SOLAR AIR HEATING SYSTEMS FOR THE INDUSTRY: A NEW APPROACH FOR THEIR DESIGN AND DEVELOPMENT

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Abstract:

Many local industries seem to have the potential to replace a significant fraction of their conventional thermal energy with solar energy from air heating collectors as indicated by studies on tea industry. The few experimental or pilot level solar air-heating systems established in the country have suffered due to inadequate engineering design and defects in their installations. This paper presents a new engineering design approach based on factory built standardized collector cells aimed at achieving better performance and cost effectiveness. Generation of better design data and development of low-cost solar collector cell fabrication methods are needed in order to develop this approach further. The approaches and methodologies suggested here can contribute to the goal of developing the local capacity for utilization of solar energy in the industry

Introduction

Simple technologies have been successfully used world over for many decades to utilize solar energy for air heating applications such as building heating, drying, and chemical process industry. However, no significant utilization of solar air heating is evidenced in Sri Lanka as yet. Our tropical climate with abundant sunshine and a multitude of industries utilizing low to medium grade heat energy provide vast opportunities for the utilization of this benign source of energy. Typical industrial use of hot air in Sri Lanka include; drying/withering of tea, rubber drying, drying of agricultural crops including spices, food drying, curing of ceramic ware, drying of paint work, and drying of timber. Most of these applications require hot air within the temperature range of 35 to 100 C. A study on the tea sector has shown that the technical potential of replacing conventional energy in drying with solar energy is in the range of 30–55 % in most of the tea growing regions (Weeratunga Arachchi, K.). Solar air pre-heating collectors operating at 50% thermal efficiency have been shown to deliver this performance. A study by Laing Design and Development Centre, UK considered a combined solar heat collection and a modified energy efficient tea drying process and concluded that $480m^2$ air collector system was adequate in supplying 80% of the heat requirements of a factory processing 800 kg of green leaf per day over 8 hours. A similar potential for conventional energy replacement can be expected from most other industries utilizing low to medium temperature process air.

Solar air heating is known to have been applied in local industries, including tea industry, at experimental or pilot level with a lower degree of success. The largest of these systems known to the author is the 400 m^2 of air pre-heating collectors installed at the tea factory at the low country tea research station at Ratnapura under a SAREC funded project titled "Solar Wood Gasifier Energy in Tea Processing" during 1994–1999 (Ziad Mohamed). It was based on the conventional design consisting of an array of small-area ($2m^2$) collectors connected in parallel by a network of air ducts. The collectors were also custom-designed for the particular application. The system showed a saving of 25–34% of the fuel oil during the preliminary trials. The capital cost of the system was reported to be high due to the large number of collectors and the associated ductwork. Project investigators suggested that larger collectors and less ductwork would be more economical. This project was expected to demonstrate the technical and economic feasibility of solar air heating systems in tea industry and to encourage similar installations in other industries such as rubber. This objective has not been achieved.

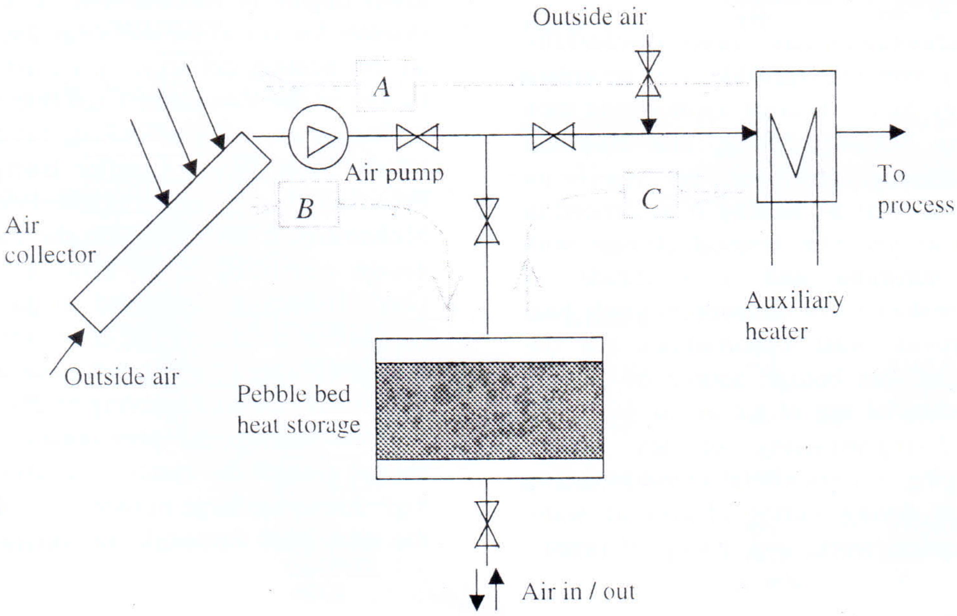
Although the performance of industrial solar air heating systems has not been well documented, one very likely reason for the failure or poor performance of these industrial systems seem to be their inadequate engineering design. Another reason is poor collector fabrication and installation resulting in low reliability (Attalage, R.A.). The present work aims at developing a new methodology for the design of solar air heating systems that would ensure better performance both technically and economically.

Industrial Solar Air Heating Systems (SAHS)

Basic elements of a SAHS are the solar collector, pebble bed storage, auxiliary heater, air pump, and a control mechanism (Figure 1). A majority of potential systems would be in retrofit applications where partial replacement of conventional energy used by existing systems is achieved. Such retrofit systems should be designed to supply hot air at existing process temperatures. The other sector is the new applications where a solar air heating system is included in the design of a new industrial process right from the beginning. Since solar air heating collectors operate more efficiently at lower temperatures, the industrial process itself should be examined in such cases, to see if the temperature of energy delivery could be optimized (Duffie and Beckman).

In retrofit systems, the existing conventional air heater would become the auxiliary heater. This auxiliary heater running on conventional fuel, such as fuel oil, diesel, or fuel wood, would be fitted with a control mechanism so as to maintain a constant temperature of hot air supplied to the process, since the degree of solar preheating would have a variation over the day. In applications that requires heating even outside daylight hours, solar energy storage would be required. Pebble beds are very commonly used for storage of solar energy from air collectors. Air heating collector in Figure 1 always heats outside air and there is no re-circulation of process air back to the air collector. These are known as once-through air collectors or air pre-heating collectors. In this paper, the discussion will be limited to SAHS employing air collectors of this type since a vast majority of potential industrial applications of SAHS in the country would fall into this category.

Figure 1 - General Schematic of Solar Air Heating System



There are three basic modes of operation of these systems. The dashed lines indicate the directions of airflow for each of these modes, *A*, *B*, and *C*.

Mode *A*: Direct delivery of air from the collectors to the process occurs.

Mode *B*: In the absence of process heat loads during daytime, hot air from collectors is diverted to the storage. Heat is stored in the pebble bed.

Mode *C*: Heat stored in the pebble bed is recovered and supplied to the process during nighttime.

In modes A and C, the auxiliary heater would be controlled to keep the temperature of hot air delivery at the required level.

Design of Solar Air Heating Collectors

Solar air collector is the key element in the system. Its design would critically affect the performance of the whole system. The prediction of the performance of an air-heating collector forms the basic approach towards their design. The performance is predicted based on theoretical equations presented in the appendix, which follows the analysis of Niles et.al. and Duffie and Beckman. Back pass type flat plate collectors with a single transparent cover have been considered in the analysis owing to their popular use in industrial applications. Predicted performance depends on the material properties of collector elements, the choice of design solar radiation (I), and a few other factors.

Table 1- Solar Collector Design Parameters

Parameter	Value
ρ	1.092 kg/m^3
k	0.0273 W/m.K
μ	1.963 e(-3) Pa.s
C_p	1007. J/kg/C
T_a	30 C
I	700 W/m^2
$(\tau\alpha)$	0.72
U_b	2.0 W/m^2/C
N	1
β	20^0
ε_g	0.88
ε_p	0.95

The term I can be taken as the hourly average solar radiation. Since solar radiation varies with time of the day and day of the year in addition to the local weather, selection of a design value for I is to be done with caution. I is not to be misunderstood as the radiation averaged over a typical day since this value will lead to too low air delivery temperatures. I can vary up to about 1000 W/m^2 measured on a horizontal surface at noon on a clear day. A solar air system design handbook, based apparently on European and North American experience, suggests a design radiation value of 600 W/m^2 (S. Robert Hastings and Morck Ove). Being a tropical country with abundant sunshine, solar heating system designs in Sri Lanka may well be based on a radiation value of 700 W/m^2. Parameters used in the prediction of air collector performance in this study, are given in Table 1.

The performance of air heating solar collectors is highly dependent on the airflow rates and air channel dimensions. For instance, the efficiencies of two identical collectors (L = 20 m, B = 5 m) except for the depth of the air channel are compared in Figure 2 at various air-flow rates (values marked against radial chain lines are specific mass flow rates of air in kg/h.m^2). Collectors with the shallow channel depth always exhibit greater efficiencies due to the fact that *Re* is greater and convective heat transfer within the air channel is enhanced. However, this improvement in efficiency is obtained at the expense of greater pressure losses in airflow across the collector. This in turn increases the pumping power requirements. Therefore, in the design of air collectors one has to make a compromise between the improved efficiency and increased cost of pumping air.

The pressure loss in the air channel is given by;

$$\Delta p = \rho g \Delta h = \rho g f \frac{L}{d_h}\frac{V^2}{2g} \quad ; \quad d_h = \frac{2Bd}{(B+d)} \approx 2d$$

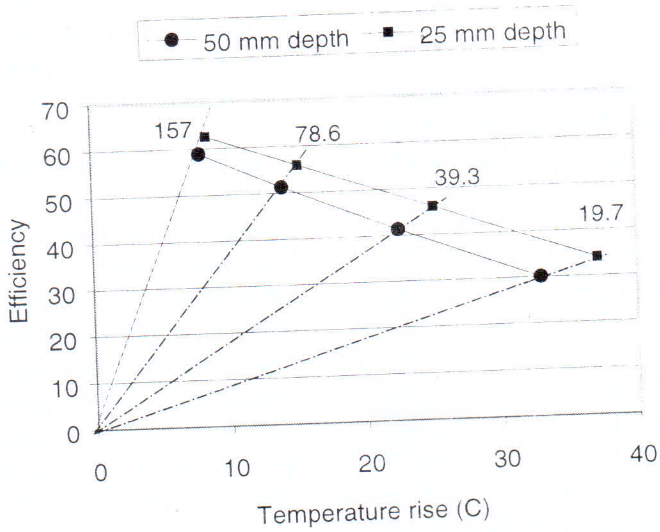
$$f = 0.262\,Re^{-0.2} \quad \text{(Niles et.al.)}$$

Since air velocity $V = \dfrac{m}{\rho B d}$,

and f is a weak function of Re and hence d.

$$\Delta p \propto (1/d^{2.8}, L) \quad \text{for a given air flow rate.}$$

Figure 2 - Comparison of Efficiencies between Collectors of Different Air Channel depths



This means that Δp varies inversely with nearly the cube of air channel depth and proportional to the length. Therefore, sizing of the air collector must consider the pressure loss across the collector and associated fan power.

A judgement has to be made in the design of solar air systems as to what would be a permissible or tolerable pressure loss in air collectors. A handbook on solar air system design suggests, as a rule of thumb for building heating applications, that the pressure drop across each commercial collector of 2 m^2 area must be kept at 8–10 Pa/m^2. This corresponds to a yearly fan energy consumption of less than 4 kWh/m^2 (S. Robert Hastings and Morck Ove).

$$\frac{\text{Fan Power}}{\text{Heat Gain}} = Z\left\{\frac{(m/\rho).\Delta P}{m C_p \Delta T}\right\} = R$$

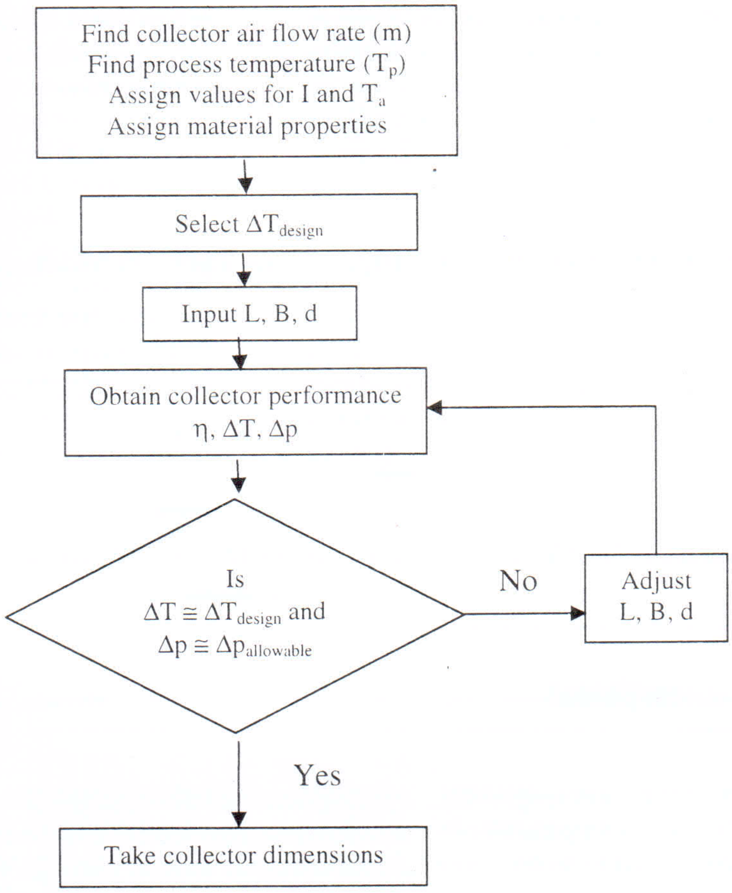
A different approach is suggested here. The allowable collector pressure drop is to be based on the ratio of energy spent for pumping air over the heat energy gained by the collector.

where ΔT is the rise of air temperature across the collector. Z is assigned a conservative value of 4.0 in order to account for parasitic pressure losses in air ducts, mechanical efficiency of the fan, and variation of ΔT with solar insolation over an average day. R is assigned a somewhat arbitrary value of 2% at this stage. Better values for Z and R can be assigned at a later stage after adequate local experience with industrial solar air heating systems has been gained. Then,

$$\Delta P_{allowable}\ (Pa) = 5\ \Delta T\ (C)$$

This criterion will be taken in the design of air collectors treated in the following sections, to optimize their thermal performance.

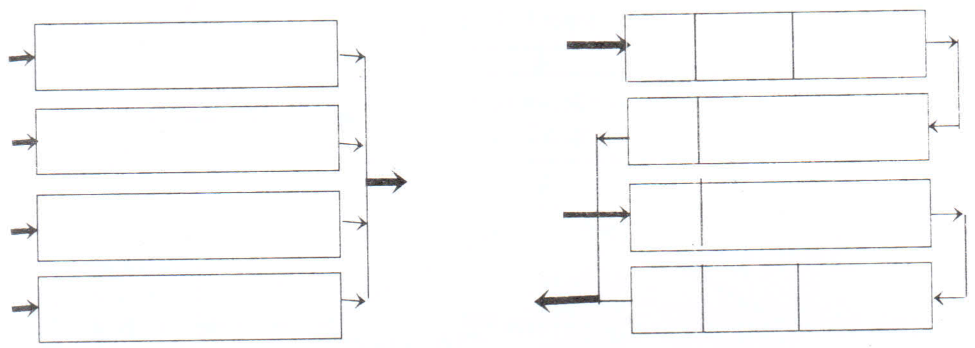
Figure 3 - Industrial Air Heating Collector Sizing Flow Chart



Theoretical equations governing the thermal and hydraulic performance of flat plate collectors can be used in their design for a particular application. The basic design process is outlined in the flow chart in Figure 3. The goal of the design exercise is to determine the dimensions of the solar collector made of selected materials that satisfy the design requirements with regard to the allowable pressure drop ($\Delta P_{allowable}$) and temperature rise (ΔT). This process, however, has not considered the economic performance that forms the primary design criteria of a solar energy system in general (Duffie nad Beckman). Selection of a design value for ΔT must be done so that the outlet temperature will not exceed the process temperature for significant periods of time, leading to energy dumping and hence lowered collector performance. It must also be remembered that the actual incident radiation levels may well exceed the design radiation level of 700 W/m^2 and reach even 1000 W/m^2.

The design engineer can offer many alternative designs that can deliver hot air at the required flow rates and temperatures. These alternatives will concern different collector system configurations – a system made up of a large number of individual small-area collectors connected by a complex network of ducts, or a system made up of a few large-area collectors. The first design alternative based on a large number of small-area collectors is not obviously preferable due to the technical and economic reasons discussed earlier. Systems composed of large-area collectors can be assembled with a number of collector modules connected in parallel combination making an array, or a number of collector modules in parallel combination where each module is in turn made up of a number of smaller collector modules connected in series (Figure 4). Selection of the length and the breadth of large area collectors can be, in most of the cases, constrained by space available for the erection of the collector or the collector construction methods. For instance, if the collector is to be installed on the existing factory roof, the size of the roof will determine the limiting values of collector dimensions. Excessively large breadths may lead to construction flaws and poor flow distribution within the collector. It is, therefore, necessary to keep the breadth of the collector within manageable limits and design for a collector array made up of few collector modules in parallel or series-parallel combination.

Figure 4 - Different Flow Configurations Through Modules of Collector Array



(a) parallel (b) series-parallel

Neglecting the entry length effects in long collector modules, the series-parallel combination essentially works as a parallel combination where the collective thermal performance of modules connected in series is similar to one lengthy module making up the total length.

Table 2 - Various Collector Module Sizes and Configurations Suitable for a Tea Drying Application

Module size L(m)x B(m)x d(mm)	Collector array configuration	Air flow through a collector module (l/s)	Array area (m^2)	Efficiency (%)
150 x 5 x 65	Single module	3700	750	36.2
75 x 10 x 35	Single module	3700	750	36.4
30 x 25 x 15	Single module	3700	750	36.8
75 x 5 x 35	2 parallel modules	1850	750	36.4
30 x 5 x 15	5 parallel modules	740	750	36.8
25 x 3 x 13	10 parallel modules	370	750	36.8
25 x 1 x 13	30 parallel modules	123	750	36.8
20 x 0.75 x 10	50 parallel modules	74	750	37.0

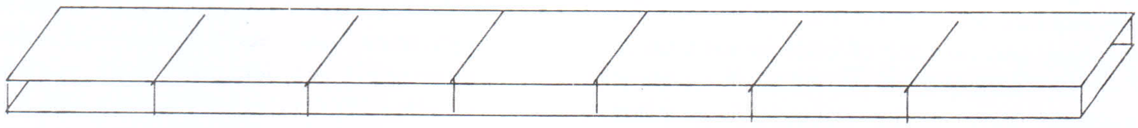
To elaborate on alternative designs, an example of delivering hot air to a tea dryer is considered. The dryer is an endless chain pressure dryer and has a out put capacity of 100 kg of made tea per hour and requires hot air at 90 C. Air flow rate is fixed at 222 m^3/min (3700 l/s) based on recommended operating parameters by Samaraweera. Using the design equation in the appendix, a solar air heating system to heat air by 40 C is designed. The pressure drop across the collector array is set at 200 Pa according to our design criteria. Various collector module sizes and configurations can be considered in this design and Table 2 shows some of the selections. The final choice of the collector configuration will depend on factors such as the space available for erection of collector modules, ducting between the collector array and the dryer, and installation cost.

Air heating collector modules can be fabricated in-situ, at times integrated with the existing structures such as roofs. To ensure the design performance, the collectors have to be:

I. air tight
II. well insulated
III. fitted with water proof covers, and secured to the supports carrying self weight and wind loads.

Building such an air collector could well be a laborious task of fabrication in outdoor conditions, and often at elevated heights, using highly skilled labour. Such fabrications could significantly add to the cost of solar heating systems.

Figure 5 - Flat Plate Collector Module Made up of a Number of Cells Connected in Series



One good alternative to overcome these problems is to assemble collector modules in the field from factory built collector cells (Figure 5) having favourable characteristics such as;

I. light weight enabling easy handling in transport and assembly.
II. made of materials of standard shapes and sizes to avoid wastage
III. manufactured to standard sizes and precise specifications using better techniques and tools
IV. manufactured in large quantities at factory in order to lower the cost.

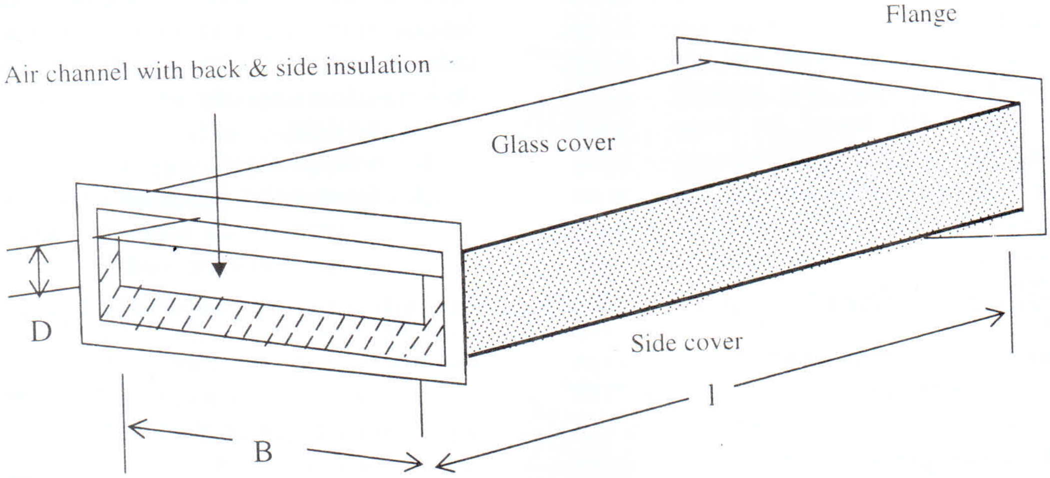
Next section discusses the sizing of a standard collector cell and the performance of collector modules made up of these cells.

Design of SAH Systems Based on Standardized Collector Cells

The major dimensions of the collector cell (Figure 6) are selected such that it gives the favourable characteristics mentioned earlier. Standard sizes of glass panes, iron sheets, insulation materials and other structural materials used in the fabrication of the collector cell are considered in its sizing in order to avoid the wastage of materials. A standard length of l = 2.4 m and breadth of B = 0.85 m has been chosen. The depth of the air channel cannot be set at one standard value because of the wide range of airflow that is encountered in industrial applications. This problem is overcome by manufacturing standardized collector cells that offer a number of channel depths. Based on theoretical performance predictions, a choice of 30 mm, 60 mm, 90 mm, and 120 mm channel depths has been made in order to handle airflows in the range of 260 – 1300 l/s for a range of 10-40 C temperature rise.

These collector cells will then be connected in-situ to form the collector modules of desired lengths. Table 3 indicates the exact airflow for each type of cells at different temperature levels. The design process outlined in Figure 3 is used to obtain the performance of a collector module by setting length L equal to the number of cells times the cell length.

Figure 6 - Geometry of the Standardized Collector Cell of Length l



Data in Table 3 can conveniently be used to design solar air collectors without going through the detailed design calculations. First, the required temperature rise is decided. Then, the type and number of cells in each module can be selected form the table. A number of modules have to be connected in parallel combination so that the required airflow rate is obtained. Now the collector array configuration is evaluated with regard to space availability for the array; layout of air ducts and manifolds connecting air collector array with process equipment; and any cost estimates. The best array configuration can be selected, out of the available options, in this manner.

If the previous design example on tea drying application is reconsidered, 15 modules connected in parallel combination, each having 26 cells of D30 type, can deliver 3900 l/s air at 40 C temperature rise. When the airflow is set at 3700 l/s, the delivery temperature will be slightly greater than the design value and the collector array efficiency will also drop slightly. This, however, is well accepted for a design of this nature.

Table 3 - Characteristics of Collector Modules Made up of Different Cell Types for Applications at Various Temperature

Temperature Rise (C) / Δp (Pa)	Number and Type of Cells	Air Flow through Module / a cell (l/s)	Specific Flow Rate (kg/h.m²)	Efficiency (%)
10 / 50	5 of D30*	315	115	57
	11 of D60*	650		56
	16 of D90*	1000		56
	22 of D120*	1300		55
20 / 100	10 of D30	300	52	52
	22 of D60	600		50
	35 of D90	950		50
	45 of D120	1250		50
30 / 150	17 of D30	275	30	44
	35 of D60	575		44
	55 of D90	875		43
40 / 200	26 of D30	260	19.5	36
	52 of D60	525		36

*type D30 has 30 mm deep air channel, type D60 has 60 mm deep air channel and so on)

Concluding Remarks

The design of solar air heating systems needs careful analysis taking into account technical and economic aspects. This paper presents an analysis of the engineering design procedure aiming at large-area collector modules assembled from factory built standardized collector cells in order to obtain a well-engineered system. Previous local experience indicates that such an approach is likely to lower the cost of industrial scale solar air heating systems while improving the reliability of performance. In addition, this new approach has led to a simple way of sizing a SAH system based on a set of tabulated data such as those shown in Table 3. Actual sizing of the standard collector cells, however, will have to be based on performance measurements as well as economic considerations.

This paper has suggested a solar radiation value of 700 W/m^2 and a design optimization criterion for large-area collectors based on

$$\Delta p = \text{Constant} \,.\Delta T$$

to be used in the design of SAH systems for industrial use. These criteria have been suggested as an initial step in the development of local design practice for SAH systems for industrial use and the criteria ought to be refined as more local experience is gained with such SAH systems.

A well-planned research and development program is needed now to launch this approach in design of solar industrial air heating systems. First, design data needs to be collected through a series of tests on collector cells made of various locally available materials. Such tests will reveal the real collector performance in relation to pressure losses and efficiencies. Fabrication methods for collector cells need to be developed as well, taking cost factors into consideration. Finally, a pilot-scale industrial solar heating system assembled from such standardized air collector cells, needs to be established.

Such experience is needed, if expanded use of solar energy resource for the industry is to be achieved in the future.

There exists local expertise and skills that can be well utilized for the design, fabrication, and installation of solar air heating systems- a feature that makes the solar air heating technology a most favorable one for developing countries. This expertise has to be pooled and developed through local research, development, training, and pilot scale projects.

Nomenclature

A_c - area of the collector plane receiving insolation (m^2) = L . B
B - width of collector (m)
C_p - specific heat capacity of air (J/kg/C)
F_o - heat removal factor referred to outlet temperature (-)
F' - collector efficiency factor
h_1 - coefficient of convective heat transfer from absorber to air stream (W/m^2/C)
I - incident solar radiation normal to the collector plane (W/m^2)
L - length of collector (m)
m - mass flow rate of air through the collector (kg/s)
T_o - outlet air temperature (C)
T_a - ambient air temperature (C)
U_l - overall coefficient of heat loss from heating fluid (air) to surrounding (W/m^2/C)
$U_l = U_t + U_b$
U_t - coefficient of top heat loss (W/m^2/C)
U_b - coefficient of bottom heat loss (W/m^2/C)
η - heat collection efficiency / collector efficiency (-)
$(\tau\alpha)$- combined optical efficiency of transparent cover and absorber surface (approximately solar transmittivity through transparent cover times solar absorptivity of absorber surface) (-)

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APPENDIX I

Theoretical Equations Governing Collector Performance

Theoretical equations that govern the performance of flat plate type collectors are presented in this section.

Figure A1: Schematic of Solar Air-Preheating Collector

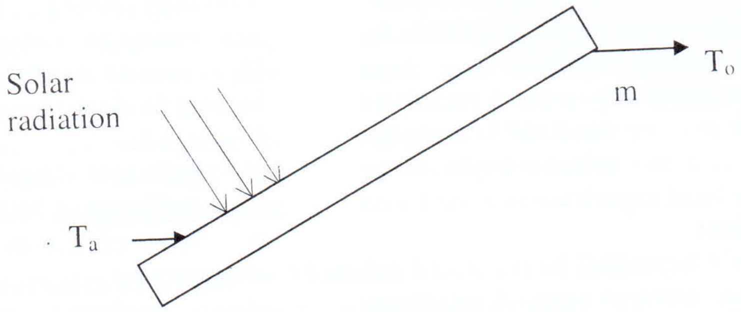
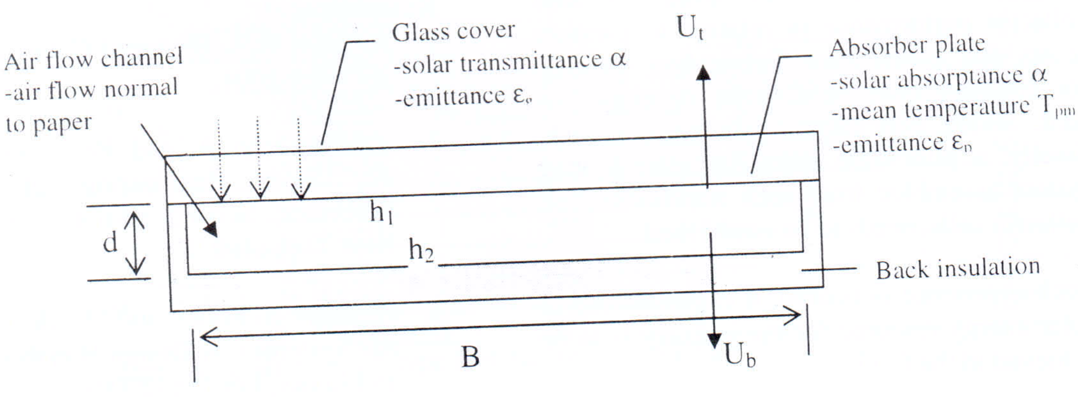


Figure A2: Cross-Section of Solar Flat Plate Collector



Following the analysis of Niles et.al. and Duffie and Beckman, the performance of a solar air-heating collector (Figures A1 and A2) can be given by;

$$\eta = \frac{mC_p(T_o - T_a)}{A_c I} = F_o(\tau\alpha) - F_o U_l \frac{(T_o - T_a)}{I} = F_R(\tau\alpha) - F_R U_l \frac{(T_i - T_a)}{I}$$

$$F_o = \gamma\{e^{F/\gamma} - 1\} \;\; ; \;\; F_R = \gamma\{1 - e^{-F/\gamma}\} \;\; ; \;\; \gamma = \frac{mC_p}{A_c U_l}$$

$$F' = \frac{1}{1 + \cfrac{U_l}{h_1 + \cfrac{1}{1/h_2 + 1/h_r}}}$$

$$U_l = U_t + U_s$$

U_t can be accurately estimated from correlations such as Klein's (Duffie and Beckman).
U_t, in this case, depends on parameters such as mean collector plate temperature (T_{pm}), number of glass covers (N), collector slope (β), emissivities of glass and collector plate (ε_g and ε_p), ambient temperature, and wind heat transfer coefficient (h_w).

h_1 has been estimated by Niles et.al. using Reynolds analogy with an experimentally determined friction factor based on measurements on 70 feet and 140 feet long collectors.

$$Nu = 0.033\, Re^{0.8}\, Pr^{1/3}$$

$$Re = \frac{2m}{\mu(B + d)}$$

h_2 has been assumed equal to h_1.

$h_r = \dfrac{4\sigma T_{pm}^3}{2/\varepsilon_p - 1}$ - coefficient of radiation heat transfer between absorber plate and bottom of air channel (W/m^2/C)

When the collector dimensions and material properties, environmental conditions (I and T_a), and airflow rates are specified, above equations can be used to predict the collector efficiency. U_t has to be estimated in an iterative manner starting with a reasonable value for T_{pm} and iterating to meet the condition:

$$T_{pm} = T_a + \eta I\left(\frac{1 - F_R}{F_R U_l}\right)$$