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Design and Performance of Evacuated Solar Collector Microchannel Plates

Roger W. Moss[1] and Stan Shire[1]

[1] School of Engineering, University of Warwick, UK

Abstract

Solar thermal collectors for buildings have traditionally used either a flat panel or evacuated tube design. A high vacuum (<1 Pa) can eliminate heat loss via gas conduction effects, thereby increasing efficiency in cold climates or in applications requiring elevated delivery temperatures e.g. for process heat. Combining the two established technologies, an evacuated flat panel design would be more architecturally elegant than tubular collectors and have a better fill factor, yet be thinner and more efficient than conventional flat panels.

Beyond the efficiency gains from vacuum insulation, the collector design aims to limit temperature non-uniformity (with its increased radiative losses) by achieving homogeneous flow across micro-channels within the plate. A design methodology has been adopted that optimises the channel hydraulic diameter in terms of plate-to-fluid temperature difference and power required to pump the fluid. Two collectors (aluminium, stainless steel) have been designed and fabricated, with machined plates and laser-welded joints.

Keywords: Solar collector, thermal, vacuum, flat panel, heat transfer, heat recovery factor, optimisation, laminar flow, micro-channel, pressure drop, pumping power, thermal expansion, laser weld.

1. Introduction

Conventional solar collectors use either a “flat panel” design or an “evacuated tube” design. The former is convenient to use but suffers from heat losses through its internal filling gas (usually air); the latter uses a vacuum insulation gap to avoid conduction and convection losses, but can suffer from a poor fill factor in terms of the fraction of the available area covered by the heat-absorbing surface.

Current flat plate designs achieve usefully high efficiencies (>60%, Chen et al, 2012) in hot locations with high insolation and with selective emissivity coatings to reduce radiative losses. At higher and colder latitudes or with a higher temperature requirement for process heating the combination of lower irradiance and higher heat loss will lead to reduced efficiency unless heat losses can be further reduced. Heat losses from the plate could be reduced by using a double-glazed cover but this would lead to increased reflection losses and panel weight; vacuum insulation has the potential for higher efficiencies and lighter weight (Benz & Beikircher 1999).

Many workers have studied the optimisation of flat panel collectors. Bracamonte (2013) found that for solar air heaters, the optimal plate shape was a function of Mach number. Eisenmann (2004) optimised plate geometry in terms of metal content. Jones (1987) calculated heat removal factors as a function of Peclet number. Chen (2012) measured the effect of varying flow rate on the efficiency. Do Ango et al (2013) studied the effects of air gap, flow rate and plate length on the efficiency of a polymer flat plate. Sharma (2011) recognised that “the optimal dimensions of the minichannel are governed by a balance between the gain in the heat transfer rate and the increase in the pressure drop” but did not explicitly calculate an optimum dimension. Farahat (2009) calculated the exergy efficiency of a flat plate collector as a function of pipe diameter and flow rate. Hegazy (1996, 1999) calculated the optimum channel depth, to maximise heat gain for a given pumping power, for turbulent flow in a solar air heater; the present work reaches an equivalent result for laminar flow of a fluid. Mansour (2013) built a mini-channel plate with improved performance relative to a conventional design, albeit with increased pressure drop.

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In the current conceptual design (Henshall, 2014), a high vacuum enclosure eliminates gas conduction and convection losses; since there is no gas conduction, the collector spacing between collector and cover glass can be made small (<1cm) without increasing the losses, thereby allowing a slim and architecturally attractive format (see also Fiaschi & Bertolli (2012), who investigated solar roofs with esthetic appeal, and Motte et al (2013) who patented a slim solar collector for building integration without visual impact). This paper describes the design and manufacture of a micro-channel collector plate suitable for use in such a vacuum enclosure and for testing under vacuum conditions in a solar simulator. Its role is to investigate the possibilities, benefits and limitations of a micro-channel design in the context of a thin flat panel collector; it will in due course be compared with alternative designs based on hydro-formed or extruded sheets.

2. Nomenclature

a,b	Rectangular passage width and depth
A_c	collector top surface area, $A_c = LB$
B	breadth of collector plate
c	fluid specific heat capacity
D_h	channel hydraulic diameter
f	Fanning friction factor
F_R	heat recovery factor
G_T	total (beam & diffuse) irradiance from Sun
h	glass to air external heat transfer coefficient
k	thermal conductivity (fluid)
L	passage length
$\dot{m}$	total mass flow rate
Nu_{cf}	Nusselt number for laminar flow, constant heat flux boundary
P	pitch of circular passages
Po	Poiseuille number, $f = \frac{\mathrm{Po}}{\mathrm{Re}}$
Q_u	rate of heat extraction by fluid
R	hydraulic diameter to pitch ratio for circular passages
Re	Reynolds number
S	solar power absorbed by the collector, per square metre
T_a	ambient temperature
T_i	fluid temperature at inlet to collector
T_o	fluid temperature at outlet from collector
T_{pm}	collector plate mean surface temperature
ΔT	the optimisation target to be minimised , $\Delta T = T_{pm} - T_i$
$\Delta \overline{T}_f$	temperature rise of fluid (mean along passage)
ΔT_h	temperature difference driving convective heat transfer
U_L	overall collector heat loss coefficient
W_p	fluid pumping power (frictional dissipation through the collector) W/m^2
μ	fluid dynamic viscosity
ρ	fluid density

η_i Efficiency defined as a function of inlet temperature

$\tau\alpha$ Effective transmittance-absorptance product

3. Thermo-fluid design of flow passages

3.1. Introduction

For simplicity we have adopted the terminology defined by Duffie & Beckman (1991):

A heat loss coefficient U_L is defined in terms of the absorbed heat S and the plate mean temperature T_{pm}

$$Q_u = A_c \left[S - U_L \left(T_{pm} - T_a \right) \right] \qquad \text{(eq.1)}$$

This heat output is less than if the whole plate were held at the fluid inlet temperature T_i; this effect is described in the Hottel-Whillier-Bliss equation by a heat recovery factor F_R:

$$Q_u = F_R A_c \left[S - U_L \left(T_i - T_a \right) \right] \qquad \text{(eq.2)}$$

More generally, one can define a collector efficiency:

$$\eta_i = \frac{Q_u}{A_c G_T} = F_R \left[(\tau\alpha) - \frac{U_L \left(T_i - T_a \right)}{G_T} \right] \qquad \text{(eq.3)}$$

To maximise the efficiency one must first maximise $\tau\alpha$ and minimise U_L. Measures to achieve this include the use of a high clarity glass such as Pilkington's OptiWhite™, possibly with anti-reflection coatings; a selective coating (high absorptance at short wavelengths, low emittance at long wavelengths) such as black chrome and a high vacuum to eliminate gas conduction. Gas pressures below 100 Pa start to eliminate convection; pressures below 0.1 Pa eliminate conduction too, for a typical glass-collector spacing (Figure 1).

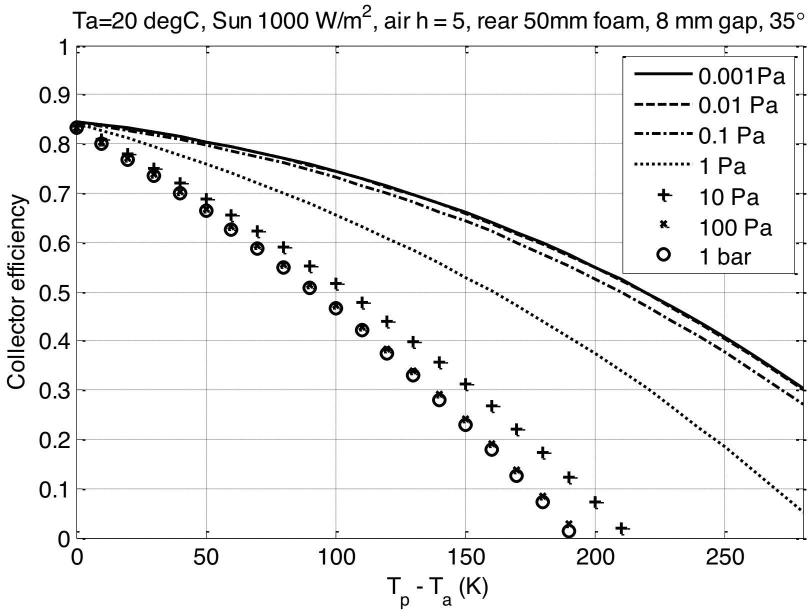

Figure 1. Predicted efficiencies for flat plate collectors as a function of internal air pressure. (Formulae from Duffie&Beckman 1991; conductivity from Beikircher et al 1996).

The vacuum-insulated panel has significantly lower heat loss coefficients than a conventional flat panel

(Figure 2):

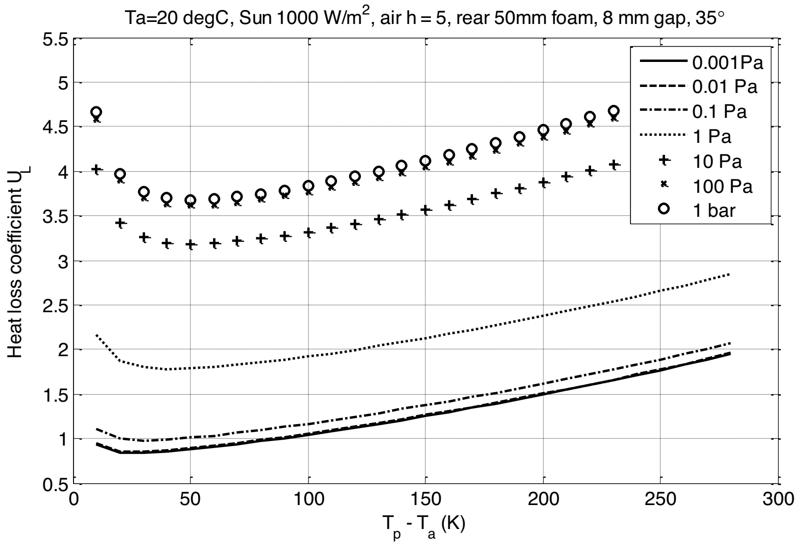

Figure 2. Predictions from Figure 1 expressed as an overall heat loss coefficient.

The second consideration is to maximise the heat recovery factor F_R by minimising the difference between the fluid inlet temperature T_i and the mean plate temperature T_{pm}. To this end one should ideally:

- use a high flow rate, such that the fluid temperature rise is small;
- minimise the spacing between flow channels and use a thick plate made of high conductivity material, such that the plate temperature varies little in the transverse direction;
- minimise the channel hydraulic diameter to provide high heat transfer coefficients

In practice, a designer must choose how much pumping power is affordable for circulating the fluid and then identify an optimum combination of channel diameter and flow rate subject to this constraint.

When deciding how much pumping power is appropriate, one might decide to add a small PV panel (perhaps 1% of the thermal collector area) to provide power for an electric pump. A typical PV panel might generate a peak of 150W/m^2; driving a 50% efficient electric pump, 0.01 m^2 PV cells could meet a peak pumping power requirement of 0.75 W for a 1 m^2 thermal collector. More usefully one might consider the daily average energy: in temperate latitudes a PV panel might average 600Wh/m^2/day, so 0.01 m^2 (again with 50% pump efficiency, neglecting any losses from batteries and electronics) could deliver 0.25W to the fluid over 12 hours of daylight. (The electrical power dissipated through friction of course adds to the apparent heat collected, though the effect is relatively insignificant at these power levels).

One may conclude that pumping power requirements (per square metre of collector) in the range W_p = 0.1 to 1 W/m^2 are of most interest.

3.2 Calculation of pumping power effects

For calculation purposes, the microchannel plate with rectangular channels is considered to be equivalent to a virtual plate of length *L* with circular passages of hydraulic diameter D_h and size to pitch ratio $R = \dfrac{D_h}{P}$

Figure 3. Definition of collector geometry parameters.

One may design a cooling system by first identifying a suitable hydraulic diameter and number of circular holes, then considering the equivalent dimensions and number of passages for the desired non-circular cross-section, which might be rectangular or a hydroformed shape. The circular hole assumption will model the pressure drop and fluid to metal temperature difference correctly provided one uses a friction coefficient f and Nusselt number Nu_{cf} appropriate for the actual passage cross-section.

$R = \frac{2}{\pi}$ is equivalent in terms of flow area and perimeter to square-section passages having *width* = 0.5×*pitch* and has been used for all graphs here.

Assuming laminar flow with friction coefficient $f = \frac{\text{Po}}{\text{Re}}$, it can be shown that the total mass flow rate is

$$\dot{m} = \left(\frac{\rho B D_h^{1.5}}{4}\right)\sqrt{\frac{2\pi R W_p}{\text{Po}\mu}} \qquad \text{(eq.4)}$$

For fully developed laminar flow with a constant heat flux boundary condition, the Nusselt number is expected to be a constant, Nu_{cf} (all conditions plotted here have Re < 2000).

Typically for a square section duct $Nu_{cf} \approx 3.612$ (Kakaç 1987) and $\text{Po} = 14.226$ (Lorenzini 2009); these values are used for all graphs in this paper. Once an initial design has been identified, the calculations can of course easily be repeated with entry length-specific friction factors and Nusselt numbers.

The cooling design aims to minimise the plate heat loss that arises when the plate is “hotter than it needs to be”. The difference between the fluid inlet temperature and the mean plate temperature, $\Delta T = T_{pm} - T_i$, must be minimised for a given pumping power.

ΔT has two components, the mean temperature rise of the fluid along the channels plus the temperature difference driving the convective heat transfer

$$\Delta T = \left(\frac{T_o - T_i}{2}\right) + \Delta T_h = \Delta \overline{T}_f + \Delta T_h \qquad \text{(eq.5)}$$

This unwanted temperature difference can be expressed in terms of fluid properties, pumping power and plate length:

$$\Delta T = S^* \left[\left(\frac{2L}{\rho c} \right) \sqrt{\frac{\text{Po}\mu}{2\pi R W_p}} D_h^{-1.5} + \left(\frac{1}{\pi k Nu_{cf} R} \right) D_h \right] \qquad \text{(eq.6)}$$

One can see that reducing the passage size will reduce ΔT_h (because the heat transfer coefficient increases) but, for a constant W_P, reduce the mass flow rate and increase $\Delta \bar{T}_f$. For any chosen R and W_p there will be an optimum D_h for a given fluid that balances these two effects to minimise ΔT (Figure 4).

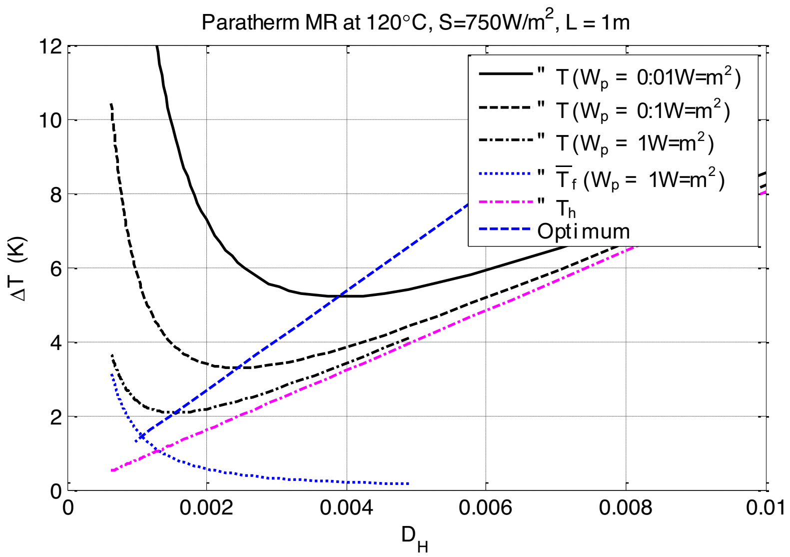

Figure 4. Effect of passage hydraulic diameter on ΔT. The curve $\Delta T\left(W_P = 1\ \text{W/m}^2\right)$ is the sum of the $\Delta \bar{T}_f\left(W_p = 1\ \text{W/m}^2\right)$ and the ΔT_h curves.

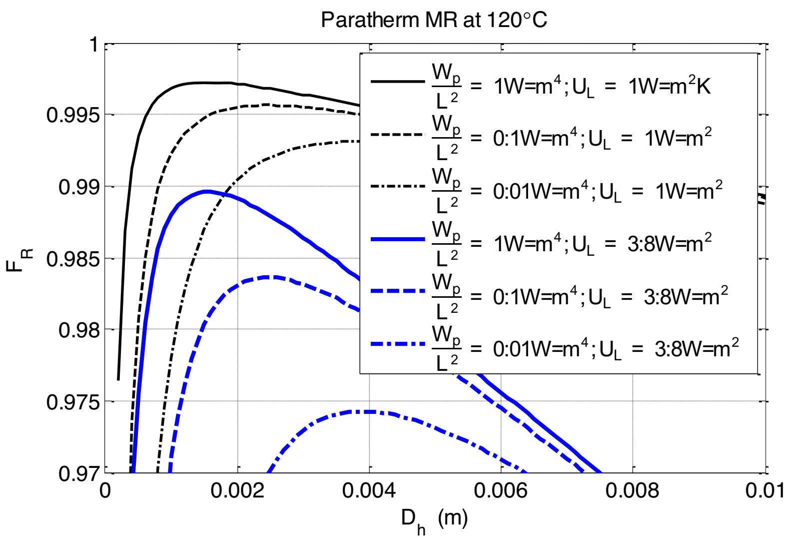

Figure 5. Effect of hydraulic diameter on heat recovery factor F_R.

Figure 5 shows the curves from Figure 4 plotted as heat recovery factor rather than temperature difference; The evacuated collector design typically achieves $U_L = 1\ \text{W/m}^2 K$, considerably lower than a conventional gas-filled flat panel with $U_L = 3.8\ \text{W/m}^2 K$ and this raises the heat recovery factor, at any given mass flow rate.

One can achieve similar ΔT over a range of designs of differing length by selecting $W_p \propto L^2$ (i.e. a constant $\frac{W_P}{L^2}$ value); if the plate length is increased (by itself) or the pumping power is reduced then the optimum passage size will increase. For a given W_p, ΔT can be reduced by choosing a design with small L, for instance a rectangular plate could have the passages running across the plate rather than along it. This is similar to the concept of connecting multiple solar collectors in parallel rather than in series: it maximises the total passage cross-section area as well as minimising the passage length, thereby (for some total mass flow rate) reducing the fluid velocity and pressure drop.

In practice the design of inlet manifolds to achieve uniform flow partition between passages will be easier with large L and small B than vice-versa; conversely a plate with many short passages would have enhanced heat transfer due to entry length affects. Practical considerations such as ease of manufacture or the size of manifolds and pipework necessary to ensure uniform flow distribution between passages may also dictate the use of smaller or larger hydraulic diameters. The temperature curves (Figure 4) have broad minima so this can be accomplished with little loss of performance.

It can also be seen from equation 6 that ΔT decreases as R increases: one should choose a geometry with the largest sensible value for R (many passages, with thin ribs between them).

4. Plate design.

Figure 5 shows that optimal values of hydraulic diameter lie in the range 1.6 to 2.4 mm for pumping powers between 1 and 0.1 W/m^2 when using Paratherm MR in a 1 m long plate; for Tyfocor-LS at 70 and 100°C the equivalent is 1.6 to 3.2 mm. The test plate is relatively small (200 mm long) but it was designed to test design features at a scale useful in a full-size panel rather than being optimised as a small panel in its own right.

One of the aims of this project was to assess the benefits of a microchannel plate in terms of thinness (leading to an architecturally elegant product) with fine passages achieving high heat transfer coefficients at low flow rates. A passage size of 1.6 × 1.6 mm was chosen as being large enough to machine and weld easily and close to the optimum dimensions for a range of fluids and flow rates.

Manifold design

Inlet and outlet manifolds for the micro-channel plate were designed to give a "Z" flow path, with prescribed pressure-drop profiles following Tondeur [2011] to achieve the same flow rate in each channel, Figure 6. The required profiles were NC milled into the ends of the micro-channel plate, Figure 7.

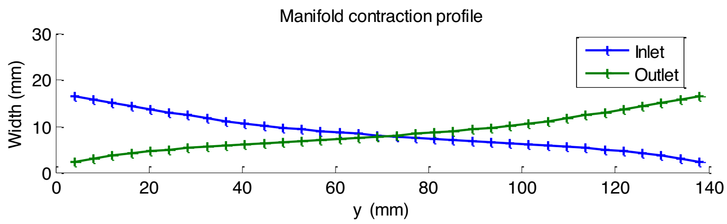

Figure 6. Manifold taper profiles to achieve uniform flow distribution between passages.

The through-holes visible in the image below are for support pins that resist atmospheric pressure and hold the glass panes apart (Henshall et al, 2014). Two plates were constructed (Figure 8): one in stainless steel, one in aluminium, to assess the suitability of laser welding these materials for vacuum solar collectors.

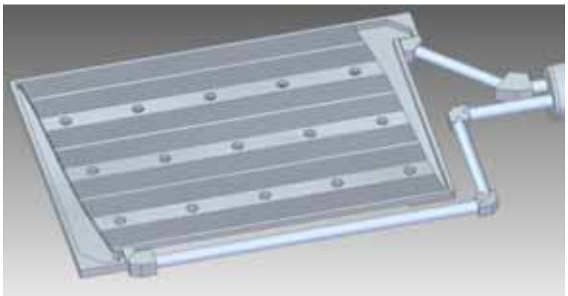

Figure 7: CAD rendering showing plate internal details. The pipe at 45° was required to provide welding access to a manifold joint.

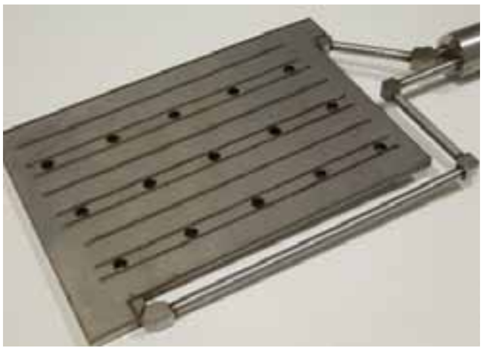

Figure 8. Stainless steel collector after laser welding.

Helium leak testing of the aluminium plate (6082 plate with 1050 lid) revealed numerous leaks from micro-cracks in the welds; the leaks persisted even after re-welding several of the joints. The 304 stainless plate however only had a few leaks and these were easily fixed; it has proved itself vacuum-tight over a period of several days (limited only by out-gassing of the test facility). The welding of these one-off prototypes was performed manually using a pulsed laser but in a production environment with suitable jigs a stainless collector could be rapidly welded using a continuous laser.

Mechanical design

A flat plate collector can be designed with its inlet and outlet pipes either passing through an "edge seal" between the front and back glass covers or with the collector sitting inside a metal tray with a single glass cover. The current plate was designed for the former configuration, with tubes passing through an edge seal.

The collector is typically hotter than its enclosure; possibly much hotter, in a worse case condition with no flow. Selective coatings with low emittance lead to high stagnation temperatures; the use of a vacuum enclosure also raises the stagnation temperature, since then heat can only be lost via radiation and (to a small extent) conduction along the pipework. The need to preserve a high vacuum throughout the service life suggests that thermal stresses at the pipe to enclosure joint should be minimised as far as possible.

It is undesirable for the weight of the plate to be taken by the pipework when the panel is inclined from the horizontal, e.g. on a roof. One solution to this is for any side-forces on the panel to be taken by one of the pillars; the plate can rotate about this point (C in Figure 9), if need be, to accommodate non-uniform thermal expansion.

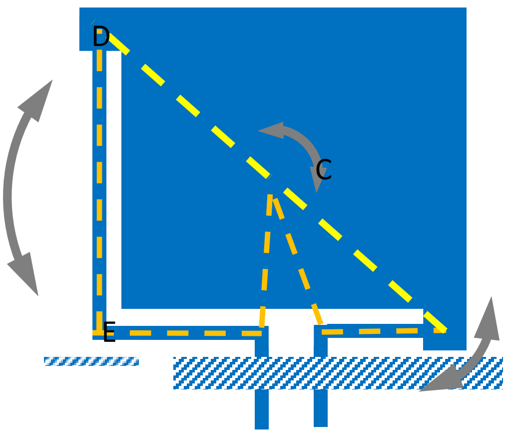

Figure 9. Conceptual design of inlet and outlet tubes as a linkage to provide flexibility for thermal expansion.

The pipework provides a little flexibility.

(a) The flow and return pipes close together, such that isothermal expansion of an unrestrained collector assembly would lead to only a small increase in pipe spacing and

(b) The pipework forms a linkage with at least four elements (enclosure, pipe, pipe, plate, Figure 9) so that expansion can be accommodated via transverse flexure of small-bore tubing instead of axial strain:

- ABC is a 3-bar linkage (plate is free to rotate about central pin C due to expansion of AB or BC; the pipes can flex).
- CDEF is a 4-bar linkage (allows the plate to rotate slightly).

5. Conclusions

A method has been developed for identifying an optimum passage size for a flat plate collector based on a pumping power limit and for assessing the reduction in heat output, for any given passage size and pumping power, relative to an ideal plate help uniformly at the fluid inlet temperature. The intake and outlet manifolds were designed with a prescribed cross-section to achieve uniform flow distribution between passages.

One feature essential for a vacuum-insulated flat plate is the provision of through holes for the glass support pillars. Two prototype plates were manufactured, complete with an array of through holes, to demonstrate the possibility of welding such features; the stainless plate had much better vacuum sealing than the aluminium one. Experimental testing is in progress and will provide a datum standard against which further plate designs can be compared and evaluated.

6. Acknowledgements

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7. References

Benz, N. and T. Beikircher, T., 1999. High Efficiency Evacuated Flat-Plate Solar Collector For Process Steam Production, Solar Energy 65 (1999) No. 2 pp 111-11

Beikircher, T., Goldemund, G. and Benz, N., 1996. Gas Heat Conduction In An Evacuated Tube Solar Collector, Solar Energy 58 (1996) no 4-6 pp213-217

Bracamonte, J and Baritto, M., 2013. Optimal aspect ratios for non-isothermal flat plate solar collectors for air heating, Solar Energy 97 (2013) 605-613

Chen, Z., Furbo, S., Perers, B., Fan, J. and Andersen, E., 2012. Efficiencies of flat plate solar collectors at different flow rates Energy Procedia 30 (2012) 65 – 7

Duffie, J.A. and Beckman, A., 1991. Solar Engineering of Thermal Processes, Wiley.

Eisenmann W., Vajen, K. And Ackermann, H., 2004. On the correlations between collector efficiency factor and material content of parallel flow flat-plate solar collectors, Solar Energy 76 (2004) 381-387

Farahat, S., Sarhaddi, F. and Ajam, H., 2009. Exergetic optimisation of flat plate solar collectors, Renewable Energy 34 (2009) 1169-1174

Fiaschi, D and Bertolli, A., 2012. Design and exergy analysis of solar roofs: A viable solution with esthetic appeal to collect solar heat, Renewable Energy 46 (2012) 60-71

Hegazy A.A., 1996. Optimization of flow-channel depth for conventional flat plate solar air heaters, Renewable Energy vol 7, No. 1 (1996) pp15-21

Hegazy A.A., 1999. Optimizing the thermohydraulic performance of plate solar air heaters operating with variable pumping power, Renewable Energy 18 (1999) 283-304

Henshall, P., Moss, R., Arya, F., Eames, P., Shire, S. and Hyde, T, 2014. An evacuated enclosure design for solar thermal energy applications, Grand Renewable Energy Conference 2014.

Jones G.F. 1987. Consideration of the heat removal factor for liquid-cooled flat-plate collectors, Solar Energy vol 38, No. 6, pp 455-458, 1987

Kakaç, S. Shah, R.K. and Aung, W., 1987. Handbook of single-phase convective heat transfer, Wiley.

Lorenzini, M. and Morini, G.L., 2009. Poiseuille and Nusselt numbers for laminar flow in microchannels with rounded corners, 2nd Micro and Nano Flows Conference, London, September 2009 http://bura.brunel.ac.uk/bitstream/2438/6936/1/MNF2009.pdf

Mansour, M.K., 2013. Thermal analysis of novel mini channel-based solar flat plate collector, Energy 60 (2013) 333-343

Mintza Do Ango, Medale, M. and Abid, C., 2013. Optimization of the design of a polymer flat plate solar collector, Solar Energy 87 (2013) 64-75

Motte, F., Notton, G., Cristofari, C and Canaletti, J-L, 2013. Design and modelling of a new patented thermal solar collector with high building integration, Applied Energy 102 (2013) 631-639

Sharma, N. and Diaz, G., 2011. Performance model of a novel evacuated-tube solar collector based on minichannels, Solar Energy 85 (2011) 881-890

Tondeur, D, Fan, Y., Commenge, J-M and Luo, L., 2011. Uniform flows in rectangular lattice networks, Chemical Engineering Science 66 (2011) 5301-5312