DESIGNING A LOW COST SOLAR COLLECTOR SYSTEM FOR PROCESS INDUSTRY APPLICATIONS

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ABSTRACT

New Zealand is a large producer and processor of primary products and has a climate with high levels of solar radiation. However, the use of solar energy for heating and cooling in the processing industries has received limited attention.

For this study, the design of a low cost solar collector is analysed and discussed. Furthermore, the methods for integrating the collector into water heating and cooling systems in a hypothetical processing environment are examined. An F-Chart analysis is used to simulate the performance of large-area arrays of the solar collector and to determine its potential contribution to heating and cooling loads.

The study shows that for a storage-based system, the contribution of solar energy is determined mainly by the collector area to storage volume ratio. It is suggested that this low cost collector could make a significant contribution to energy use in processing plants and may be an attractive future technology.

INTRODUCTION

New Zealand as a large producer of primary products uses significant amounts of energy in their processing. For example, the New Zealand dairy industry produces over 14 billion litres of milk annually (LIC, 2006). As this is far in excess of local demand, the majority of it is processed for export markets. To process such large amounts of dairy products, it is necessary to supply a large amount of energy.

To illustrate this point; Lovell-Smith and Vickers (1983) found that the production of whole milk powder used in excess of 14 GJ/t. Similarly, Vickers and Shannon (1977) found that significant amounts of energy were used for generating hot water to be used in the production of cheese and for cleaning in place (CIP) operations. A number of technologies have been proposed for reducing energy use in the dairy processing industry, both in New Zealand and internationally. Benz et. al. (1998) and Benz et. al. (1999) examined the suitability of solar thermal processing heating systems to some German food processing plants.

In Benz et. al. studies, they found that solar collectors were suited to heating applications in a milk spray drying factory. In their 1999 study they found that solar thermal systems could supply heating, over a 20 year period, at a cost of $100 US/MWh. Furthermore, they noted that under a favourable climate, heat costs could be halved and performance doubled.

In a recent study of energy use in the European dairy industry Ramirez et. al. (2006) noted that CIP accounts for approximately 70% of the energy use in evaporators and up

to 26% of the energy used in dryers. They also note that these operations typically use temperatures in the range of 65°C to 75°C.

Similarly, Schnitzer et. al. (2007) found that in the Austrian dairy industry over 80% of the heating demand was for temperatures in the range from 60°C to 80°C, making the use of solar energy ideally suited to heating in dairy processing plants. Furthermore they note that this temperature is suitable for operations such as washing water in cheese production, preheating of cheese milk, outside cleaning, pasteurization, whey conditioning and cleaning in place (CIP) operations. All of which are commonly used in the New Zealand dairy industry.

New Zealand presents an ideal environment for the utilisation of solar energy, especially when compared to some northern hemisphere locations. In New Zealand approximately 30% of all dairy farms, and a large number of dairy processing plants, are located in the Waikato and surrounding region, to the south of Auckland (LIC, 2006).

In addition to providing useful heating, solar radiation aids in the production of feed for dairy herds. As such, with the onset of calving in early spring, and increasing levels of solar radiation, there is also an increase in milk production. Vickers and Shannon (1977) and Benseman (1986) showed that in New Zealand, this peak in production occurred in mid to late spring before gradually reducing over summer. In Figure 1 the trends in the production of milk is compared with the solar radiation levels for a June to May year. From this it can be seen that milk production levels and solar radiation exhibit a degree of correlation.
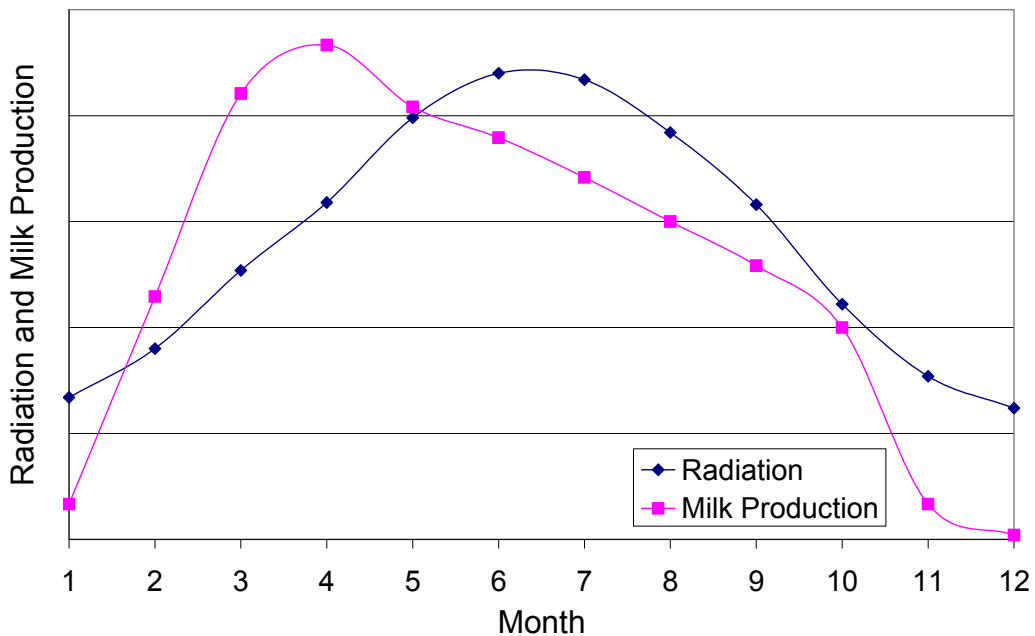

Fig. 1: Qualitative relationship between milk production and solar radiation levels for a typical milk production season

Given the relationship between the production of milk and the increase in solar radiation combined with the energy used in the New Zealand dairy processing industry, there is obviously significant scope to utilise solar energy in the processing of dairy products.

However, one of the possible impediments to the use of large scale solar thermal use in the process industries is the initial cost of such systems, and therefore the return on investment time. In a report by EECA (2004), it was noted that one of the largest impediments to the uptake of solar waters, domestically, was the initial cost and it could easily be assumed that this would apply to industry also.

As such this work will look at the design and performance of a low-cost solar collector for potential application in the New Zealand processing plants.

COLLECTOR DESIGN

Unlike many commercially available collectors the collectors in this study were not constructed from finned copper tubes Instead the collectors were fabricated from two colour coated mild steel sheets that were folded to form a rectangular cross section tube. Additionally mineral wool insulation was placed behind the absorber sheet and a low-iron glass cover above the collector as shown in Figure 2. The nature of the design has resulted in an approximate 50% reduction in the cost of collectors relative to existing technologies.
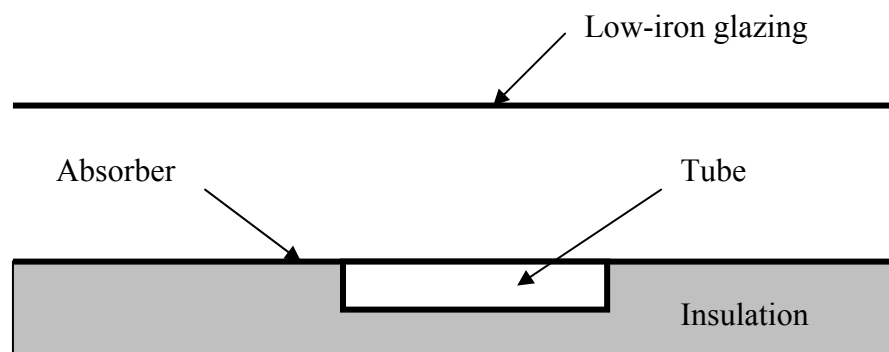
Fig. 2: Partial cross-section of collector

Although the fabrication of finned copper tube style collectors is well understood, the unconventional design of the collector, and the desire for it to be made from relatively low-cost pre-coated steel, presented a number of challenges. The main challenge is due to the fact that the material is galvanised and coated in paint. As such the material cannot be welded without removing both these coatings. In order to circumvent this issue, it was decided to bond the secondary folded sheet to the coloured absorber with a high temperature Silicone adhesive.

Due to the batch production nature of the prototype, the secondary sheet was folded using a brake press, holes were drilled to allow fluid into the underside of the rectangular tube, nipples were soldered to the rear surface around these holes to allow a manifold to be attached, the ends were sealed and the top absorber sheet was bonded

into place. Finally, a removable low-iron-glass cover was placed over the collector to prevent convection losses.

MEASUREMENT OF ABSORBER OPTICAL PROPERTIES

When examining the performance of either glazed flat plate solar collectors, it is important that we are able to characterise their spectral absorption characteristics. From a theoretical perspective the thermal efficiency of a flat plate solar collector can be represented by a relationship between the collectors heat removal factor (F_r), the collector heat loss coefficient (U_L), the inlet (T_i) and ambient temperatures (T_a), solar radiation ($G"$) and the collector transmittance-absorptance product ($\tau\alpha$) as shown in Equation. 1.

$$\eta = F_r(\tau\alpha) - F_r U_L\left(\frac{T_{in} - T_a}{G"}\right) \quad (1)$$

Of these parameters, the transmittance-absorptance product is the only one that is based solely on a physical property of the collector materials. The absorptance provides a measurement of the optical properties of the radiation absorbing surface, in this case the coloured absorber, while the transmission component measures the portion of the radiation transmitted by any glazing layer. Therefore, in order to understand the optical characteristics of the coloured collectors it was decided to determine their absorptance properties over the solar radiation spectrum.

To determine the absorption of the colour coated mild steel the diffuse reflectance (ρ) of a grey and black sample were measured at 20 nm wavelength intervals between 300 nm to 2500 nm using a spectrophotometer and a 6° integrating sphere at Industrial Research Limited (Wellington, NZ). Based on the reflectance measurements it is possible to determine the absorptance (α) component using Equation 2, as it can be assumed that there is no "transmittance" component in air (Duffie and Beckman, 2006).

$$\alpha = 1 - \rho \quad (2)$$

By integrating the absorptance derived from the measurements of the reflectance it was found that the black painted steel had relatively constant reflectance characteristics across the measured wavelengths of the Air Mass 1.5 (AM1.5) solar spectrum. However the grey sample was slightly more sensitive to wavelength.

Having determined the absorption characteristics of the absorber, to determine the transmittance-absoptance product of a glazed coloured collector it is necessary to substitute the measured spectral absorption characteristics and the low iron glass transmittance characteristics of Dietz (1954) into Equation. 3.

$$\tau\alpha = \frac{\int_{\lambda_1}^{\lambda_2} \tau_\lambda \alpha_\lambda I_{\lambda,i} d\lambda}{\int_{\lambda_1}^{\lambda_2} I_{\lambda,i} d\lambda} \quad (3)$$

By integrating these values over the AM1.5 spectrum it is found that the transmittance-absorptance product for glazed solar collectors. Based on this method, the transmittance-absorptance values determined for the two collectors are given in Table 1.

Table 1: Transmittance-absorptance product of glazed collectors

Collector	Transmittance-absorptance product
Black	0.87
Grey	0.81

THEORETICAL COLLECTOR PERFORMANCE

Having determined the transmittance-absorptance product for the glazed collectors it is possible to determine their theoretical performance using a one-dimensional steady state thermal model based on the Hottel-Whillier-Bliss equations presented by Duffie and Beckman (2006).

Under these conditions the useful heat gain can calculated using Equation 4.

$$Q = AF_R \left[(\tau\alpha).G'' - U_L(T_{in} - T_a) \right] \text{ (4)}$$

Where the useful heat gain (Q) is given by a relationship between the collector area (A), the heat removal efficiency factor (F_R), the transmittance-absorptance product of the coloured collector ($\tau\alpha$), the solar radiation (G"), the collector heat loss coefficient (U_L) and the temperature difference between the collector inlet temperature (T_{in}) and the ambient temperature (T_a).

The heat removal efficiency factor (F_R) can be derived from Equation 5, which accounts for the mass flow rate in the collector (m) and the specific heat of the collector fluid (Cp).

$$F_R = \frac{mC_P}{AU_{loss}} \left[1 - e^{-\frac{AU_{loss}F'}{mC_P}} \right] \text{ (5)}$$

To determine the heat removal efficiency factor it is necessary to calculate a value for the corrected fin efficiency (F'). This is done by first calculating the fin efficiency (F) using Equation 6. This determines the efficiency of the finned area between adjacent tubes and takes into account the influence of the tube pitch (W) and the tube width (d). Furthermore, the coefficient (M) accounts for the thermal conductivity of the absorber and is derived from Equation 7.

$$F = \frac{\tanh\left(M\, \frac{W-d}{2} \right)}{\left(M\, \frac{W-d}{2} \right)} \text{ (6)}$$

$$M = \sqrt{\frac{U_{loss}}{K_{abs}L_{abs}}} \quad (7)$$

Therefore, the corrected fin efficiency (F') can be calculated using Equation 8, noting that there is not bond resistance term as would be found in the analysis of a finned tube analysis and where the overall heat loss coefficient (U_L) of the collector is the summation of the collector's edge, bottom and top losses. It is taken that the bottom loss coefficient is given by the inverse of the insulations R-value (i.e. K_b/L_b) and Equation 9 gives the edge losses, where p is the collector perimeter and t is the absorber thickness.

$$F' = \frac{\dfrac{1}{U_L}}{W\left[\dfrac{1}{U_{loss}(d + (W-d)F)}\right] + \dfrac{1}{\pi d h_{fluid}}} \quad (8)$$

$$U_{edge} = \frac{K_{edge}pt}{L_{edge}A_{collector}} \quad (9)$$

The top loss coefficient is a function of both radiation and wind and can be calculated using Klein's empirical equation (Equation 10) (Duffie and Beckman, 2006).

$$U_{top} = \left\{\frac{N}{\dfrac{c}{T_{pm}}\left(\dfrac{T_{pm}-T_a}{N-f}\right)^e} + \frac{1}{h_w}\right\}^{-1} + \frac{\sigma\left(T_{pm}+T_a\right)\left(T_{pm}^2+T_a^2\right)}{\left(\varepsilon_p + 0.00591Nh_w\right)^{-1} + \dfrac{2N+f-1+0.133\varepsilon_p}{\varepsilon_g} - N} \quad (10)$$

Where:

$$c = (520 - 0.000051\beta^2) \qquad f = (1 + 0.089h_w - 0.1166h_w\varepsilon_p)(1 + 0.07866N)$$

$$e = 0.430(1 - \frac{100}{T_{pm}}) \qquad T_{pm} = T_{in} + \frac{Q/A_{collector}}{F_R U_{loss}}\left(1 - F_R\right)$$

and β is the collector mounting, σ is the Stefan-Boltzmann constant, N is the number of covers or glazing layers, εg is the emittance of the glazing, εp is the emittance of the plate and h_w is the convection heat transfer due to the wind.

From these equations it is then possible to calculate the useful heat gain from the solar collector. By taking the ratio of the useful heat gain to the total radiation falling on the collector area (Q/AG") we can subsequently determine the theoretical efficiency as given in Equation 1.

Therefore, by substituting the design parameters listed in Table 2 into the equations listed above, in combination with the measured transmittance-absorptance products for

glazed collectors, it is possible to determine their theoretical thermal efficiency. The predicted theoretical thermal efficiency for each of the collectors is shown in Figure 3.

Table 2: Design parameters for solar collectors

| Parameter	Symbol	Value	Unit
Number of covers	N	1	
Emittance of plate	ε_p	0.95	
Emittance of cover	ε_c	0.88	
Number of tubes	n	2	
Collector Length	L	1.96	m
Collector Breadth	b	0.5	m
Collector Area	A	0.98	m^2
Absorber thickness	t	0.5	mm
Tube Hydraulic Diameter	d_h	9	mm
Tube Spacing	W	0.2	m
Tube Width	d	50	mm
Insulation Conductivity	k	0.045	W/mK
Back Insulation Thickness	L_b	0.1	m
Edge Insulation Thickness	L_{edge}	0.025	m
Absorber Conductivity	k_{abs}	50	W/mK
Mounting Angle	β	37	degrees
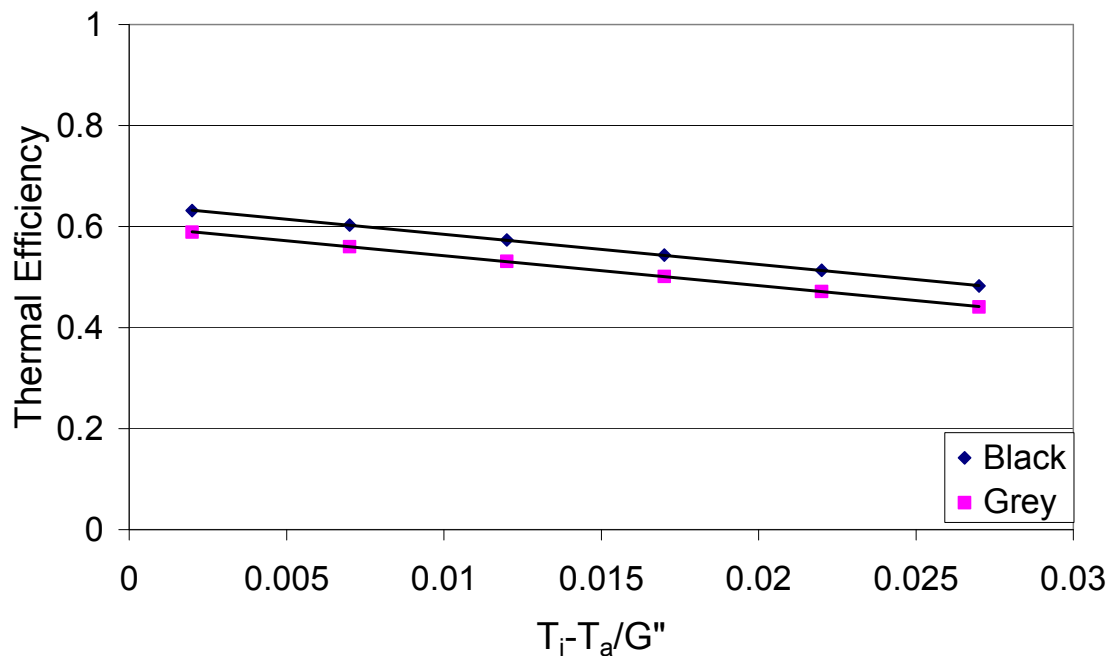
Fig. 3: Theoretical efficiency of glazed solar collectors

EXPERIMENTAL COLLECTOR PERFORMANCE ANALYSIS

Although Figure 3 illustrates the potential performance of the glazed collectors, good practice necessitates validating the model experimentally. As such a grey prototype collector was constructed for testing. Although there are a number of potential methods for determining the thermal efficiency of solar water heaters; for this study a steady state outdoor thermal test setup similar to that recommended in AS/NZS 2535.1 (1999) was used, as shown in Figure 4.
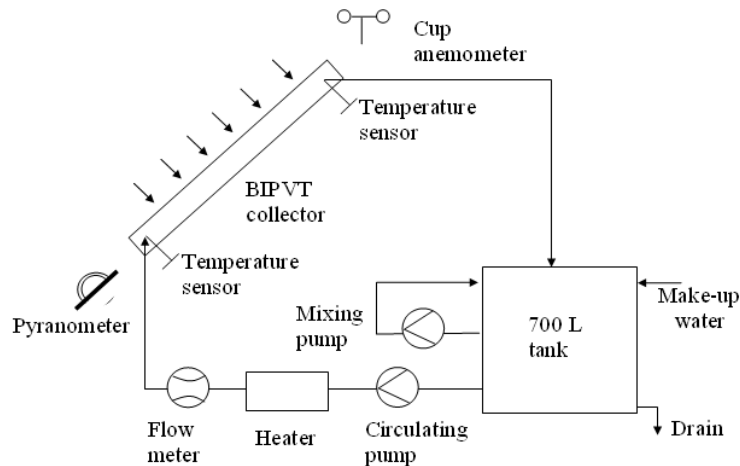
Fig, 4: Collector test system

In order to test the prototype collectors, an unimpeded north facing test location was found on the University of Waikato library roof. The global solar radiation incident on the collectors' surface was measured using a calibrated WMO First Class pyranometer mounted inline with the collector at an angle equal to the local latitude (37 degrees).

Calibrated T-type thermocouples ($\pm0.3K$) were used to measure the inlet and outlet temperatures to the collector, and the ambient air temperature. A cup anemometer mounted adjacent to the test stand was used to monitor the wind speed. The flow rate through the collector was set at a constant rate and monitored throughout the testing periods by measuring the time taken for a known mass of water to pass through the collector.

Additionally, an instantaneous electric water heater with temperature control was mounted on the inlet side of the collector to provide a controllable inlet water temperature. The outlet from the collector was returned to a 700-litre water tank where it was well mixed to ensure that the heater did not encounter large instantaneous temperature variations.

A prerequisite to accurately determining the performance of the collector is to conduct a number of outdoor tests under a range of ambient conditions and allow it to reach steady state for each condition. Subsequently, when analysing the collectors, the instantaneous collector efficiency can be reached directly from the experimental results by taking the ratio of heat transfer in the collector to the product of the collector area and the global solar irradiance.

From the experimental data collected during the testing it was possible to derive the efficiency equation of the both coloured collectors using a linear least squares regression analysis. The experimental data yields an equation that describes the grey collector efficiency as shown in Equation 11.

$$\eta = 0.65 - 10.4 \frac{T_i - T_a}{G''} \quad (11)$$

Although this is one common way of presenting the efficiency of the collector it can be better understood from an inspection of Figures 4, where the theoretically predicted efficiency of the collector is also shown. From this it can be seen that the theoretical prediction corresponds fairly well with the experimental data.
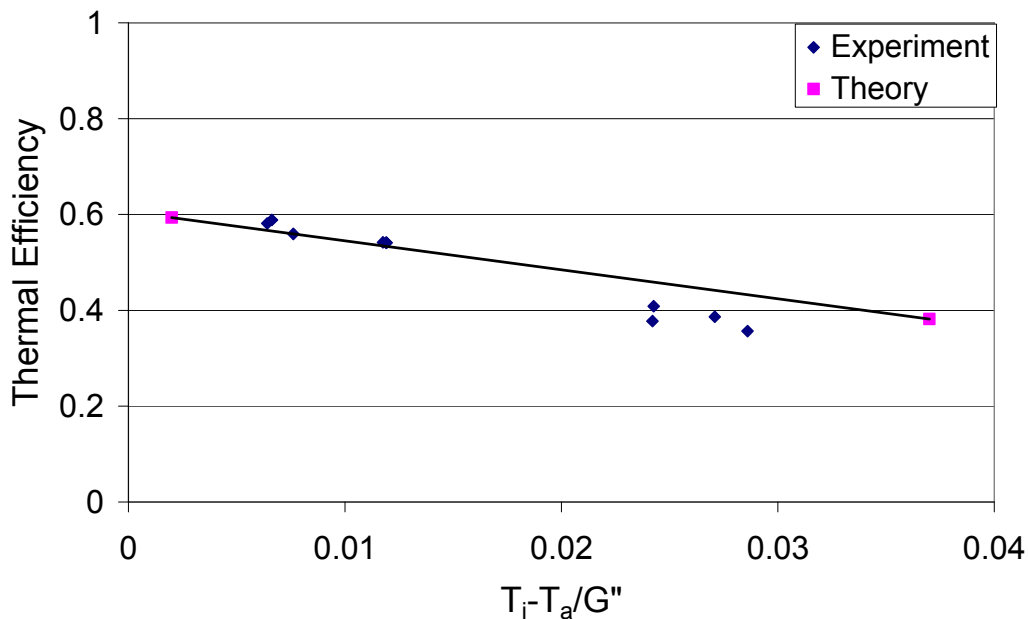
INTEGRATING COLLECTORS WITH PROCESSING OPERATIONS

Perhaps the most common difficulty in the use of solar energy in a continuous operation is where and how to integrate it. One common method used in the determination of where heat sources should be integrated into a process is the pinch method.

The pinch method uses composite heating and cooling curves as a visual representation of heat and temperature demand in process industries, it shows the point (ie. the "pinch") above which it is necessary for heat to be added and below which cooling is required.

The pinch method was used by Schnitzer et. al. (2007) to show that in a typical cheese production line the "pinch" point occurred at approximately 20°C. As such it confirms that the use of solar energy is ideally suited for application in the dairy environment, as solar heating and cooling systems would be able to deliver energy both above and below this point.

In order to integrate this energy into the system it is possible to use both direct and indirect integration. In Figures 5 and 6 two means for the direct integration of solar

heating are shown. The first of these shows the solar array effectively acting as a inline heat exchanger preheating the water, or heating fluid, returning from a process before entering the boiler or heat source. In the second the flow can be diverted such that instead of using the boiler or heat source, the heat can be supplied by the solar array.
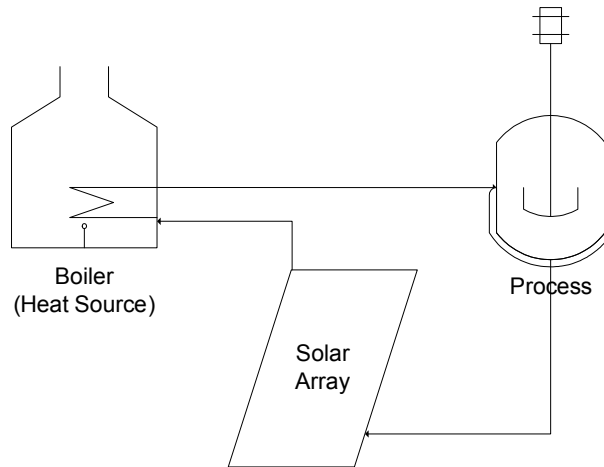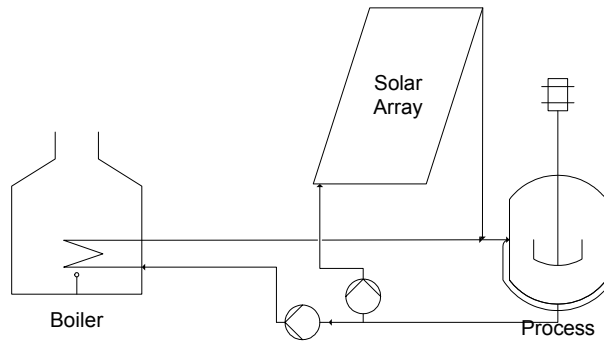
Fig. 5: Direct integration of heat from a solar array

Fig. 6: Direct integration of solar heat into a process

There are a number of arguments for and against using a direct integration as shown in Figures 5 and 6. Typically the setup cost for these systems tends to be relatively low; however they do require continuous control and by their nature limit the size of the solar array to ensure that they do not provide energy at a higher temperature than is required by the process. Furthermore, these systems will only function during the day.

These shortcomings can be overcome by use of an indirect, storage based system as shown in Figure 7. By installing an intermediate storage tank, heat can be added to the tank by the solar array and used as required. As such the system does not need to be continuously controlled as with a direct system and furthermore, by adding a storage vessel it is possible for heat to be stored and used during periods of low or no solar radiation. The main drawback of an indirect system is that they tend to have a higher initial cost and also a longer payback time.
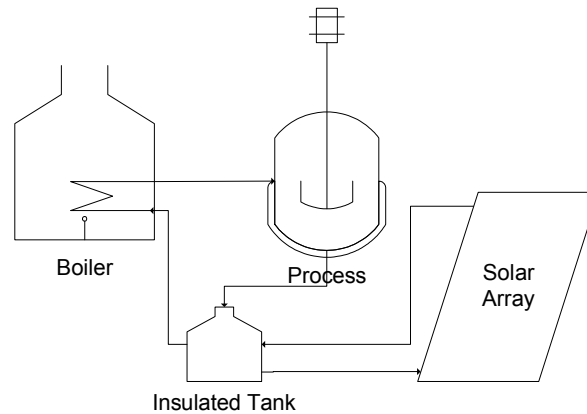
Fig. 7: Indirect heat integration from a solar array utilising a storage system

After examining the options for integrating solar energy into a processing operation, it appeared that given the continuous nature of New Zealand's processing industries, and the dairy industry in particular, that the use of a storage based solution offered the best solution. This would allow solar energy to be utilised with minimal disturbance to existing systems, would require little control and would be able to be used irrespective of prevailing solar conditions. As such a simulation study was conducted to determine the performance of large area arrays of solar collectors coupled with thermal storage tanks for use in the industry, using the Waikato region as a case study.

PERFORMANCE OF A LARGE SCALE INDIRECT SOLAR PROCESS HEATER

Having demonstrated and validated the design of the coloured solar collectors, it was decided to examine the fraction of a typical domestic water-heating load that could be provided by the various theoretical coloured collectors. Therefore, an F-chart (Duffie and Beckman, 2006) was constructed for the operation of the collectors in Hamilton based on weather data was taken from NIWA (2007). Although not as "in-depth" as a full annual transient analysis, such as could be performed by a program such as TRNSYS, the F-chart has been shown to provide good prediction of annual solar fractions.

For the purposes of this study three solar heating systems were modelled, in the first scenario it was assumed that the solar heating system was coupled to a water tank with a volume of $10m^3$, in the second a tank volume of $25 m^3$ and in the third a tank of $100 m^3$. Additionally, it was assumed that the water in each of the tanks would be heated from 40°C to 80°C over the period of a day. This would make it suitable for the applications discussed by Schnitzer et. al. (2007) and be typical of the daily volumes of hot water used in small, medium and large dairy processing plants (CRES, 2008)

Based on the assumptions for the solar collector system, it would be necessary to produce and store 1668 MJ, 4170MJ and 16680 MJ respectively. In each case it was assumed that the collectors were in a clear north facing location and mounted at an angle equal to the locations latitude, approximately 38 degrees for Hamilton.

As a general rule, solar water heating systems require between 50 and 100 litres of storage volume per square meter of collector area. As such for each tank volume, a collector array of approximately 50 L/m^2, 75 L/m^2 and 100 L/m^2 were modelled. Thus for this study, arrays of between 100m^2 and 2000m^2 were modelled, however larger arrays are obviously possible. For each scenario, calculations were based on the gross absorber area using the theoretical efficiency equation of the black collector, given by Equation 12.

$$\eta = 0.65 - 6.0\frac{T_i - T_a}{G''} \quad (12)$$

From the analysis it was found that the solar fraction provided by the solar heating and cooling system was determined solely by the array area to tank volume ratio. As would be expected the greatest solar fraction is achieved at a ratio of 50 L/m^2 for all tank volumes for the heating systems.

In Figure 8 it can be seen that as the volume to collector area ratio increases, the solar fraction decreases, thus meaning that heating is occurring at lower temperatures. This is favourable for flat plates because at lower heating temperatures their efficiency is higher relative to their heat loss.
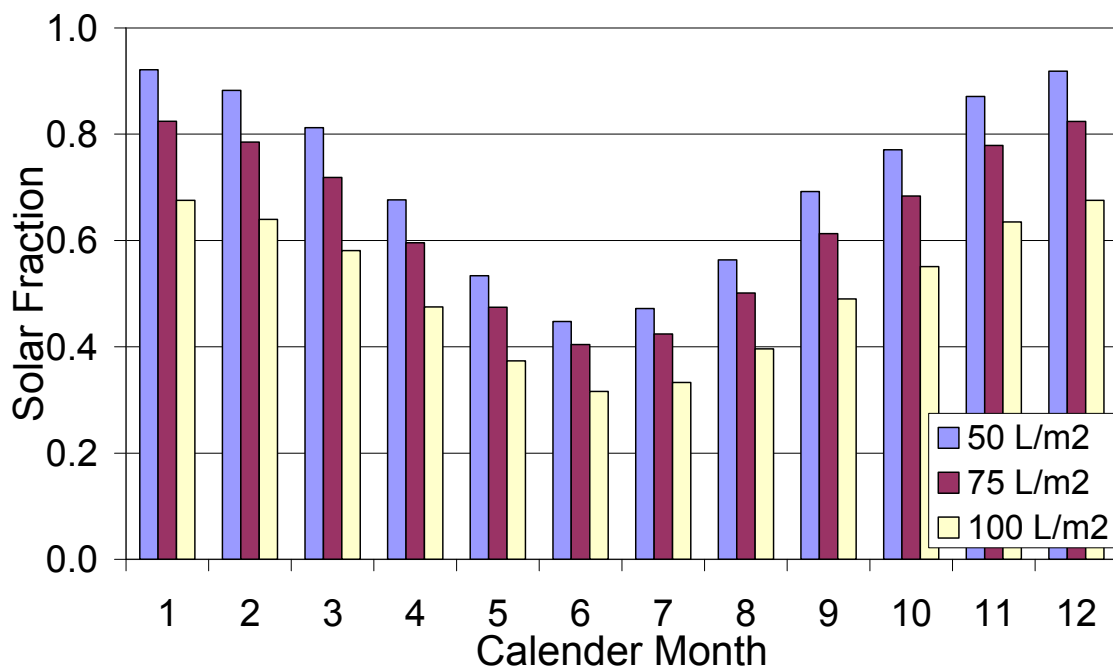

Fig. 8: Solar fraction for process heating systems based on storage volume to collector area ratio

POSSIBLE ENERGY SAVINGS IN A LARGE DAIRY PROCESSING PLANT

To further highlight the advantages of using a solar heating system, it was decided to examine the magnitude of the energy produced by the four collector arrays in a large processing environment.

In a typical large dairy factory the hot water consumption is approximately 120-150 m³ per day, with cleaning operations occurring at up to 80°C (CRES, 2008). As such it was decided to model the ability of an array of 2000m² coupled to a tank with a volume of 100m³ to provide a load of 16680 MJ, or 100m³ of 80°C water per day. This water would be suitable for cleaning operations at the end of a day's production, as suggested by Worley Consultants Ltd (1983).

In Figure 9 it can be seen that during the summer months, the solar heating system is able to meet the majority of the heating load. However, for early spring and autumn, it would be necessary to rely on auxiliary heat from a supplementary boiler.
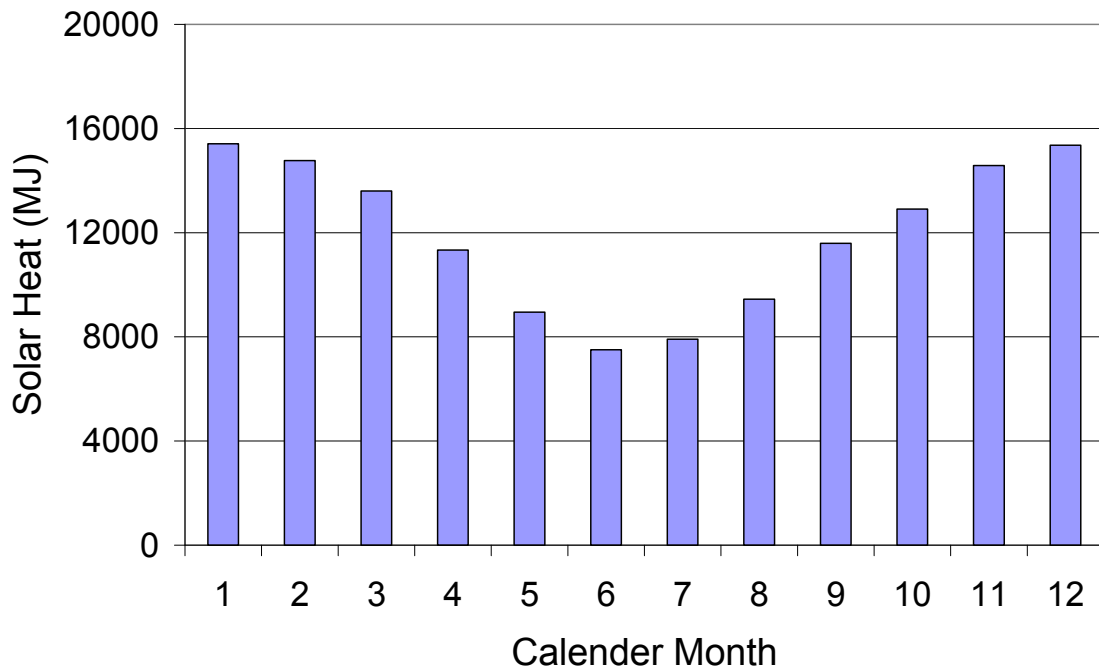

Fig. 9: Possible solar heat production for a large dairy processing plant

(2000m² array and 100m³ store)

As an alternative to a supplementary heat source, it may be possible to increase the size of the solar array. However, it should be considered that for this example, the heating load has been assumed to be constant over the year. Given the seasonal nature of the dairy industry, it is likely that this load would vary with production levels and so the array may meet the demand outside the conditions used here.

ECONOMIC BENEFITS OF SOLAR HEATING IN A LARGE DAIRY PROCESSING PLANT

Previously it was shown that solar heating could make useful contributions to a hypothetical heating load in a large dairy processing plant. Furthermore, it has been noted that the cost of this collector is relatively inexpensive in comparison to existing collector technologies. However, to illustrate the benefits of the low cost system, the cost per unit of energy should also be considered.

Taking the preceding example of an array of 2000m^2 coupled to a tank with a volume of 100m^3 to provide a load of 16680 MJ, or 100m^3 of 80°C water per day. Bura (2007) suggested that a system of this nature would sell for approximately \$150/m^2; therefore the cost of the array would be \$300000.

From the F-Chart analysis it was found that the array would produce approximately 4000GJ of thermal energy per annum. Assuming a 10 year life for the array, and that all the thermal energy from it was utilised, the cost of the energy produced by it would be less than \$1/GJ.

CONCLUSIONS

In this study, the design of a low cost solar collector was examined. Using a combination of theoretical modelling and experimental testing the performance of the collector was quantified.

Further, it was shown that the collector had potential to make significant contributions to the heating loads in a processing environment through the use of an indirect storage based heating system.

Finally, it was shown that by reducing the cost of the collectors in the array, that the cost of energy produced by the collector was relatively inexpensive over its life.

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