ADVANCES IN DESIGN OF AIR-HEATING COLLECTORS

by

A JOHANNSEN

Head, Heat Mechanics Division, National Mechanical Engineering Research Institute, CSIR, P O Box 395, Pretoria 0001, South Africa

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Head, Heat Mechanics Division, National Mechanical Engineering Research Institute, CSIR, P O Box 395, Pretoria, 0001

ABSTRACT

Principles of the operation of air-heating collectors are discussed. The fundamental differences between the design principles of air-heating as opposed to water-heating collectors are highlighted. Design and operational criteria for efficient air-heating collectors are identified. The main requirement is the transfer of heat from the solar absorber to the air with minimum thermal losses and minimum pumping power.

To achieve this objective, the basic arrangement with flow of air on the shaded side of the absorber is preferred. An optimum design of the absorber-to-air heat exchanger may be obtained by providing the shaded side of the absorber with slotted or interrupted fins in a staggered arrangement. The design of such heat exchangers should be based on correlations for the developing flow regime in the hydrodynamic and thermal entry regions.

KEYWORDS

Solar energy, air-heating collectors, space heating, finned surfaces, developing flow.

INTRODUCTION

Water and air are the most common heat transfer fluids in many types of energy conversion systems. Since the early days of solar energy utilisation, water has been widely employed as the heat collection medium and the final receiver of energy in water heating systems. The technology and application of water-heating collectors, the main components of these systems, have therefore achieved an advanced stage of development. In contrast, research and development in solar air-heating only commenced in the 1940's and major commercialisation of these systems occurred only in the late 1970's.

Air-heating collectors are particularly suitable for the heating of buildings and greenhouses, for crop and timber drying, as well as for desiccant regeneration in open-cycle cooling systems. This is so because in the space heating applications, air is the final receiver of energy, while it serves as a drying agent in the case of drying and desiccant regeneration systems.

PRINCIPLES OF OPERATION

The basic principles of operation of flat-plate air-heating collectors are almost identical to those applying to the liquid types. Air is heated through contact with a sun-exposed black absorber surface which is covered by one or more transparent covers for heat loss reduction. Depending on the design, the flow of air may be above, below or on both sides of the absorber surface. The absorber surface itself may be in a variety of forms, such as flat metal sheets, corrugated metal sheets or foil, finned metal sheets, overlapped, spaced, clear and black glass plates, flow-through metal matrix, and others.

As is the case with liquid-heating collectors, thermal losses to the environment are reduced by the provision of transparent covers with or without honeycomb packing and by thermal insulation of the bottom and sides of the collector. The radiation losses from the absorber surface can be reduced by the application of selective coatings.

PERFORMANCE ANALYSIS

The classical analysis of flat-plate collectors (Whillier, 1953 and 1967; Hottel and Whillier, 1958; and Bliss, 1959) can be employed to evaluate the performance of air-heating collectors and identify ways of possible improvements. According to this analysis, the useful heat gain in the collector can be expressed as

$$Q_u = A_c F_R \left[I_c(\tau\alpha) - U_L (t_{f,in} - t_a)\right] \quad \ldots\ldots\ldots\ldots\ldots\ (1)$$

and the efficiency of the collector

$$\eta = \frac{Q_u}{A_c I_c} = F_R (\tau\alpha) - F_R U_L \frac{t_{f,in} - t_a}{I_c} \quad \ldots\ldots\ldots\ldots\ (2)$$

These are convenient expressions since the useful heat gain and the efficiency are related to the fluid inlet temperature which is usually known at the design stage. The correction factor F_R, known as the heat removal factor takes into account the fact that the fluid temperature increases along the flow path and that the absorber plate temperature is always greater than the fluid temperature. Physically, the heat removal factor can be interpreted as the ratio of the heat actually removed to that which would be removed if the whole absorber plate was operating at the temperature of the entering fluid, $t_{f,in}$. Theoretically, this would be possible if the fluid flow rate was so large that there would be no rise in temperature of the fluid passing through the collector, and that the temperature difference between the absorber surface and the fluid would be negligible.

The heat removal factor can be expressed as

$$F_R = \frac{Gc_p}{A_c U_L} \left[1 - \exp\left(-U_L F'/Gc_p\right) \right] \quad \ldots\ldots\ldots\ldots (3)$$

where F' is the efficiency factor of the collector.

For air-heating collectors, where the entire absorber surface is in contact with the air, the efficiency factor can be expressed as

$$F' = \frac{1}{1 + \dfrac{U_L}{h_{eff}}} \quad \ldots\ldots\ldots\ldots (4)$$

where h_{eff} is the effective conductance of the air-flow side of the absorber.

From Eqns (1) and (2), it is evident that for the best performance at given $t_{f,in}$, t_a and I_c, $(\tau\alpha)$ and F_R should be as high as possible and U_L should be as low as possible. Methods to satisfy the first and third requirements, i.e. to keep the optical losses and the top loss coefficient small, are well developed in the present day liquid-heating collectors. The same methods can be applied in air-heating collectors and will not be further discussed here.

Of particular importance in air-heating collectors is the requirement for a high heat removal factor, F_R. From Eqns (3) and (4), it can be seen that a high F_R will result if the air flow rate G and the efficiency factor F' are high. The flow rate however is usually limited by system requirements since, with too high a flow rate, the leaving air temperature will be too low.

From Eqn (4) it is evident that once U_L has been reduced to its practical limits, the only other way to increase F' is by increasing the effective conductance h_{eff}. Methods of achieving this goal will be discussed later.

COMPARISON OF PERFORMANCE OF WATER AND AIR-HEATING COLLECTORS

The main differences between the performance of water and air-heating collectors stem from the different properties of the two fluids. For equal mass flow rates of water and air, which is usually the case in well designed systems, the temperature rise for the air is about four times greater than for the water. This is due to the lower specific heat of air which also results in a lower heat removal factor for air-heating collectors as is evident from Eqn (3). Furthermore, the heat transfer coefficient from air in forced flow to the absorber plate is typically from 20 to 80 W/m^2K, as compared with about 200 to 500 W/m^2K for water in natural convection systems and about 1 000 to 2 000 W/m^2K for water in forced flow. As a result, the efficiency factor F' with air is lower than with water which is manifested by typical absorber-fluid temperature differences of about 15 to 25 °C for air, and 5 to 10 °C for water.

Typical temperature profiles in water and air-heating collectors used for space heating are shown in Fig. 1 (Kreith et al, 1981). In this application, the heat removal factor is typically about 0,7 for air and 0,9 for water, but due to the much lower inlet air temperature, the overall efficiency of the two collector types is roughly the same.

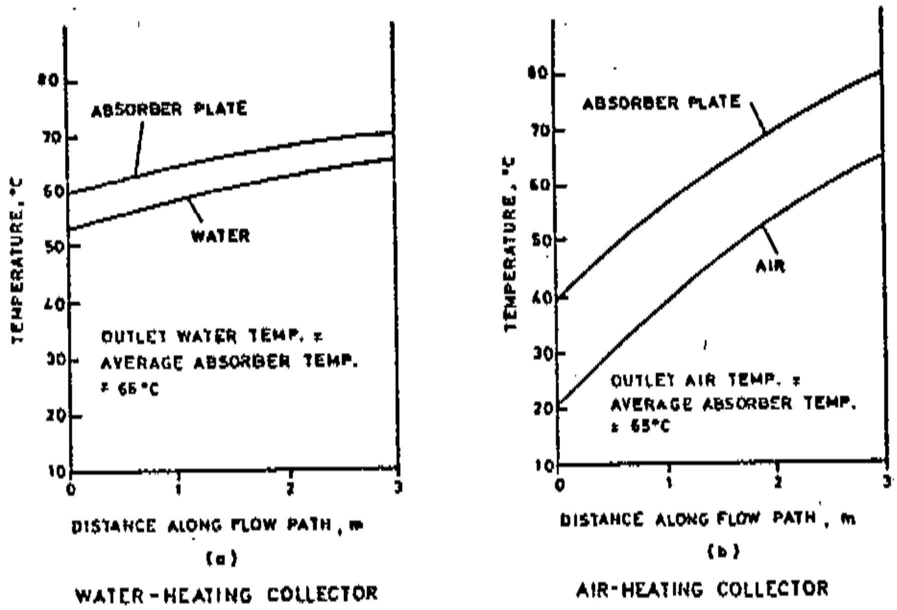

FIG. 1 : TYPICAL TEMPERATURE PROFILES IN WATER AND AIR-HEATING COLLECTORS.

A detailed comparison of design data, operating conditions and calculated performance for water and air-heating collectors is presented in Table 1. It can be seen that at the high insolation level, the two collector types operate at identical efficiency and at the reduced insolation, the efficiency of the air-heating collector is greater. However, if, at the lower insolation level, the water inlet temperature were reduced, comparable efficiencies for the two collectors would be obtained.

TABLE 1: COMPARISON OF TYPICAL DESIGN AND OPERATIONAL DATA FOR AIR AND WATER-HEATING COLLECTORS FOR SPACE HEATING

	WATER		AIR	
Design data				
Heat recovery factor, F_R	0,91		0,7	
Heat loss coefficient, U_L (W/m²K)	4,0		4,0	
Transmittance - absorbtance product $(\tau\alpha)$	0,81		0,81	
$F_R\ (\tau\alpha)$	0,73		0,57	
$F_R\ U_L$	3,64		2,8	
Operating conditions				
Ambient temperature, t_a (°C)	5	5	5	5
Fluid inlet temperature, $t_{f,in}$ (°C)	55	45	25	25
Solar radiation intensity, I_c (W/m²)	800	400	800	400
Fluid flow rate, G/A_c (ℓ/s m²)	0,012	0,012	10	10
$(t_{f,in} - t_a)/I_c$ (°C m²/W)	0,0625	0,1	0,025	0,05
Calculated performance				
$F_R\ U_L\ (t_{f,in} - t_a)/I_c$	0,23	0,36	0,07	0,14
$\eta = F_R\ (\tau\alpha) - F_R\ U_L\ (t_{f,in} - t_a)/I_c$	0,50	0,37	0,50	0,43
Computed outlet temperature, $t_{f,out}$ (°C)	63	47,9	58	39

PERFORMANCE ENHANCEMENT IN AIR-HEATING COLLECTORS

In the preceding paragraphs it was shown that the key to increased performance of air-heating collectors is the achievement of a high effective conductance between the circulating air and the absorber plate. The effective conductance can be expressed as

$$h_{eff} = \frac{(A_a + A_f\ \varepsilon)\ h_a}{A_c} \qquad \text{.............................} \quad (5)$$

The surface area A_a of direct contact between the air and the absorber plate is usually equal or slightly smaller than the absorber area A_c. One way of increasing A_a would be to circulate the air on both sides of the absorber, in which case A_a would be roughly twice as large as A_c. The main disadvantage of this arrangement is an increase in thermal losses through the collector cover due to the higher coefficient of heat transfer between the moving air and the cover.

The effective conductance can also be increased by the provision of fins extending from the absorber plate into the air flow channels. The usefulness of the fin surface area A_f depends of course on the fin efficiency ε (see Eqn (5)), which in turn is a function of the heat transfer coefficient, h_a, the fin thickness, fin length and the conductivity of the fin material.

The obvious way of increasing h_{eff} by increasing the air velocity through the air channels and thereby increasing the heat transfer coefficient h_a is limited by design considerations. The overall ratio of the air flow rate to absorber area, G/A_c, is fixed* by the required air temperature rise in the system. For a given G/A_c, the air velocity can be increased by reducing the cross-sectional area of the air channels or by reducing the number of parallel channels and increasing the length of flow path. However, both these measures result in a sharp increase in the air pressure drop through the collector and, consequently, in the fan power requirements which should be kept to a minimum.

A recently proposed design with jet air impingement on the absorber (Rask, 1977) suffers from a similar disadvantage of a high increase in pressure drop against a relatively small improvement in heat transfer.

It appears that the most promising approach to increase h_{eff} is through the provision of interrupted fins extending into the air flow channels, in a staggered arrangement. In such a design, the air flow path consists of short interrupted ducts, in which the temperature and velocity profiles of the air are not fully developed. In fully developed flow, the boundary layer acts as a thermal insulator inhibiting the heat transfer between the surface and the air.

The main effect of the interrupted fin surfaces is the prevention of the build-up of the boundary layer along the surface so that it can never become thick.

A graph showing variations of the mean Nusselt number, Nu ($= h_a d/k$) as a function of duct length for rectangular ducts with high aspect ratio under laminar flow conditions is shown in Fig. 2. It can be seen that the Nusselt number, and thus the heat transfer coefficient h_a, rises at an increasing rate with reducing duct length. Substantial improvements in h_a are therefore possible using short entry lengths as compared with a smooth duct design. Similar relationships apply to turbulent flow conditions and/or other duct geometries.

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* Strictly speaking, for a given temperature rise, G/A_c is a function of the collector efficiency which depends on h_a and in turn on the air velocity.

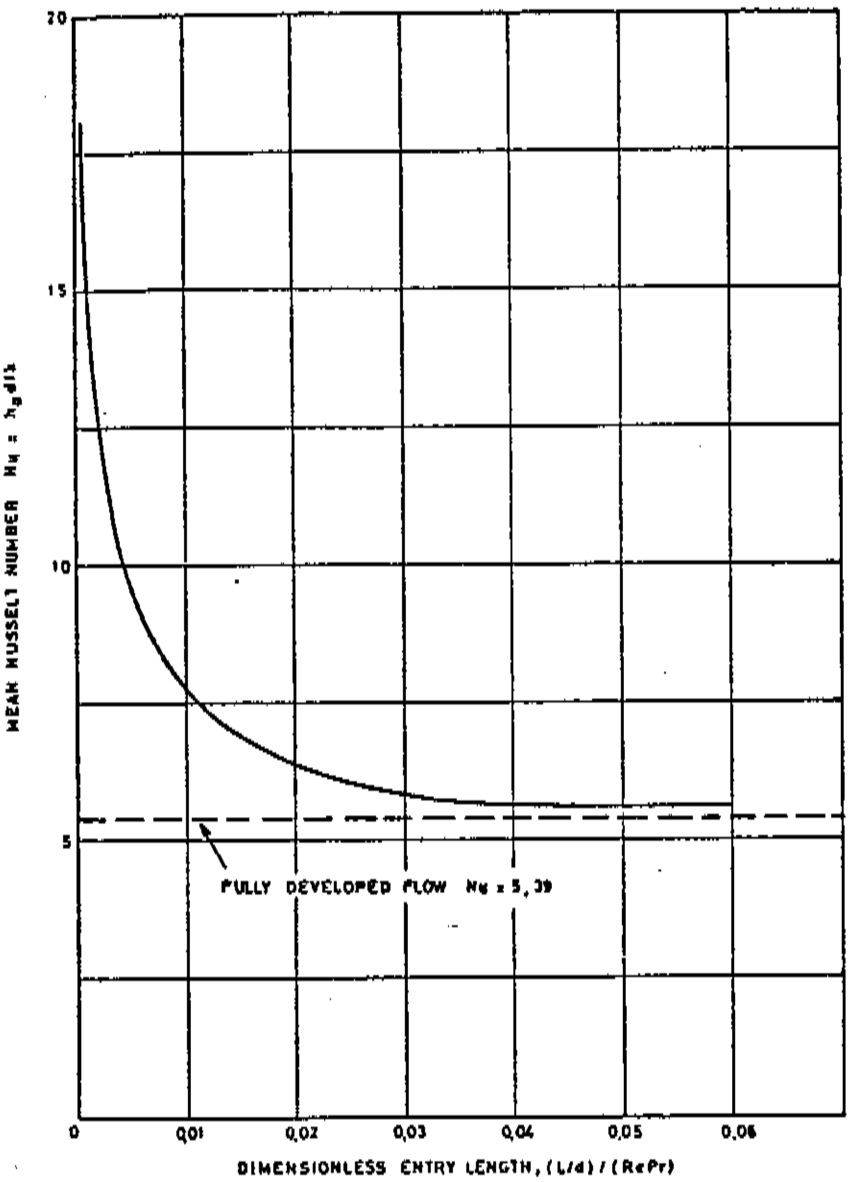

FIG. 2: MEAN NUSSELT NUMBER AT CONSTANT HEAT FLUX AS A FUNCTION OF ENTRY LENGTH FOR LAMINAR FLOW IN RECTANGULAR DUCTS OF HIGH ASPECT RATIO.

Friction factors also increase, roughly in the same proportion as h_a, with shorter entry lengths, but a small increase in h_a can more than offset a large friction factor increase because the velocity can then be reduced. Whereas h_a varies as the velocity to the power from 0,3 to 0,8, pressure drop varies almost with the square for laminar flow or cube for turbulent flow of the velocity because for a constant mass flow rate, the length of the flow path must also change with the velocity.

The principles of design of heat transfer surfaces for undeveloped flow conditions with short entry lengths are well established in the field of compact high performance heat exchangers. Generalised correlations exist which enable the optimisation of the geometrical details of the finned surface

The existing correlations are unfortunately limited to compact designs with very small air flow channels between the fins and are therefore not suitable for air-heating collectors. For approximate calculations, general correlations for single ducts such as given below (Hausen, 1948, for laminar flow; Kreith, 1973, for turbulent flow) may be employed.

For laminar flow: $$Nu = 5,39 + \frac{0,0668\ (d/L)\ Re\ Pr}{1 + 0,04\ \left[(d/L)\ Re\ Pr\right]^{2/3}} \quad \ldots\ldots\ldots\ldots (6)$$

For turbulent flow: $$Nu = \frac{Re\ Pr\ d}{4L}\ \ell n \left[1 - \frac{2,654}{Pr^{0,167}(Re\ Pr\ d/L)^{0,5}}\right]^{-1} \quad \ldots (7)$$

These correlations can only be treated as approximate for finned interrupted surfaces in air-heating collectors because in this case:

(i) the boundary layer is interrupted only on the fin surfaces and not on the plate surface;

(ii) the peripheral heat flux or temperature for a given flow passage is not uniform; and

(iii) the approaching flow is also non-uniform due to turbulence generated by the blunt fin leading edges and wake flow at the fin trailing edges.

Generally, it appears that despite the present lack of accurate design information, the development of less compact finned surfaces with high effective conductance should be possible, thus resulting in an overall improvement in the performance of air-heating collectors.

NOTATION

A_a	Absorber area in contact with circulating air, m^2
A_c	Absorber area, m^2
A_f	Fin area, m^2
c_p	Specific heat of fluid, J/kg K
d	Equivalent diameter of passage = 4 x flow area/wetted perimeter, m
F_R	Heat removal factor defined by Eqn (3)
F'	Efficiency factor defined by Eqn (4)
G	Mass flow rate of fluid, kg/s
h_a	Heat transfer coefficient from absorber to circulating air, W/m^2K

h_{eff}	Effective conductance defined by Eqn (4), W/m^2K
I_c	Solar radiation intensity reaching collector, W/m^2
k	Thermal conductivity, W/m K
L	Length of flow path, m
Nu	Nusselt number = h_a d/k
Pr	Prandtl number
Q_u	Rate of useful heat gain in collector, W
Re	Reynolds number referred to d
t_a	Ambient air temperature, ^oC
$t_{f,in}$	Fluid inlet temperature, ^oC
$t_{f,out}$	Fluid outlet temperature, ^oC
U_L	Heat loss coefficient of collector, W/m^2K
ε	Efficiency of fin
η	Efficiency of collector
$(\tau\alpha)$	Transmittance - absorptance product

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