range considered. By examining the data in Fig. 5 it appears this is not the

case; U_o increases with increasing flowrate. Changing the input water flowrate in the computer model (from 0.079 to 0.142 kg/sec) does not give a detectable change from the curve shown in Fig. 5 (0.126 kg/sec). The experimental trend of U_o changing with water flowrate could be explained as some systematic error, possibly in the temperature measurement system.

If it were assumed that the water temperature measurements were in error, such that the temperature change across the heat exchanger was measured to be approximately 0.8°C greater than the true value, the apparent dependence of U_o on water flowrate would vanish. Furthermore the experimentally measured U_o values would only be 10% larger than the theoretical values (Fig. 5). This explanation certainly looks attractive; however, a simple test was performed to check the thermocouple measurement error and the results were not supportive. Room temperature water was run through the heat exchanger with the fan off (there should be no heat transfer), and although there was an apparent temperature measurement error in the proper direction, it was less than 0.3°C . Further testing for an error source was not possible.

From the comparison of the experimental data and the computer model results, it is clear that the model will give conservative estimates when used as a tool to investigate the feasibility of a large scale plastic panel heat exchanger. In order to compare the plastic heat exchanger concept to the commonly used fin-tube-type heat exchanger, a simple case study was done.

TVA operates an experimental greenhouse at Muscle Shoals, Alabama⁶ which uses a fin-tube heat exchanger as part of the heating system. A hydronic boiler is used to simulate the waste heat given off by a power plant steam condenser cooling loop. The design and performance specifications for the heat exchanger are shown in Table 3. These performance specifications were then used as criteria to size a replacement plastic heat exchanger, using the computer model as a design tool.

Table 4 summarizes the results of using the computer model to design the replacement heat exchanger. The design was accomplished by fixing the input flowrates and changing the overall dimensions to achieve the necessary air temperature change (4°C) and to minimize air pressure drop across the exchanger.

Table 3. Summary of fin tube heat exchanger specifications
at the Muscle Shoals Greenhouse

Air flowrate	18.3 kg/sec (145,000 lb/hr)
Inlet air temperature	16°C
Water flowrate	2.5 kg/sec (20,000 lb/hr)
Inlet water temperature	35°C
Heat transfer rate	70 kW (240,000 Btu/hr)
Overall heat transfer coefficient	12.4 W/m ² -°C (2.18 Btu/hr-ft ² -°F)
Number of tubes	40
Number of tube passes	4
Number of rows	2
Front dimensions	2.1 × 6.1 × .15 m
Fin pitch	4
Extended surface area	409 m ² (4400 ft ²)
ΔT air	4°C
ΔT water	7°C
ΔP air	12.5 pascals
ΔP water	62 kilopascals
Face velocity	1.3 m/sec

Table 4. Computer model design of the plastic
panel heat exchanger

Input Specifications

Air flowrate	18.3 kg/sec (145,000 lb/hr)
Air temperature	15.6°C (60°F)
Water flowrate	2.52 kg/sec (20,000 lb/hr)
Water temperature	35.0°C (95°F)

Design Results

Number of panels	50
Panel dimensions	1.52 × 1.83 m (5.0 × 6.0 ft)
Front dimensions	1.52 × 2.44 m (6.0 × 8.0 ft)
Total panel surface area	279 m ² (3000 ft ²)
ΔT air	3.9°C (7.0°F)
ΔT water	6.8°C (12.3°F)
ΔP air	4.0 pascals (.016 in H ₂ O)
ΔP water	not calculated
Face velocity	3.8 m/sec (12.5 ft/sec)
Overall heat transfer coefficient, U _o	18.3 W/m ² -°C (3.22 Btu/hr-ft ² -°F)

Examination of Tables 3 and 4 yields some interesting results.* The overall heat transfer coefficient, U_o , is higher for the plastic panel heat exchanger; however, the overall dimensions show the plastic exchanger to be 3 to 4 times larger by volume than the fin-tube type. Also the frontal air velocity is higher for the plastic heat exchanger but the air pressure drop is much lower. The large relative volume of the plastic exchanger would not be a major drawback for greenhouse applications. At worst a few feet of growing space might be lost. The lower air pressure drop would represent an energy and cost savings due to reduced fan power requirements.

The design results appear to be quite reasonable from a technical standpoint; that is, the plastic heat exchanger described in Table 4 could be built with little technical difficulty. The question of cost, however, cannot be determined by this study. Actual fabrication costs and the heat exchanger lifetime are uncertain as are the essential parameters needed for an economic evaluation of the plastic heat exchanger concept.

A check is needed to determine whether the analytical model has been properly extended, i.e., that it was used within some reasonable extrapolation limits. The heat transfer characteristics of the heat exchanger should be affected by the air and water velocities, inlet temperatures, panel spacing, and panel dimensions. Table 5 provides a comparison between these parameters for the experimental tests and the case study design results. It can be seen that no extrapolation has really been made with respect to the flow stream conditions.

The similarity between the projected design unit and the experimental unit is extensive. The one major difference is the panel size and the only effect this will have is to change the significance of the leading edge effects (entrance effect). The empirical correlation used in the model (Eq. 2) should be more reliable for cases in which entrance effects

*When interpreting these results it should be noted that the fin-tube heat exchanger does not necessarily represent the optimum design for greenhouse use, and that the user is concerned with minimizing the overall cost (capital and energy) of the heat exchange system.

Table 5. Comparison of parameters for the experimental tests and the case study heat exchanger design

Parameter	Experimental tests	Case study design
Air velocity	3.0–4.9 m/sec	3.4 m/sec
Air inlet temperature	16–21°C	16°C
Air side Reynolds number	16,000–27,000	20,000
Panel spacing	4.1 cm	4.40 cm
Panel dimensions (h × w)	0.81 × 0.31 m	1.83 × 1.52 m
Water velocity	0.0081–0.0146 m/sec	0.010 m/sec
Water inlet temperature	35–38°C	35°C

are reduced (greater panel length in the direction of airflow). It can, therefore, be concluded that the analytical model has been extended properly, and the plastic heat exchanger design is somewhat conservative.

5. CONCLUSIONS AND RECOMMENDATIONS

From comparison of the experimental data points to the analytical results in Fig. 5, it is seen that the model predictions are conservative by 30% when simulating the performance of the experimental heat exchanger unit. It is implied from these results that extending the model to simulate a larger, greenhouse scale plastic heat exchanger unit, would give somewhat conservative design results.

Because the plastic panel exchanger concept is viable from a performance and technical viewpoint, further study is necessary. A detailed engineering design/cost study could determine the design details, as well as the capital and operation and maintenance costs. A materials study would also be useful to find the best material for heat exchanger performance and lifetime.

A simple study could be done to optimize the overall design of fin-tube heat exchangers for greenhouse use. The objective would be to find the optimum balance between capital investment and energy costs. It is believed large heat exchangers with lower air and water pressure losses (and lower energy consumption) would prove a better choice than presently used units.

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Appendix

NOTES ON THE COMPUTER
MODEL CALCULATIONS

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In general terms, the calculations were done using a simple forward stepping grid method. A representative heat exchanger panel surface was divided into $n \times m$ rectangular sections as shown in Fig. 6, dividing the air flow into n streams and the water flow into m streams. It was assumed that the mass flowrate for each stream in a given direction is the same and that no mixing occurred between streams.

The calculations were performed beginning with row 1, as shown in Fig. 6, and proceeding in the direction of air flow (sections 1,1 to 1,m). The next calculations are for row 2 (sections 2,1 to 2,m) and so on for each successive row.

Fig. 6. Illustrations of grid calculation procedure.

Heat Transfer

Heat transfer and energy equations are applied at each of the $n \times m$ sections. Looking at an arbitrary panel section i, j , the overall heat transfer coefficient is found from the relations

$$U_o(i, j) = \frac{1}{\frac{1}{h_w(i, j)} + \frac{\Delta X}{k_p} + \frac{1}{h_a(i, j)}}$$

$$h_w(i, j) = 3.66 (k_w/Dh)$$

and

$$h_a(i, j) = 0.023 \frac{K_a}{Dh} Re_{Dh}^{0.8} Pr^{1/3} \left[1 + Dh^{0.7} \frac{(b^{0.3} - a^{0.3})}{b - a} \right],$$

where

$a \equiv$ distance from the leading edge of the panel to the nearest edge of the small section considered

$b \equiv$ distance from the leading edge of the panel to the farthest edge of the small section considered

$Dh \equiv$ hydraulic diameter of the panel water tubes; hydraulic diameter of the air passages

$h_a(i, j) \equiv$ air side heat transfer coefficient for panel section i, j

$h_w(i, j) \equiv$ water side heat transfer coefficient for panel section i, j

$k_a \equiv$ air thermal conductivity

$k_p \equiv$ plastic thermal conductivity

$k_w \equiv$ water thermal conductivity

$Pr \equiv$ Prandtl number

$U_o(i, j) \equiv$ Overall heat transfer coefficient for panel section i, j

$\Delta X \equiv$ plastic skin (wall) thickness.

The amount of heat transferred, $Q(i, j)$, the new air temperature $T_a(i, j + 1)$ and new water temperature $T_w(i + 1, j)$ are calculated from the expressions

$$Q(i, j) = U_o(i, j) A(i, j) [T_w(i, j) - T_a(i, j)],$$

$$T_a(i, j + 1) = T_a(i, j) + Q(i, j) / (\dot{m}_a C_{p,a}) ,$$

$$T_w(i + 1, j) = T_w(i, j) - Q(i, j) / (\dot{m}_w C_{p,w}) ,$$

where $A(i, j)$ is the area of panel i, j , $\dot{m}_a$ and $\dot{m}_w$ are the air and water mass flowrates, and $C_{p,a}$ and $C_{p,w}$ are the air and water heat capacities.

Air Pressure Losses

It was of interest to find the air pressure drop across the heat exchanger (Table 4). Pressure drop through a duct is calculated by the expression

$$\Delta p = \frac{\rho f L V^2}{2 g_c D_h} ,$$

where f is found from the implicit relation⁶

$$f^{-1/2} = 2.03 \log [Re \cdot f^{1/2}] - 0.47$$

and

$f \equiv$ friction factor

$g_c \equiv$ gravitational constant

$L \equiv$ air duct length or heat exchanger depth

$\Delta p \equiv$ pressure drop across the heat exchanger

$V \equiv$ air velocity

$\rho \equiv$ air density.

Flow Parameters and Fluid Properties

The temperature variations of air and water properties were considered throughout the calculations. Table 6 gives the actual curve fit relations which were used. These relations are only applicable to the narrow temperature ranges given in Table 7.

The parameter ranges expected to be encountered in modeling the plastic panel heat exchanger are given in Table 7. The Reynold numbers are particularly important as they show the air flow to be always fully turbulent and the water flow to be laminar.

Table 6. Property estimation correlations^a

<u>Air properties</u>	
Viscosity (ft ² /hr).....	$0.5115 + \frac{0.0767 (T - 30)}{40}$
Thermal conductivity (Btu/hr-ft-°F)....	$0.0139 + \frac{0.0101 (T - 30)}{40}$
Prandtl number.....	$0.716 + \frac{0.006 (T - 30)}{40}$
Density (lb/ft ³).....	$0.08104 - \frac{0.00613 (T - 30)}{40}$
Heat capacity (Btu/lb °F).....	0.2404
<u>Water properties</u>	
Viscosity (ft ² /hr).....	$0.07654 - 7.097 \times 10^{-4} T + 2.119 \times 10^{-6} T^2$
Thermal conductivity (Btu/hr-ft-°F)....	$0.355 + \frac{0.023 (T - 80)}{60}$
Prandtl number.....	$14.397 - 0.14168 T + 4.321 \times 10^{-4} T^2$
Density (lb/ft ³).....	$62.17 - \frac{0.78 (T - 80)}{60}$
Heat capacity (Btu/lb-°F).....	0.997

^aT is the local film temperature expressed in °F. Correlations were developed from data in Ref. 6.

Table 7. Expected parameter ranges for the plastic panel heat exchanger model and design

<u>Air Side Parameter Ranges</u>	
Velocity.....	3-8 m/sec (10-25 ft/sec)
Temperature.....	4-27°C (40-80°F)
Hydraulic diameter.....	0.05-0.12 m (0.2-0.4 ft)
Reynolds number.....	10 ⁴ -10 ⁵ (fully turbulent flow)
Prandtl number.....	0.71
<u>Water Side Parameter Ranges</u>	
Velocity.....	0.06-0.6 m/sec (0.02-0.2 ft/sec)
Temperature.....	20-55°C (70-130°F)
Hydraulic diameter.....	0.00366 m (0.012 ft)
Reynolds number.....	20-400 (laminar flow)
Prandtl number.....	3.3-6.8

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