OPTIMIZING HEAT EXCHANGER FLOW RATES IN LOW FLOW SYSTEMS

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ABSTRACT

There has been some debate between researchers as to whether optimum flow rates exist in a solar domestic hot water system utilizing a heat exchanger between the collector and the storage tank.

It was found that there exists thermal optimum or at least economical optimum flow rates since low flow rates incur less hydraulic costs. Peak performance was always found to occur when the heat exchanger tank-side flow rate was approximately equal to the average load flow rate. For low collector-side flow rates a small deviation from the optimum flow rate can dramatically effect system performance. However, system performance is insensitive to flow rate for high collector-side flow rates.

Antifreeze solutions have temperature dependent properties such as density, viscosity and specific heat. The effect of large temperature dependent property variations experienced by ethylene glycol and propylene glycol affect the optimum flow rate through the collector-side of the heat exchanger.

1. INTRODUCTION

Freeze protection is required in many climates. The most common form of freeze protection is the use of a closed loop system with a heat exchanger. The heat exchanger allows the circulation of 'freeze resistant' glycol on the collector side of the heat exchanger. The heat exchanger could also allow higher circulation through the collector side therefore promoting the collector heat removal factor without directly affecting tank stratification. Unfortunately, the use of a heat exchanger incorporates an inherent performance and economic penalty.

There seems to be a disagreement as to whether optimum flow rates exist for both sides of the heat exchanger. Hollands (1992) argues that there is an optimum if the *UA* of the heat exchanger is held conceptually fixed. He argues that the difference obtained by Fanney and Klein (1988), who concluded that there were no optimum flow rates for a system, is due to using a fixed heat exchanger whose overall conductance, *UA*, is a strongly increasing function of flow rate. Hollands states that a person skilled in heat exchanger design can design a suitable heat exchanger of any specified *UA* once the flow rates have been given.

Hollands ascertains that if the collector flow rate is kept at its optimum value for the current tank flow rate, then the simulations performed in the past on systems without heat exchangers are directly applicable to systems with heat exchangers.

Fanney and Klein (1988) state that significant improvement of SDHW system performance was not observed by reducing the flow rate on the storage side of the heat exchanger. The experiments performed consisted of a 50% by weight ethylene glycol mixture circulated through the collector loop at a flow rate of 0.0151 kg/s-m^2. Flow rates of 0.020 kg/s-m^2 and 0.0025 kg/s-m^2 were circulated through the storage side of the loop. It was found that the system with the lower tank flow rate required almost 7% more auxiliary energy. However, it is difficult to determine whether these tests were sufficient to determine an optimum with just one collector flow rate. The decrease in performance for the lower flow rate is explained by a significant decrease in the heat exchanger *UA* resulting from the lower flow rate. The heat exchanger penalty was found to more than offset the improved stratification within the storage tank.

2. FLOW OPTIMIZATION WITH CONSTANT *NTU*

TRNSYS (1996) simulations were performed in order to determine the optimal flow rates on either side of the heat exchanger various. All simulations were performed with a collector area of 3.185 m^2, tank volume of 0.4 m^3 and average load of 0.0035 kg/s-m^2. In this section, simulations were performed independent of the heat exchanger geometry. A modified *NTU* was defined as $UA/\dot{m}Cp_{collector}$ regardless of whether or not $\dot{m}Cp_{collector}$ was the minimum capacitance rate in the collector-tank heat exchanger. Simulations were performed for different ratios of $\dot{m}Cp_{collector}$ to $\dot{m}Cp_{tank}$ ranging from 0.25 to 4.

Figures 1 and 2 represent the solar fraction as a function of *NTU* for various ratios of $\dot{m}Cp_{collector}$ to $\dot{m}Cp_{tank}$. Other collector ratios were investigated (Dayan, 1998). Each figure represents a different constant collector flow rate.

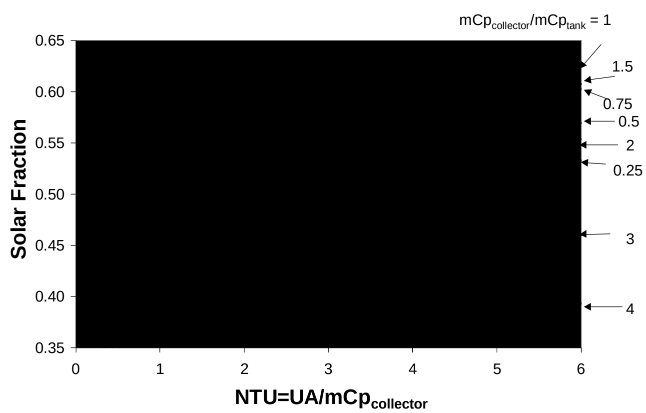

Fig. 1: Solar fraction vs. *NTU* for a collector flow rate of 0.004 kg/s-m^2

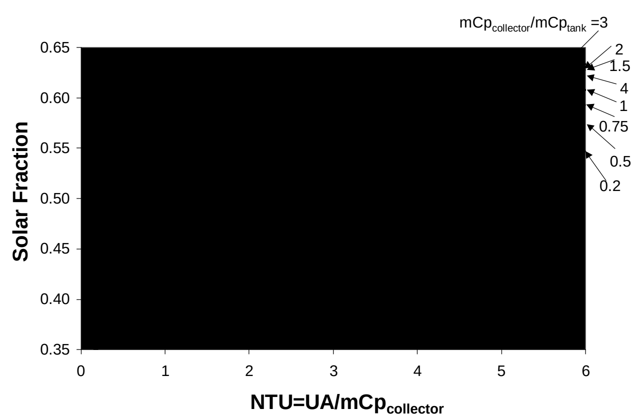

Fig. 2: Solar fraction vs. *NTU* for a collector flow rate of 0.015 kg/s-m^2

It can be seen that the maximum solar fraction is between 0.63 and 0.64 for all flow rates investigated. The optimum occurs at all flow rates for an *NTU* of about six, although for some combinations the optimum is very flat. The advantage of a low collector flow rate is that the *UA* of the heat exchanger need not be as large since the thermal stratification that occurs at low flow rates enhances performance. For example, the *UA* for the flow rate of 0.004 kg/s-m^2 and $\dot{m}Cp_{collector}$ to $\dot{m}Cp_{tank}$ ratio of 1 has a *UA* of 273 W/m^2 for an *NTU* of six, shown in Figure 1. For the same *NTU* and a flow rate of 0.015 kg/sm^2 and $\dot{m}Cp_{collector}$ to $\dot{m}Cp_{tank}$ ratio of 3, shown in Figure 2, the *UA* is 1026 W/m^2. Observing Figure 2, it is clear that the same solar fraction can be achieved at an *NTU* of 3 which signifies a *UA* of 500 W/m^2K, this is still higher than the *UA* required for lower flows. A smaller *UA* implies a smaller and more economical heat exchanger.

An interesting outcome of these results is that the optimal ratio of the collector capacitance rate to the tank capacitance rate is not always the same; ranging from 1 to 2 as the collector flow rate increases. For the lower collector flow rates, the optimal tank flow is close to the average daily load draw of 0.0035 kg/s-m^2.

At collector flow rates of 0.015 kg/s-m^2 the ratio of the collector capacitance to the tank capacitance has less effect on the solar fraction than at lower collector flow rates. For example, ratios of collector capacitance to tank capacitance of 3, 2, 1.5 and 4 result in almost the same solar fraction. However, at low flow rates of 0.004 kg/s-m^2 there is a greater difference in solar fraction at these capacitance rate ratios. It is therefore necessary to carefully choose the flow rates on the tank side when the collector flow rate is low in order to improve system performance.

3. OPTIMIZING FLOW RATES WITH A GIVEN HEAT EXCHANGER MODEL

The results presented in the previous section are convenient for comparing heat exchangers that have the same *NTU* for different flow rates. However, it is more intuitive to look at a fixed heat exchanger and examine the effects of varying the flow rate.

An external tube-in-shell heat exchanger was chosen to produce the following simulation results. The antifreeze solution flows inside the tubes to help prevent fouling and water is used in the shell. The fluid in the collector loop is usually a solution of 50% water and 50% glycol. Ethylene glycol or propylene glycol is often used. Ethylene glycol is toxic and often requires a double-wall heat exchanger to comply with domestic hot water regulations whereas a single-wall heat exchanger can often be used with propylene glycol.

The viscosities of ethylene glycol and propylene glycol have a large temperature dependence. At low temperatures it is very viscous and if the flow rate is low, the flow regime may be laminar so that the heat transfer is dramatically reduced.

A tube-in-shell heat exchanger was designed with a *UA* of approximately 250 W/m^2-K for a collector flow rate of 0.004 kg/s-m^2 and a tank flow rate of 0.004 kg/s-m^2 using the results of Figure 1. The resulting heat exchanger and dimensions are shown in Figure 3.

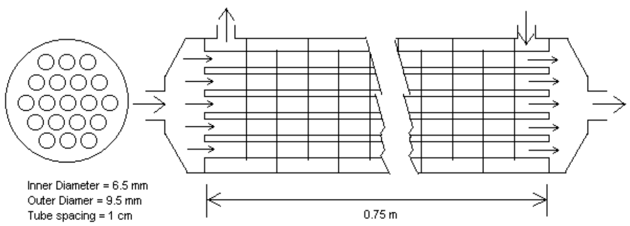

Fig. 3: Tube-In-Shell Heat Exchanger Dimensions

3.1 <u>Antifreeze temperature dependent properties</u>

For ethylene glycol the transition from laminar to turbulent flow for the given heat exchanger occurs at about 23°C for conventional flow and 68°C for low flow. The transition from laminar to turbulent flow for propylene glycol occurs about 15°C higher than the ethylene glycol. The properties are evaluated at the average of the inlet and outlet temperatures of the heat exchanger. However, with such a sharp change from laminar to turbulent flow, evaluating the properties at the average of the inlet and outlet temperatures of the heat exchanger is probably not correct, except when the flow is far from transition. The temperature of the glycol, on average, should be higher than 23°C when the pump is circulating. Therefore, at low flow rates one will expect laminar flow, for the given heat exchanger, for a greater range of temperatures than for the conventional flow system.

Using the specified tube-in-shell heat exchanger, a comparison was made using ethylene glycol with constant properties (independent of temperature) and with temperature dependent properties. The results for the annual solar fraction for various tank flow rate and collector flow rate combinations for constant properties are shown in Figure 4, while the results for temperature dependent properties are shown in Figure 5. Figure 4 reveals that an optimal tank flow rate exists at 0.004 kg/s-m^2 (approximately equal to the average daily load) and an optimal collector flow rate exists at 0.006 kg/s-m^2. However, Figure 5 shows this will not be the case. The increased viscosity of glycol impedes the onset of turbulence, which is detrimental to the heat exchanger *UA*. In this case, the optimal tank flow rate is maintained approximately equal to the average daily water draw, but it seems that increasing the collector flow rate will increase the performance. Note that a design would not be far off from the optimal tank flow rate if the designer chose the design based on fixed properties rather than variable properties. The only draw back is that the system performance calculated with constant properties would be overestimated.

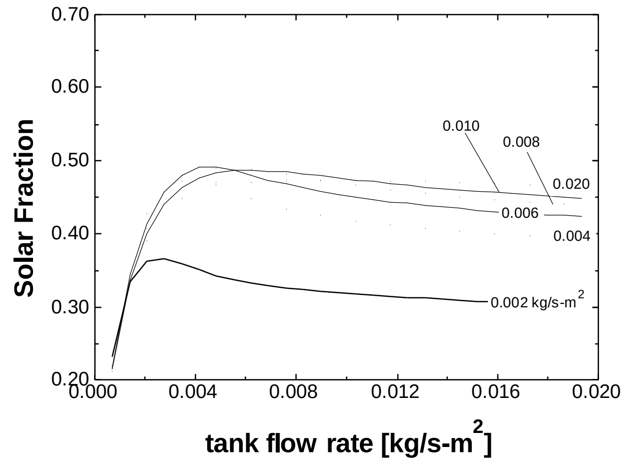

Fig. 4: Annual solar fraction for Madison maintaining ethylene glycol properties fixed.

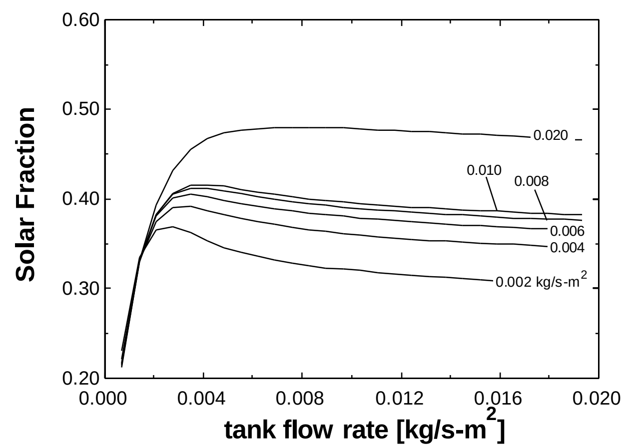

Fig. 5: Annual solar fraction for Madison with temperature dependent properties for ethylene glycol.

Similarly, the results are shown for propylene glycol in Figure 6. The viscosities of both substances are similar, however the thermal conductivity for ethylene glycol is slightly higher than the conductivity for propylene glycol, whereas the specific heat for propylene glycol is higher than ethylene glycol. The increased conductivity for ethylene glycol causes it to be a better antifreeze fluid, although it does require the use of a double-wall heat exchanger in some locations.

To further emphasize the effects of increased viscosity for the antifreeze fluids a simulation with a water-water heat exchanger was performed as shown in Figure 7. The results are very similar to those obtained for the ethylene glycol model (Figure 5) with constant properties. In Figure 7, the water properties are temperature dependent. Again an optimum tank flow rate is found to be at the average daily load, but the ideal collector flow rate is found to be quite low, 0.004 kg/s-m^2. Unfortunately, this simulation only points out the penalty of using an antifreeze solution, and it has no practical application.

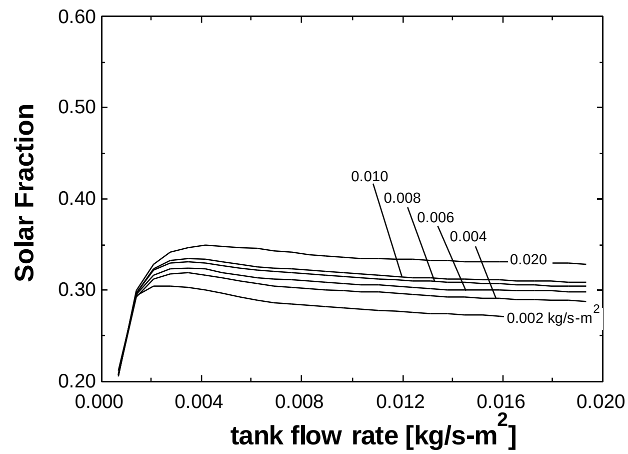

Fig. 6: Annual solar fraction for Madison with temperature dependent properties for propylene glycol.

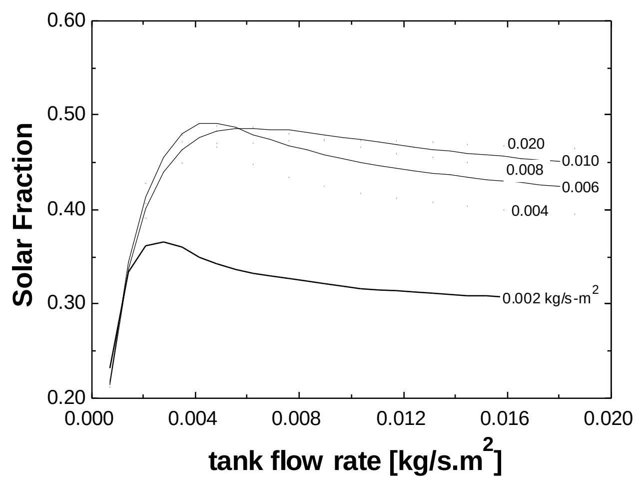

Fig. 7: Water-Water heat exchanger for Madison

3.2 Tank Stratification

The next stage in heat exchanger analysis is to observe the effects of using propylene glycol with a fully mixed tank. Figure 8 demonstrates the impact of a fully mixed tank. Unlike the previous simulations where a stratified tank gave an optimal tank flow rate at the average daily load of 0.0035 kg/s-m^2, there is no optimal tank flow rate. System performance increases with increasing tank flow rate. Also, note that system performance is considerably reduced when compared to the same system with a stratified tank shown in Figure 6.

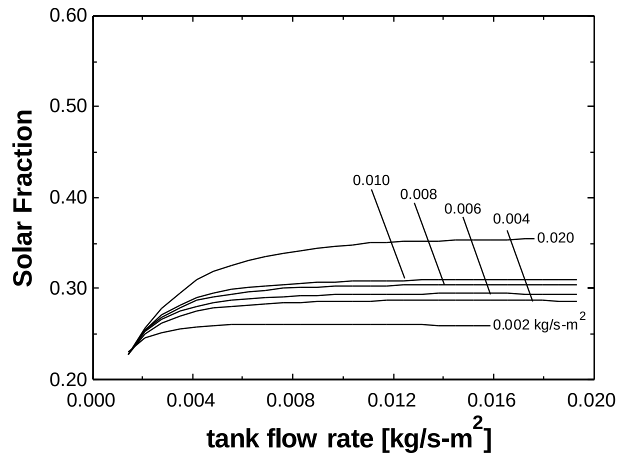

Fig. 8: Fully mixed tank using propylene glycol for Madison

3.3 Effect of Season and location

The curves given in Figure 5 and 6 reveal some interesting insights into optimal heat exchanger operation, but in order to generalize these results more information is needed on how performance will be affected by season and location.

Figure 9 demonstrates the effect of tank flow rate on the heat exchanger performance for the month of July in Madison with propylene glycol as the heat transfer fluid. The same trend as shown in Figure 6 is observed, however the optimum tank flow rate is more marked.

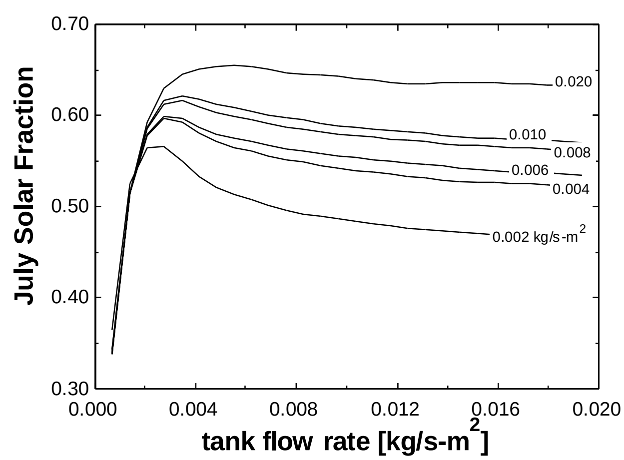

Fig. 9: Heat exchanger with propylene glycol in Madison for the month of July

Similarly, the same trend is observed for the annual solar fraction for Miami using propylene glycol and ethylene glycol, as shown in Figures 10 and 11.

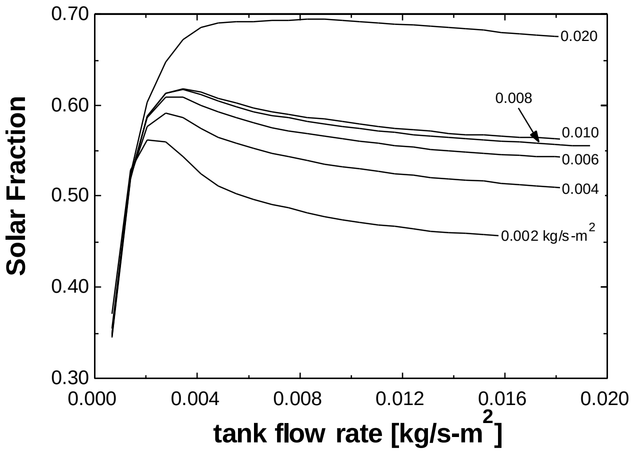

Fig. 10: Heat exchanger with ethylene glycol in Miami

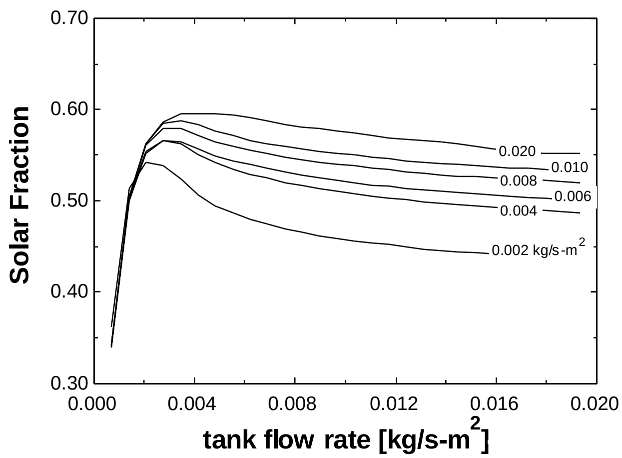

Fig. 11: Heat exchanger with propylene glycol in Miami

3.4 Variation of Heat Exchanger Performance

The heat exchanger used to obtain the results in the previous sections has an extremely good performance. The effect of decreasing and increasing the heat transfer areas is presented in Figures 12 and 13. Figure 14 presents the performance of a propylene–water heat exchanger with an effectiveness of one.

For a tank flow rate of 0.0035 kg/s-m^2 and a collector flow rate of 0.004 kg/s-m^2 the annual solar fraction is 0.37 for the heat exchanger given in Figure 6. Decreasing the heat exchanger length by half and therefore reducing the heat exchange area by half produces an annual solar fraction of 0.32 for the same conditions, shown in Figure 12. On the other hand doubling the heat exchanger area, shown in Figure 13, increases the solar fraction to about 0.42. Therefore, doubling the heat exchanger area will increase the solar fraction by five percent and decreasing the area will decrease the solar fraction by five percent. A perfect heat exchanger with an effectiveness of one, given in Figure 14, yields a solar fraction of almost 0.45.

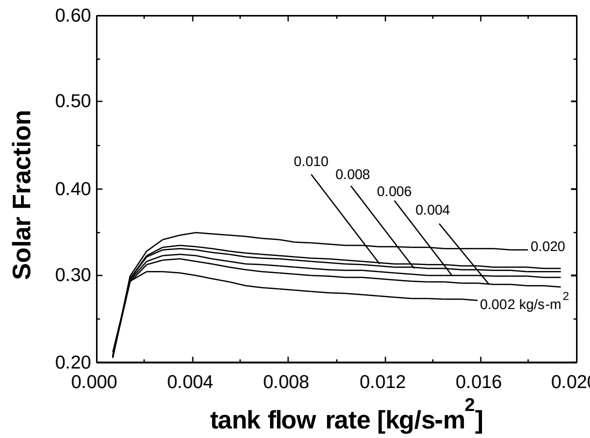

Fig. 12: Heat exchanger with area halved for Madison

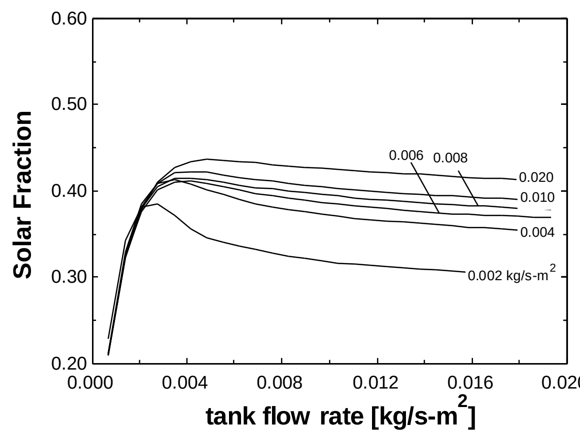

Fig. 13: Heat exchanger with area doubled for Madison

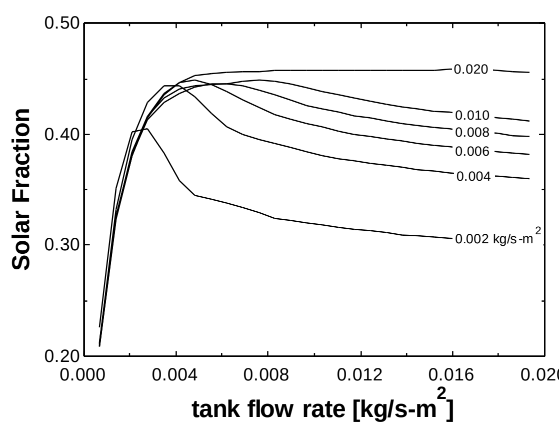

Fig. 14: Heat Exchanger with an effectiveness of one for Madison

4. CONCLUSIONS

In determining heat exchanger dimensions and flow rates there exists a thermal optimum in combination with an economical optimum. The optimum heat exchanger flow rates and dimensions are not a strong function of season and location.

The heat exchanger tank-side flow rate should always be approximately equal to the average load flow rate. This optimum is defined in terms of maximum thermal performance, but also in terms of decreased economic costs. Lower flows require smaller hydraulic systems, which are less expensive. For high collector flow rates, the optimum tank flow rate is less marked; therefore increasing the tank flow rate will not largely affect system performance, but will incur increased economic costs. Large collector flow rates may increase system performance for a given heat exchanger, but will incur larger hydraulic costs in terms of pumping power and piping. Large flow rates also make it more difficult to successfully implement a photovoltaic driven pump due to increased pressure drops.

Use of different antifreeze fluids could also increase system performance. The general trend for the solar fraction as a function of tank and collector flow rates was found to be independent of location and season. The optimization of the heat exchanger size than becomes a question of economics.

Although natural convection heat exchangers were not specifically studied, they enhance tank stratification and some of the results of this study are applicable. However, if the hot water draw is much greater than the natural circulation, poor system performance could be expected. It has been shown that for forced circulation, when the flow rates are lower than the average load, the system performance is reduced. The reason that a very low tank flows reduces system performance is that the collector sees a higher than necessary inlet temperature. It is better to heat the daily load in the collector to a modest temperature than it is to heat less water to a higher temperature. Unfortunately, the determination of the tank flow rate must be predicted by the anticipated hot water draw. Over-predicting the hot water draw will be less detrimental to system performance than under-predicting the draw.

Hollands is correct in stating that optimum flow rates exist for a heat exchanger with a conceptually fixed *UA*. Even with a heat exchanger with *UA* as a function of flow rate, an optimum flow rate exists (even if it is just an economic optimum). The results obtained by Fanney and Klein are also correct. For a heat exchanger with a conventional collector flow rate, such as 0.0151 kg/s-m^2 used by Fanney and Klein, the tank flow rate has little influence given the tank flow rate is larger than the average load. Since Fanney and Klein found reduced performance for a lower tank flow rate, it is possible that the tank flow rate used was below the average load and mixing was occurring.

5. REFERENCES

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