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Analysis of Boiling Flat-Plate Collectors

A detailed model for use with TRNSYS, capable of modelling a wide range of boiling collector types, was used to analyze boiling flat-plate collector systems. This model can account for a subcooled liquid entering the collector, heat losses in the vapor and the liquid return line, pressure drops due to friction in the collector and piping, and pressure drops due to the hydrostatic head of the fluid. The model has been used to determine the yearly performance of boiling flat-plate solar collector systems. A simplified approach was also developed which can be used with the f-Chart method to predict yearly performance of boiling flat-plate collector systems.

Introduction

A boiling fluid (e.g., an organic refrigerant) can be used in place of air or nonboiling liquids as the heat exchange fluid in a solar collector. Claimed advantages for boiling fluid collectors are increased heat transfer coefficients, inherent freeze protection, reduced parasitic energy use and improved transient response to changing meteorological variables. The boiling fluid may either be pumped or circulated passively by thermosyphon action. In the latter case, it is necessary to locate the condenser in a position above the collector. Thermosyphon circulation is considered here.

Several potential disadvantages exist for boiling fluid solar collectors. When organic fluids are used, it may be necessary to have the installation done by a refrigeration/air conditioning specialist unless the system is factory-charged. In a thermosyphon system, the condenser must be placed inside a heated space so it will not freeze. The cost of the boiling fluid is significant, but comparable to the cost of glycol or other antifreeze solutions used in nondraining freeze-protected collectors. Leakage of an organic fluid to the environment is undesirable.

Several studies have been done on boiling flat-plate solar collectors. Schreyer [1] experimentally investigated the use of a thermosyphon refrigerant (R-11) charged solar collector for residential applications. He found that for two identical collectors, the peak instantaneous efficiency of a boiling refrigerant charged collector was 6 percent greater than that of a hydronic fluid circulating solar collector. Soin et al. [2] studied a boiling thermosyphon collector containing an acetone and petroleum ether mixture and developed a modified form of the Hottel-Whillier [3] equation which would account for the fraction of the liquid in a particular collector. Downing and Waldin [4] experimentally studied the heat transfer processes in boiling solar domestic hot water systems using R-11 and R-114. They determined that phase change heat transfer fluids operate with better efficiency and faster response than circulating liquids in solar applications.

A detailed study of boiling fluid solar collectors was done by Al-Tamimi [5] and by Al-Tamimi and Clark [6, 7]. They tested a boiling collector containing R-11 and developed an analytical model to investigate the effect on collector efficiency of subcooling the fluid entering the collector and the level of fluid in the collector. They also investigated the fluid circulation rate, pressure drops, the temperature distribution in the collector, stagnation conditions, and the collector dynamic response. One of the primary results of their work was that collector efficiency was found to be a strong function of solar radiation and $(T_i - T_a)/I$, unlike nonboiling collectors for which efficiency depends primarily on $(T_i - T_a)/I$. Solar radiation dependence results from subcooled fluid entering the collector. If the fluid entering the collector is below its boiling temperature at the pressure it enters the collector, it must be heated up before it will begin to boil. Al-Tamimi and Clark define Z^* to be the fraction of the collector required to heat the fluid to its boiling temperature. They also concluded that the flow rate of refrigerant through the collector was a function of the intensity of solar radiation and dependent upon the system geometry. A modification of the collector heat removal factor, F_R, in the Hottel-Whillier collector equation was developed to account for the boiling and the subcooled portions of the flat-plate boiling collector. However, to use this equation, Z^* and the mass flowrate of fluid through the collector must be known. Variations in the thermodynamic properties of R-11 were not considered.

A recent ASHRAE Standard (109) was developed for testing the thermal performance of flat-plate solar collectors containing a boiling liquid [8]. The testing procedure is largely based upon the analytical methods developed by Al-Tamimi

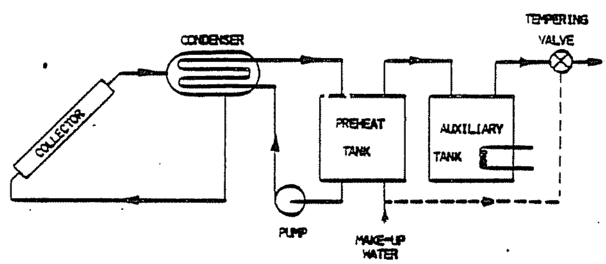

Fig. 1 Boiling flat-plate solar collector system

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and Clark [7]. The test is similar to the ASHRAE 93-77 [9] test except that three separate tests of instantaneous thermal efficiency at different levels of subcooling are required: no subcooling ($T_{sc} < 3C$) and subcooled entering states of 6 and 15C. These tests are designed to determine the dependence of efficiency on the intensity of solar radiation, and the effect of subcooling on efficiency. The 109 Standard also requires the determination of the collector time constant.

Fanney and Terlizzi [10] used the ASHRAE Standard 93-77 test procedure to determine the thermal performance of a boiling flat-plate collector-condenser system. They found that the collector-condenser system tested did not exhibit a dependence on solar radiation.

Confusion exists as to how these boiling fluid collectors work, how they can be modelled, and whether or not they require a separate new testing procedure. This paper will attempt to clear up some of the questions about boiling (2-phase) solar collectors using a detailed model developed for use with TRNSYS [11], capable of modelling a wide range of boiling collector types and situations.

System Description

Figure 1 is a schematic of the system being modelled. It is a 2-tank solar domestic hot water (SDHW) system. The boiling collector and condenser used in this study are respresentative of products in the marketplace for domestic water heating. The system uses a flat-plate collector and a coiled heat exchanger for a condenser. The primary heat exchange fluid (R-11) operates in a thermosyphon mode. Water is circulated through the secondary loop between the condenser and the preheat tank. An auxiliary tank with heating elements is included to ensure that the water is supplied at the set temperature. When a load is drawn from the tank, it is replaced with solar heated water from the preheat tank. A tempering value is included which limits the temperature of the delivered water, if necessary, by mixing it with mains water to achieve the set temperature. The collector loop is only partially filled with liquid refrigerant during installation, enough to fill the collector 2/3 to 7/8 full of liquid. When the absorbed solar radiation is sufficient to overcome collector losses to the environment, the liquid refrigerant is heated and begins to boil. The vapor then rises to the condenser. When the temperature in the condenser reaches a specified temperature above the water temperature in the bottom of the preheat

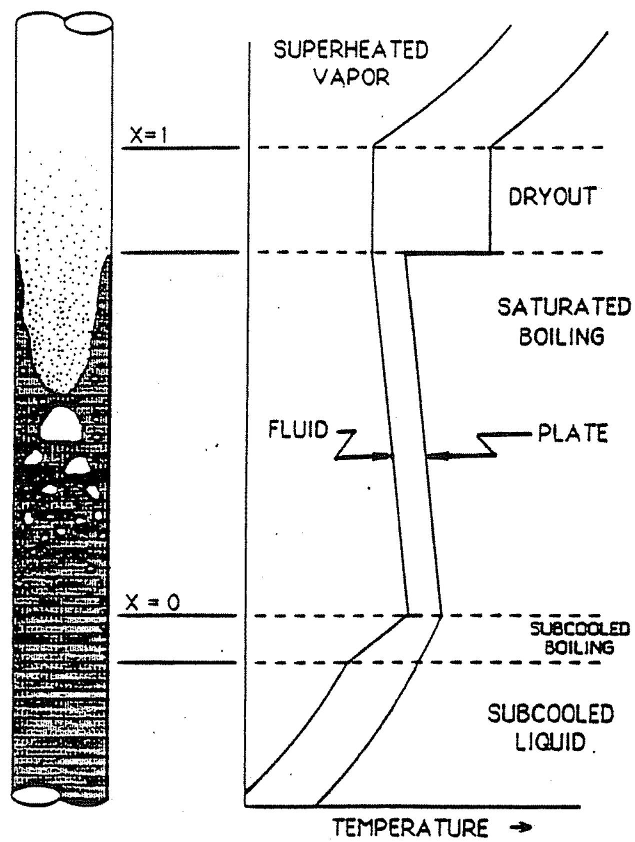

Fig. 2 Temperature profile in a boiling collector

Nomenclature

- A_c = collector area, m^2
- β = collector slope
- C_b = bond conductance, W/mC
- D = riser diameter, m
- d_v = diameter of vapor pipe, m
- ϵ = heat exchanger effectiveness
- f = friction factor
- F = fin efficiency factor
- F' = collector efficiency factor
- F_{boil} = boiling collector efficiency factor
- F_R = collector heat removal factor
- F'_R = modified collector heat removal factor
- g = gravitational constant, m/s^2
- h_{fg} = heat of vaporization, kJ/kg
- h_{fi} = fluid heat transfer coefficient, W/m^{2}C
- I = solar radiation per unit area on collector surface, W/m^2
- L_v = length of vapor line, m
- L_c = length of collector riser, m
- $(\dot{m}C_p)_{\text{ref}}$ = refrigerant capacitance rate, kJ/hr-C
- $(\dot{m}C_p)_{\min}$ = minimum capacitance rate, kJ/hr-C
- $(\dot{m}C_p)_w$ = water capacitance rate, kJ/hr-C
- P_{boil} = pressure at collector outlet, Pa
- P_{cond} = pressure in condenser, Pa
- P_{sat} = saturation boiling pressure, Pa
- ΔP_{stat} = hydrostatic head pressure difference, Pa
- ΔP_v = frictional pressure drop in vapor line, Pa
- Q_{cond} = condenser energy output, kJ/hr
- $Q_{\text{loss},l}$ = heat losses in liquid line, kJ/hr
- $Q_{\text{loss},v}$ = heat losses in vapor line, kJ/hr
- Q_u = utilizable energy gain in collector, kJ/hr
- $Q_{u,b}$ = energy gain in boiling section of collector, kJ/hr
- $Q_{u,nb}$ = energy gain in non-boiling section of collector, kJ/hr
- R = ideal gas constant, Nm/kgK
- Re_D = Reynolds Number
- ρ = density of liquid refrigerant, kg/m^3
- s = initial fill of collector, fraction
- S = absorbed solar energy, W/m^2
- T_a = ambient temperature, K
- T_{boil} = collector boiling temperature, K
- T_{cond} = condenser temperature, K
- T_i = condenser inlet water temperature, K
- $T_{c,i}$ = collector inlet fluid temperature, K
- ΔT_{lm} = log mean temperature difference, K
- ΔT_{sc} = subcooling due to heat losses, K
- $(\tau\alpha)$ = transmittance, absorptance product
- UA_{cond} = condenser heat transfer conductance W/C
- UA_l = liquid line heat transfer conductance W/C
- UA_v = vapor line heat transfer conductance W/C
- U_L = collector loss coefficient, W/C
- v = vapor velocity, m/s
- W = distance between collector riser tubes, m
- Z^* = fraction of collector in a subcooled state

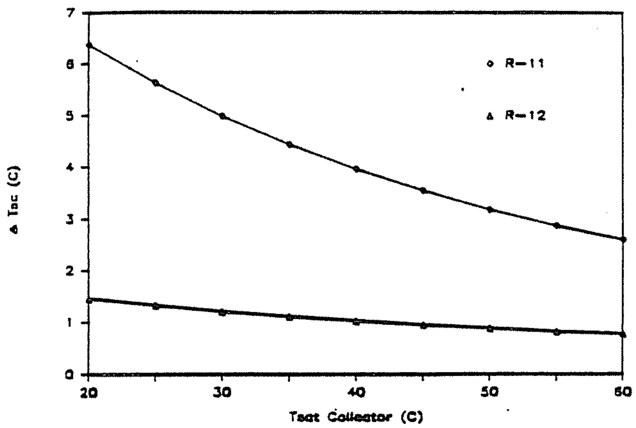

Fig. 3 Subcooling due to the hydrostatic head of the fluid in the collector

tank, the circulation pump in the secondary loop is activated. The vapor is condensed by heat transfer to the circulating water in the condenser coil, drips off, and returns to the collector by the liquid return line. The thermosyphon action will continue as long as the condenser temperature is below the saturation temperature of the entering vapor.

In standard hydronic collectors, the fluid enters at a low temperature and is heated as it passes through the collector. The situation is quite different in a boiling collector, as illustrated in Fig. 2. In a boiling collector-condenser system, in which the fluid entering the collector is subcooled, the fluid must flow part way up the collector before it is heated to the saturation temperature and begins to boil. Subcooling may occur as a result of frictional pressure drops in the collector and vapor lines, hydrostatic head of the fluid in the collector, subcooling in the condenser, and heat losses in the liquid return line. Except for heat losses, all of the energy gained in the subcooled portion of the collector will be released at the condenser.

Once boiling begins, the fluid temperature no longer increases as it flows through the collector. The (saturated) fluid temperature will actually decrease slightly in the flow direction due to the reduced hydrostatic pressure. In most cases, the variation in boiling temperature due to pressure reduction is only a few degrees. Figure 3 shows the maximum amount of subcooling which could occur due to the hydrostatic head in a collector 1.8 meters long and at a 60 deg slope for R-11 and R-12. ΔT_{sc} is the amount of subcooling, and T_{sat} is the saturation boiling temperature at the top of the collector. This figure demonstrates a maximum subcooling due to pressure reduction in the collector since it assumes the collector is completely full of refrigerant when it is actually only 2/3 to 7/8 full depending on the initial charge. From the figure it can be seen that the amount of subcooling depends on both temperature and type of refrigerant. In the results presented below, the variation in boiling temperature through the collector was approximately 2C and was neglected. The temperature in the boiling section of the collector was assumed to be constant.

It is also possible for the fluid to become superheated in the upper portion of the collector. This is not desirable since it reduces collector efficiency by increasing the collector temperature and by reducing F_R, the collector heat removal factor. Superheating is most likely to occur at times of either low or very high solar radiation. In either case, the superheating is due to "dry out" in the collector. Dryout occurs when the top portion of the collector is not wetted by liquid refrigerant. During times of low solar radiation, the mass flow rate of refrigerant is insufficient to wet the upper position or the collector. Since the collectors are not completely filled with refrigerant, they rely on the turbulence and effective density change of boiling to wet the top of the collector. Some boiling-collectors are designed with the inlet at the top of the collector which helps in the wetting process. At high levels of solar radiation, the mass flow rate of refrigerant is again not sufficient to wet the entire collector due to the friction in the collector, vapor, and liquid refrigerant lines which restricts the flow of liquid refrigerant. To maximize collector efficiency, it is desirable to avoid subcooling and superheating in the collector in order to take advantage of the uniform temperature and high boiling heat transfer coefficient. For the flat-plate collector considered in this paper, fluid superheating does not appear to be significant [5, 10] and is not considered.

Frictional pressure drops in the collector-condenser system occur in the vapor line between the collector and the condenser, the liquid return line, and the collector. Friction in the vapor line produces three noticeable effects on the system. First, the pressure difference between the collector and condenser will result in the vapor entering the condenser in a superheated state. The rate of heat transfer in the condenser will thus be reduced compared to the ideal case of no friction because of a reduced condenser heat transfer coefficient. For a particular rate of heat transfer in the condenser, the average collector temperature increases with increasing pressure difference between the collector and condenser which lowers the collection efficiency. A second effect of friction in the vapor line is that an additional height of liquid in the return line is necessary to increase the hydrostatic head of the liquid enough to overcome the friction and maintain the circulation of refrigerant. The condenser should be located far enough above the collector such that it will not be flooded by the additional liquid head. The displacement of fluid from the collector to the liquid return line may cause "dry out" conditions in the collector. A third effect of friction in the vapor line is that the fluid entering the collector will be subcooled due to the lower saturation temperature in the condenser, resulting in a portion of the collector being subcooled.

The major effect of friction in the return line is the head of liquid necessary to overcome the friction. This will not cause any subcooling at the collector inlet because the pressure increase due to the liquid hydrostatic head is equal to the pressure drop caused by friction. However, friction in the liquid return line may cause superheating of the exit vapor due to an insufficient fluid flow rate in the collector.

Friction in the collector results in effects similar to friction in the vapor and liquid return lines. There will be a larger pressure drop across the collector than for the no friction case, and thus more subcooling of the collector inlet fluid. The vapor entering the condenser may be superheated due to dryout caused by the restriction of flow in the collector.

In any of the situations mentioned above, friction has a negative effect on the efficiency in the boiling collector-condenser system. A well designed system should attempt to minimize frictional effects.

Model Description

The boiling collector water heating system in Fig. 1 was modelled using the TRNSYS 12.1 simulation program [11]. The system model employed standard TRNSYS library components for tanks, pump, controls, and solar radiation processing. The system incorporates a daily mass flow load profile developed by a RAND Corporation survey [12] for a "typical" residence. Since the condenser is an integral part of the boiling collector-condenser system, a separate component was developed to model both the boiling collector and condenser together.

The basic structure of the boiling collector-condenser model is an energy balance which assumes a quasi-equilibrium state such that the energy gain of the collector equals the energy transfer to the water in the condenser. In its simplest form, the model assumes that the entire collector is in a fully boiling condition and at a constant temperature (i.e., no subcooling, superheating, pipe heat losses or property variations).

$$Q_u = Q_{cond} \tag{1}$$

Heat transfer in the condenser is modelled using a log-mean temperature difference with a condenser heat transfer coefficient, UA_{cond}.

$$Q_{cond} = UA_{cond} \Delta T_{LMTD} \tag{2}$$

The collector-condenser model assumes that only condensation heat transfer occurs in the condenser, and there is no subcooling of the condensed liquid. This is a good assumption for fluorocarbon refrigerants in a drip style condenser.

A variation of the Clausius-Clapeyron [13] equation is used to account for the variation in boiling temperature due to variations in pressure caused by friction and the hydrostatic head of the fluid. The temperature in the condenser is found using:

$$T_{cond} = T_{boil} + \left[\frac{RT^2{}_{boil}}{h_{fg}}\right] ln\left[\frac{P_{cond}}{P_{boil}}\right] \tag{3}$$

Vapor pressures are calculated using an equation developed by Martin [14] with the form:

$$\log P_{sat} = A + B/T + DT + E(F-T)/T \log(F-T) \tag{4}$$

The coefficients for many different refrigerants are found in [15].

The heat transfer in a boiling flat-plate collector can be modelled in a manner similar to a flat-plate hydronic collector, with the assumption that the fluid boils at a constant pressure and thus constant temperature. The basic collector equation [16] applied to the boiling flat-plate collector is:

$$Q_u = A_c F_R [S - U_L (T_{boil} - T_a)] \tag{5}$$

where

$$S = (\tau\alpha)_n I \tag{6}$$

$$F_{boil} = \frac{\dfrac{1}{U_L}}{W\left[\dfrac{1}{U_L[D + (W-D)F]} + \dfrac{1}{C_b} + \dfrac{1}{D_i h_{fi}}\right]} \tag{7}$$

In equation (7), h_{fi} is the boiling heat transfer coefficient between the fluid and the collector, and T_{boil} is the saturation boiling temperature in the collector.

In most situations the collector does not operate in a fully boiling mode. Generally the fluid enters the collector as a subcooled liquid. The approach for modelling a boiling collector with subcooling is to break the collector into a boiling section of area, $(1 - Z^*)A_c$, and a nonboiling section of area, $Z^* A_c$.

A modified collector heat removal factor, F_R', is defined by deWinter [17] which accounts for the effect of a heat exchanger.

$$F_R' = \frac{F_R}{1 + \left[\dfrac{A_C F_R U_L}{(\dot{m}C_p)_C}\right]\left[\dfrac{(\dot{m}C_p)_{coll}}{\epsilon(\dot{m}C_p)_{min}} - 1\right]} \tag{8}$$

Table 1 System Parameters

Collector			Physical Characteristics
	Gross Area		4.08 m^2
	Net Area		3.51 m^2
	Absorber Plate		Steel
			1.8 m long
			1.94 m wide
			1.76 mm thick
			$\epsilon = 0.75$
			$\alpha = 0.91$
	Risers		Steel
			32 flow tube
			9.53 mm OD
			7.77 mm ID
	Cover Plates		2, Tedlar
			0.1 mm thick
			$\tau = 0.93$
			Index of Refraction = 1.45
			15.9 mm space between covers
			63.5 mm space between plate and cover
	Insulation		Glass Fiber
			back 63.5 mm
			edge 25.4 mm
Derived Collector Parameters		Base	High Quality
	Area	4.08 m^2	4.08 m^2
	F_{boil}	0.96	0.96
	F'	0.56	0.56
	U_L	7.5 W/m^2°C	3.6 W/m^2°C
	$(\tau\alpha)$	0.76	0.80
Preheat Tank			
	Volume	303 liters	
	Thermal Conductance	1.081 W/m^2°C	
	Envelope Temperature	21°C	
Auxiliary Tank			
	Volume	151 liters	
	Thermal Conductance	1.047 W/m^2°C	
	Envelope Temperature	21°C	
	Set Temperature	60°C	
	Rand Load Profile	300 liters/day	
Connecting Lines		Vapor	Liquid
	Diameter	0.0141 m	0.0141 m
	Length	10.0 m	10.0 m
	UA	2.5 W/m^2°C	5.0 W/m^2°C

where ϵ is the effectiveness of the heat exchanger. For a condenser, the effectiveness is:

$$\epsilon = 1 - \exp\left(\frac{-UA_{\text{cond}}}{(\dot{m}C_p)_w}\right) \tag{9}$$

Using the area of the boiling section of the collector, $(1-Z^*)A_C$, a modified collector heat removal factor, F'_R can be written:

$$F'_R = \frac{F_R}{1 + \dfrac{A_c(1-Z^*)F_R U_L}{(\dot{m}C_p)_w\left[1-\exp\left(\dfrac{UA_{\text{cond}}}{(\dot{m}C_p)_w}\right)\right]}} \tag{10}$$

Using F'_R, the condenser inlet water temperature, T_i, can be used in the collector equation in place of the collector fluid inlet temperature. The useful energy gain in the boiling portion of the collector can then be expressed:

$$Q_{u,b} = A_c(1-Z^*)F'_R[S - U_L(T_i - T_a)] \tag{11}$$

The nonboiling section is analyzed as a nonboiling collector, with the exception that the outlet temperature of the nonboiling section, T_{boil}, is known. The temperature distribution in the nonboiling section of the collector takes the form [16],

$$\frac{T_{\text{boil}} - T_a - \dfrac{S}{U_L}}{T_{c,i} - T_a - \dfrac{S}{U_L}} = e^{-[Z^* A_c U_L F'/(\dot{m}C_P)_{\text{ref}}]} \tag{12}$$

where $T_{c,i}$ is the temperature of liquid entering the collector. This can be rearranged to find Z^*, the fraction of the collector necessary to heat the collector fluid up to T_{boil}.

$$Z^* = \frac{(\dot{m}C_P)_{\text{ref}}}{A_c U_L F'} \ln\left(\frac{T_{c,i} - T_a - \dfrac{S}{U_L}}{T_{\text{boil}} - T_a - \dfrac{S}{U_L}}\right) \tag{13}$$

The energy gain in the nonboiling section is found using:

$$Q_{u,nb} = A_c Z^* F_R[S - U_L(T_{c,i} - T_a)] \tag{14}$$

The total energy gain in the collector is:

$$Q_u = Q_{u,b} + Q_{u,nb} \tag{15}$$

The mass flowrate of refrigerant can be approximated using:

$$\dot{m}_{\text{ref}} = Q_{u,b}/h_{fg} \tag{16}$$

Energy losses in the collector-condenser loop will have different effects on the system depending on where they occur. Heat transfer from the vapor line will cause liquid to condense in the line and flow back to the top of the collector. Ignoring pressure drops in the line, this condensation occurs at a constant temperature and these losses take the form:

$$Q_{\text{loss},v} = UA_v(T_{\text{boil}} - T_a) \tag{17}$$

Beckman [18] showed that these losses can be included in the collector equation by defining a new collector loss coefficient U_L:

$$U'_L = U_L + \frac{UA_v}{(1-Z^*)A_c F_R} \tag{18}$$

where

$$F_R = F_{\text{boil}}$$

Losses which occur in the liquid return line cannot be accounted for by modifying the parameters in the boiling collector equation because they cause the liquid to be subcooled. The liquid line losses can be determined using:

$$\begin{aligned} Q_{\text{loss},l} &= UA_l \Delta T_{\text{LMTD}} \\ &= UA_l(T_{c,i} - T_a) \end{aligned} \tag{19}$$

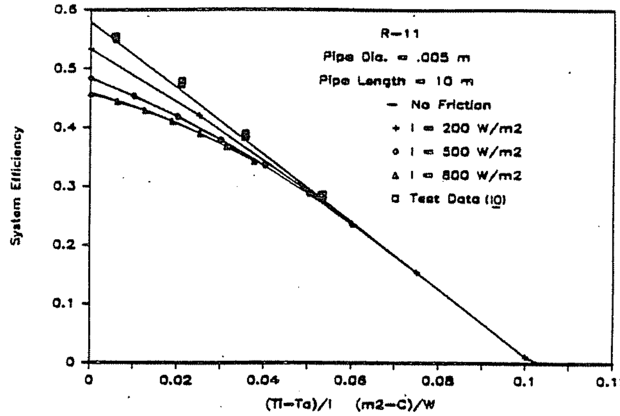

Fig. 4 The effect of friction on instantaneous system efficiency

The amount of subcooling caused by liquid line losses is approximately:

$$\Delta T_{sc} = \frac{Q_{\text{loss},l}}{(\dot{m}C_P)_{\text{ref}}} \tag{20}$$

Losses from the condenser jacket, vapor line and liquid return line to the heated space can be accounted for in a similar fashion.

Pressure drops in the vapor line due to friction are accounted for using

$$-\Delta P_v = \frac{f\rho L v^2}{2d_v} \tag{21}$$

where the friction factor f, is determined using a correlation for turbulent flow in smooth pipes [19].

$$1/\sqrt{f} = 0.87\ \ln(\text{Re}_D\sqrt{f}) - 0.8 \tag{22}$$

The hydrostatic head of the refrigerant in the collector is calculated using:

$$\Delta P_{\text{stat}} = (s - Z^*)\rho g L_c \sin\beta \tag{23}$$

where β is the slope of the collector, L_c is the length of the collector, and s is the initial fill of the collector.

The model also accounts for property variations with temperature. A quadratic interpolation of tabular data from [20] is used to determine density, viscosity, conductivity, specific heat, and the heat of vaporization of the fluid.

Fluid property variations with temperature and pressure introduce a nonlinearity to the equations. Thus an interactive scheme is necessary to solve for the utilizable energy gain from the boiling collector.

Parametric Studies

The model described in the previous section, was developed to study the performance of a boiling collector-condenser system and to determine its relative sensitivity to design variables and control strategies. Both the instantaneous and long-term performance of these systems have been investigated.

Instantaneous Performance. The boiling collector-condenser system chosen as the base case system is the one tested by Fanney and Terlizzi [10]. The physical characteristics of this system are listed in Table 1. The collector loss coefficient and transmittance-absorptance product, and the condenser heat transfer coefficient were analytically determined from the data in Table 1. The instantaneous collector-condenser efficiency curve versus $(T_i - T_a)/I$, determined by the model described in the preceding section, is shown in Fig. 4.[1] The top line represents the ideal case which assumes no

[1]Note that symbols are used in Figs. 4–6 to label the results from the model for different parameters. Only the squares in Fig. 4 represent test data.

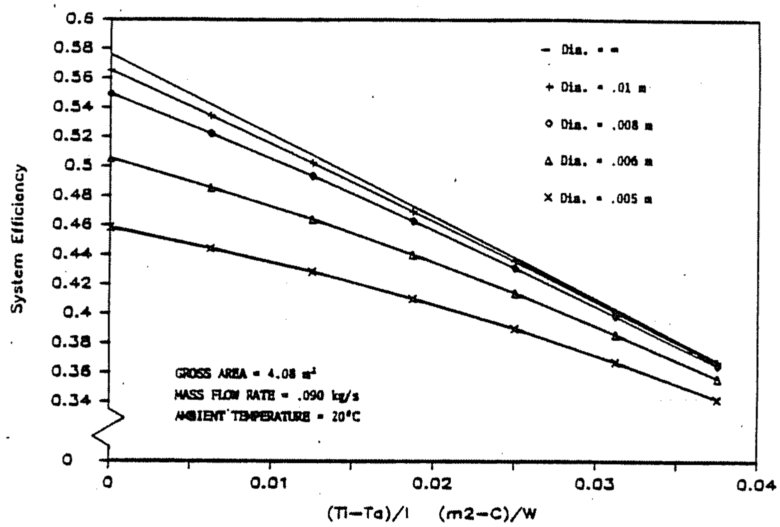

Fig. 5 The effect of pipe diameter on instantaneous system efficiency

friction in the collector, vapor line or liquid return line, no heat losses except from the collector, a constant boiling temperature in the collector, and as a result, no subcooling or superheating in the collector. In this case, the collector-condenser system efficiency versus $(T_i - T_a)/I$ is a straight line similar to most hydronic flat-plate collectors with no detectable dependence on solar radiation.

The three lower curves in Fig. 4 are for a collector where vapor line friction and collector hydrostatic head have been included. Figure 4 was drawn assuming that the vapor line has a diameter of 0.005 m and a length of 10 m. From Fig. 4 it can be seen that the collector-condenser system efficiency is dependent on the intensity of solar radiation as well as $(T_i - T_a)/I$. By decreasing $(T_i - T_a)/I$, the deviation from the ideal case efficiency increases. Also, the larger the solar radiation intensity, the larger the reduction in system efficiency from the ideal case.

The results of Fig. 4 appear to be in contradiction with the experimental data of Al-Tamimi and Clark [5, 6, 7]. They observe efficiency to increase with increasing solar radiation at fixed $(T_i - T_a)/I$. In their experiment, the collector fluid was subcooled by heat exchange to an external fluid before entering the collector. In this manner, points having differing values of solar radiation but the same value of $(T_i - T_a)/I$ could be obtained by altering T_i and thereby the degree of subcooling. As I increases, $(T_i - T_a)$ must also increase to keep $(T_i - T_a)/I$ at a constant value which, for constant T_a, implies that the collector fluid enters the collector at a higher T_i and thus a less subcooled state. All else being the same, Z^*, the fraction of the collector required to heat the collector fluid to its boiling temperature, will be lower when the fluid enters in a less subcooled state. Since efficiency is a decreasing function of Z^*, as shown experimentally by Al-Tamimi and Clark and seen from equation [10], the efficiency should increase with increasing solar radiation. The point is that this effect occurs not as a result of fluid friction but as a direct result of the external subcooling of the collector fluid.

The results in Fig. 4 show that, when friction in the vapor line is considered (without external subcooling), efficiency decreases with increasing radiation at fixed $(T_i - T_a)/I$. This effect is largest at low values of $(T_i - T_a)/I$ corresponding to high values of I. As solar radiation is increased, the mass flowrate of collector fluid in the vapor line increases, which increases the pressure difference between the collector and the condenser. A lower pressure in the condenser results in a lower temperature for condensation and, as a result, a lower log-mean temperature difference between the collector fluid and the water. Since the rate of energy collection in the collector must be equal to the rate of heat transfer in the condenser (neglecting pipe energy losses), the collector temperature (and thus pressure) must rise, lowering the collector efficiency, as seen in Fig. 4.

Some of the experimental data taken by Fanney and Terlizzi [10] for radiation levels of 300, 575, and 850 W/m^2 have also been included in Fig. 4. In their system, the vapor line was only 1.5 m long and the diameter was 0.0141 m, nearly three times larger than that used in Fig. 4. When the recommended [21] vapor line diameter (0.0141 m) is used in the model, the dependence of system efficiency on solar radiation is negligible. The standard efficiency curve for this system lies almost on top of the ideal system efficiency curve and is independent of solar radiation intensity, as observed by Fanney and Terlizzi.

Figure 5 illustrates how system efficiency drops off by decreasing vapor line diameter for a solar radiation intensity of 800 W/m^2 and a pipe length of 10 m. Figure 6 shows the system efficiency for various vapor line lengths and with a diameter of 0.0141 m. In Figs. 4, 5, and 6, thermal capacitance effects, heat losses from the vapor line, and the hydrostatic head of the vapor were not included. If these effects had been considered, the efficiency in Fig. 6 would have been much lower.

In the systems considered for Figs. 4, 5, and 6, the hydrostatic head in the liquid return line necessary to overcome the frictional pressure drop between the collector and condenser was sometimes as large as 15 m as a result of the unrealistic vapor line lengths or diameters examined. In a real system, the pressure drop between the collector and condenser would be limited by the physical height of the condenser above the collector. If the pressure drop in the vapor line is large, the condenser may flood. There will also be a net transport of liquid from the collector to the liquid return line which may cause collector "dry-out". If the physical configuration of the system does not allow the hydrostatic head to get this large, then the collector-condenser system will adjust such that a new equilibrium condition is met at a higher collector temperature and a lower refrigerant mass flowrate. In this case the pressure drop would be equal to the maximum hydrostatic head possible in the liquid return line.

Long-Term Performance. A plot of instantaneous collector-condenser system efficiency is informative when comparing different collectors, but it is less useful for estimating the long-term performance of a SDHW system. The yearly solar fraction, the quantity of energy supplied by solar energy, is a useful measure of system performance. The SDHW system modelled in this study is described in Section 3.

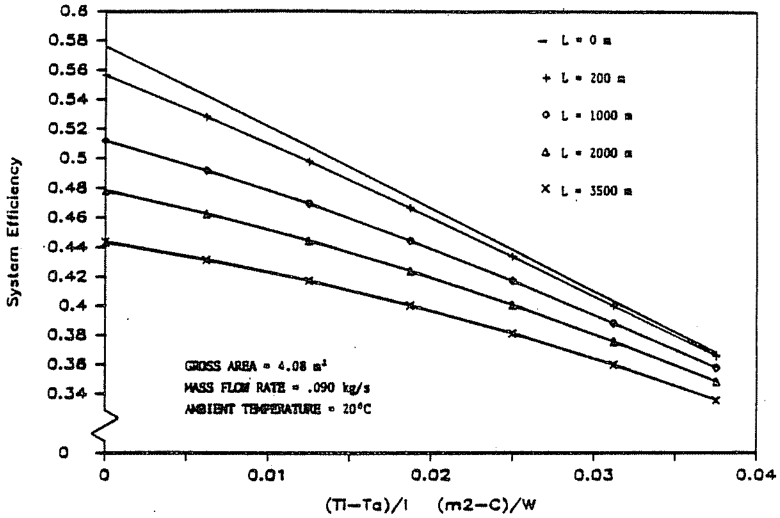

Fig. 6 The effect of pipe length on instantaneous system efficiency

Table 2 Yearly solar fraction for a boiling collector-condenser system in Madison, Wisconsin

	Yearly Solar Fraction	% Difference
1. Ideal Boiling Collector (no friction or hydrostatic head)	0.2967	--
2. No Friction Boiling Collector (with hydrostatic head included)	0.2951	0.5%
3. NBS Test Collector (Friction and hydrostatic head)	0.2950	0.6%
4. Base Case Boiling Collector (Friction and hydrostatic head)	0.2949	0.6%
5. Base Case Boiling Collector with Heat Losses in the Vapor and Liquid Return Lines	0.2742	7.6%

Table 1 gives the physical dimensions and parameters used in the base case system studied here.

Several yearly simulations were run for boiling collector-condenser systems in Madison, Wis., to determine if the reduction in system efficiency caused by friction, hydrostatic head and heat losses are significant. The results are listed in Table 2. Of the three, only heat losses appear to have a significant effect on annual system performance for the base case boiling collector-condenser system. Parametric studies were performed using monthly simulations for March in Madison, Wis. The parametric studies included the effect of condenser size, UA_{cond}, and collector area, A_c, on monthly solar fraction. These studies were performed for the base case system, and for a higher quality, selective surface collector described in Table 1. Figure 7 shows the effect of condenser size on monthly solar fraction. The base case system condenser appears to be of more than adequate size for both collectors, as seen from the flatness of the curve for the size of the base case condenser. Figure 8 shows how the monthly solar fraction varies with collector area. This figure assumes a constant condenser size, but the mass flowrate of water in the condenser is proportional to the collector area (90 kg/hr-m^2). Figure 8 appears very much like the curve of solar fraction versus collector area for hydronic SDHW systems.

An interesting control strategy investigated by Wuestling [22] for standard hydronic solar collectors, is the reduction of the mass flowrate of water through the collector, to increase thermal stratification in the preheat storage tank. The collector heat removal factor, F_R, is reduced by doing this, but the increased temperature stratification in the preheat tank lowers the average temperature of the water entering the collector and increases the average temperature of the fluid being supplied to the load. Wuestling determined that optimum system performance occurred when the average daily collector mass flowrate was approximately equal to the average daily load flow, a condition which occurs at approximately 20 percent of normal collector flowrates (i.e., 10 l/hr-m^2). At reduced flows, annual system performance was as much as 15 percent higher than systems with high collector flowrates and therefore unstratified storage.

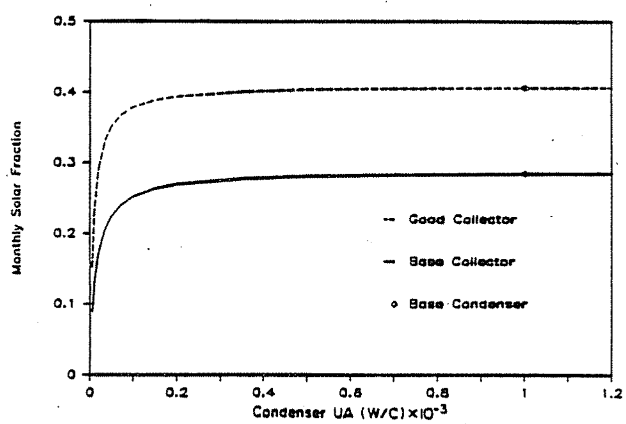

Fig. 7 The effect of condenser size on solar fraction

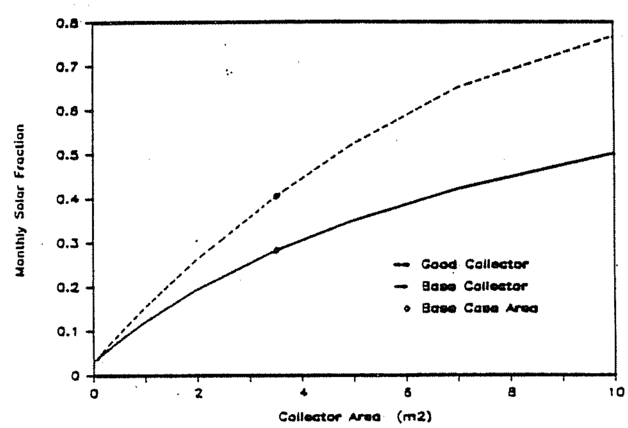

Fig. 8 The effect of collector area on solar fraction

In systems with a heat exchanger located between the collector and preheat storage tank, it is not the mass flowrate in the collector but rather the tank-heat exchanger flow which must be reduced to reduce recirculation of water in the storage tank. However, the overall heat transfer coefficient of the heat exchanger will be reduced if the mass flowrate of the tank-heat exchanger flow is reduced. The trade off in the boiling collector SDHW system to obtain more stratification in the preheat storage tank is a reduction in condenser UA, and therefore in F'_R.

The effect of condenser water flow rate on the performance of boiling collector water heating systems was examined by simulations in the Madison climate for both fully-mixed and thermally stratified preheat storage tanks. The results for the fully-mixed tank show an increasing but asymptotic dependence of monthly solar fraction on flowrate. With a thermally stratified pre-heat tank, the simulation results indicate that there is an optimum condenser water flowrate. The optimum varies from month to month and is lowest during the summer months. For the base case system, the optimum flowrate was between 10 to 15 kg/hr-m^2 in five locations within the US. The optimum flowrate for a boiling collector system is always higher than for a comparable hydronic system because F'_R in equation (10) is more sensitive to flowrate than is F'_R for a hydronic collector. As UA_{cond}, and therefore the physical size of the condenser unit, is increased the optimum condenser water flowrate decreases. Undersized condensers may not have a finite optimum flowrate.

Conclusions

The effect of friction in the collector, vapor line, and liquid return line can be a very important factor in determining the performance of a boiling collector-condenser system. Frictional pressure drops are responsible for both subcooling and superheating in the collector. Subcooled and superheated states in the collector cause the solar radiation dependence on the instantaneous collector efficiency, as reported by Al-Tamimi [5, 6, 7].

For an ideal boiling flat-plate collector-condenser system, for which a saturated liquid enters the collector, and a saturated vapor exits, the efficiency curve has the same form as that of conventional nonboiling flat-plate collectors. The boiling flat-plate collector-system tested by Fanney and Terlizzi [10] exhibits this form. It appears that careful design can reduce the amount of subcooling and superheating to a point that they have a negligible effect on collector performance.

In many boiling flat-plate collector systems where heat losses and the effects of friction are minimal, a simplified approach can be used to model the system performance. A modified form of the heat removal factor, F'_R, is used to account for the effect of the condenser (equation (20)). The f-Chart method [23] can then be used to predict the long-term performance of the boiling collector system. The assumptions made using this simplified approach are optimistic and yield a maximum performance estimate. However, studies done with the detailed model show that the sensitivity of long-term performance to subcooling and moderate pressure losses is small.

Acknowledgments

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References

1 Schreyer, J. M., "Residential Application of Refrigerant Charged Solar Collectors," *Solar Energy*, Vol. 26, 1981, pp. 307–312.

2 Soin, R. S., Rao, K. S., Rao, D. P., and Rao, K. S., "Performance of Flat-Plate Solar Collector with Fluid Undergoing Phase Change," *Solar Energy*, Vol. 23, 1979, pp. 69–73.

3 Hottel, H. C., and Whillier, A., "Evaluation of Flat-Plate Collector Performance," Trans. of Conf. on Use of Solar Energy, Part 1, Vol. 74, University of Arizona Press, 1958.

4 Downing, R. C., and Waldin, V. W., "Phase-Change Heat Transfer in Solar Hot Water Heating Using R-11 and R-114," ASHRAE Trans. 86, Part I, 1980.

5 Al-Tamimi, A. I., "Performance of Flat-Plate Solar Collector in a Closed-Loop Thermosiphon Using Refrigerant-11," Ph.D Thesis, Chemical Engineering Department, University of Michigan, Ann Arbor, Sept. 1982.

6 Al-Tamimi, A. I., and Clark, J. A., "Thermal Analysis of a Solar Collector Containing a Boiling Fluid," *Proceedings of the 1983 Annual Meeting of the American Solar Energy Society*, Minneapolis, MN, June 1–3, 1983.

7 Al-Tamimi, A. I., and Clark, J. A., "Thermal Performance of a Solar Collector Containing a Boiling Fluid (R-11)," ASHRAE Winter Meeting, Atlanta, GA, Jan. 29–Feb. 2, 1984.

8 ASHRAE 109P, "Methods of Testing to Determine the Thermal Performance of Flat-Plate Solar Collectors Containing a Boiling Liquid," ASHRAE, Atlanta, GA, 1983.

9 ASHRAE 93-77, "Methods of Testing to Determine the Thermal Performance of Solar Collectors," ASHRAE, Atlanta, GA, revised printing, 1978.

10 Fanney, A. H., and Terlizzi, C. P., "Testing of Refrigerant-Charged Solar Domestic Hot Water Systems," *Solar Energy*, Vol. 35, No. 4, pp. 353–366.

11 Klein, S. A., et al., "TRNSYS—A Transient Simulation Program," Engineering Experiment Station Report 38-12, University of Wisconsin-Madison, 1983.

12 Mutch, J. J., "Residential Water Heating, Fuel Consumption Economics and Public rotary, RAND, Dept. R1498, NSF, 1974.

13 Van Wylen, G. J., and Sonntag, R. E., *Fundamentals of Classical Thermodynamics*, 2nd ed., revised printing, Wiley, New York, 1978.

14 Martin, J. J., "Correlations and Equations Used in Calculating Thermodynamic Properties of 'FREON' Refrigerants," *Thermodynamic and Transport Properties of Gases, Liquids and Solids*, New York, 1959, p. 110.

15 "Thermodynamic Properties of FREON-11 Refrigerant," E.I. DuPont DeNemours & Co., Wilmington, Del. 1965.

16 Duffie, J. A., and Beckman, W. A., *Solar Engineering of Thermal Processes*, Wiley Interscience, New York, 1980.

17 Beckman, W. A., "Duct and Pipe Losses in Solar Energy Systems," *Solar Energy*, Vol. 21, 1978, pp. 531–532.

18 deWinter, F., "Heat Exchanger Penalties in Double-Loop Solar Water Heating Systems," *Solar Energy*, Vol. 17, 1975, pp. 335–337.

19 Chapman, A. J., *Heat Transfer*, fourth ed., Macmillan Publishing Co., New York, 1984.

20 ASHRAE Handbook 1981 Fundamentals, ASHRAE, Atlanta, GA, 1982.

21 "Installation Instructions, Refrigerant + Charger Solar Systems – Passive," Solar Research, Division of Refrigeration Research, Brighton, MI, 1984.

22 Wuestling, M. D., Klein, S. A., and Duffie, J. A., "Promising Control Alternatives for Solar Water Heating Systems," ASME Journal of Solar Energy Engineering, Vol. 6, 1983.

23 Beckman, W. A., Klein, S. A., and Duffie, J. A., *Solar Heating Design by the f-Chart Method*, Wiley-Interscience, New York, 1977.