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A COMPARATIVE ECONOMIC ANALYSIS OF STRAIGHT-THROUGH AND RECIRCULATION SOLAR HOT WATER SYSTEMS

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Abstract—A thermal and life cycle analysis was conducted to compare two types of solar service hot water systems: (1) the straight-through system with the water driven by the pressure of the mains through a heat exchanger in which it is heated by the warmer water in the solar-thermal storage tank, whenever demand for service hot water is created, and (2) the recirculation system in which a temperature-difference-controlled recirculation loop is used between the solar-thermal storage tank and the auxiliary (or backup) service hot water tank. The latter configuration which needs to have these additional components (controller, pump, and piping) and may thus also be less reliable, is more frequently used when the solar system is to supply both space heat and service hot water, because it is generally though to provide a larger fraction of the hot water from the solar source.

A computer program, combining TRNSYS with a comprehensive present-value life-cycle model and an optimization link was prepared and run for a representative year in Philadelphia to determine the thermal storage, heat exchanger, and controller-setting combination which produces the minimal present-value life-cycle costs. To provide more generality to the results, these calculations were repeated for a wide range of temperature levels and transient patterns of the solar-collector-heated water in storage and of the service hot water demand patterns, quantities, and temperatures. Apart from some uncommon cases, it was found that the straight-through solar heating system is better than the recirculation one with respect to all comparison criteria: present-value life-cycle cost, auxiliary energy consumption, and reliability.

1. INTRODUCTION

Many solar service hot water heating systems, and particularly those which are also designed for space heating, use a flow scheme similar to that shown in Fig. 1.[1,2] In essence, it consists of two separately controlled loops: one to collect heat from solar collectors and store it in the thermal storage tank, and the other to charge the service hot water tank from the storage tank. The first loop is controlled by a differential temperature controller which turns the circulation pump on whenever the collector temperature exceeds that of the water in the storage tank by a specified amount. In a similar way, in the second loop, another differential temperature controller turns that loop's pump on whenever the temperature in the storage tank exceeds that in the service hot water supply tank, which also contains the backup heat source (gas, electricity, etc.). Heat from the storage tank may also be supplied for other purposes, such as space heating. This type of scheme is referred to as the recirculation system.

A much simpler system is also used[3,4] with the cold water driven by the pressure of the mains and preheated directly by the thermal storage tank before it flows through the backup service hot water supply tank when such hot water is consumed (see Fig. 2). In such a scheme, referred to as the straight-through system, the second controller and pump are not used, and their energy consumption is thus also eliminated. Apart from these savings in capital cost and parasitic energy, it is also more reliable because it uses fewer parts. In spite of these advantages, it was generally believed by solar system designers that a smaller fraction of solar energy is used in this scheme than in the recirculation one, since the controlled recirculation loop allows a better match between the rather stochastic hot water demand and solar energy supply.

In the ongoing efforts to reduce the costs of solar energy, it is important to compare the economics of the recirculating and straight-through schemes, especially because of the popularity and near-future potential of solar water heating. Choi and Morehouse[5] have made such a comparison for the specific case of a new installation in a 15-story office building. The solar

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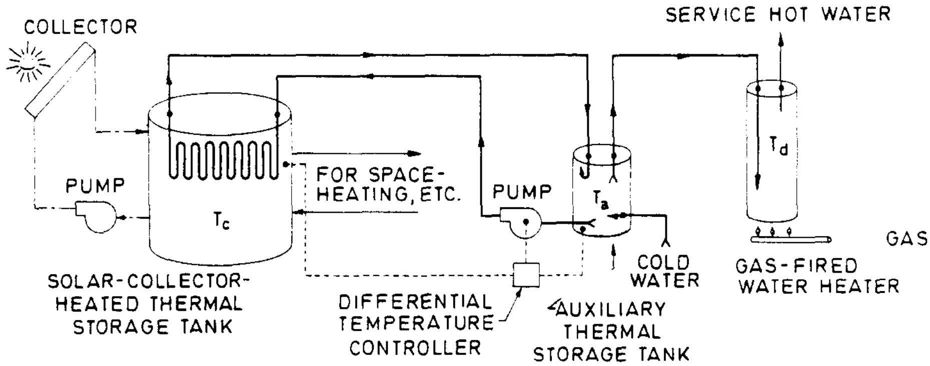

Fig. 1. Flow diagram of recirculation system.

simulation computer program TRNSYS[6] was used to compute hourly and integrated energy quantities for the entire solar system (including, apart from the hot water system, also the collectors, main storage, etc.). The results were used to compute the life-cycle costs of the two systems for this office building and it was found that the recirculating had only an insignificant ($\sim 1\%$) advantage over the straight-through system.

The present study applies a combined thermal and economic analysis to compare the present-value life-cycle costs, solar fractions, and backup energy consumption of the two schemes in a broad range of the major parameters which influence the performance. The base-case for the analysis is the solar space and service hot water system built and used in the retrofitted University of Pennsylvania row home ("SolaRow", see Refs. 3 and 7) shown in Fig. 3 (it incorporates the straight-through scheme for service hot water). The actual specifications of the system components and the measured flow rates and temperatures were used to initialize this base-case analysis. For one typical week in each of the four seasons, the weather, insolation, and thermal storage temperature based on measurements at 5-min intervals in SolaRow were condensed to create a typical 24 hr day for each season (Table 1). The typical 24 hr hot water demand schedule presented by Little[8] was used for the same day. These inputs served as the forcing functions in the base-case analysis. The condensation of the data to these 4 characteristic days was necessary because of the highly transient nature of hot water demand, which requires computation at very short time intervals. Thus, it was possible to make computations at 6-min intervals without having to incur the extremely high costs of simulation over 365 days. Such condensation of the stochastic data in solar systems was also successfully done by many others.[9]

As a second step, the volume of the auxiliary tank, the turn-on and turn-off temperature differences for the recirculating system, and the size and effectiveness of the heat exchanger for both systems, were varied to determine the configurations which produce the minimal present-value life-cycle costs. To provide even further generality to the results, a sensitivity analysis was then performed by varying the major forcing functions around the base-case values, and

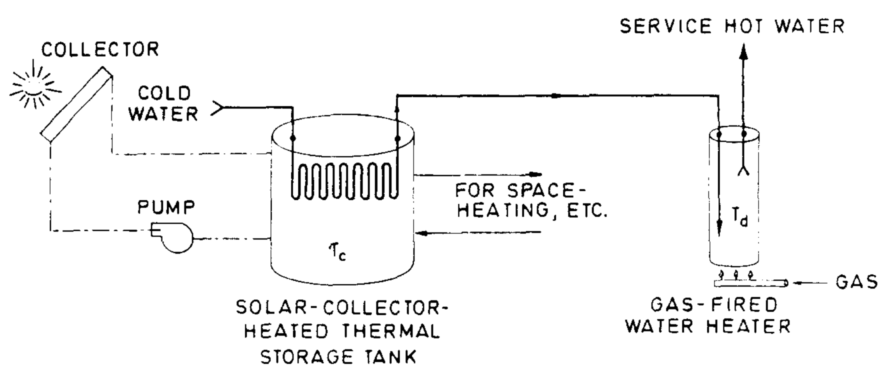

Fig. 2. Flow diagram of the straight-through system.

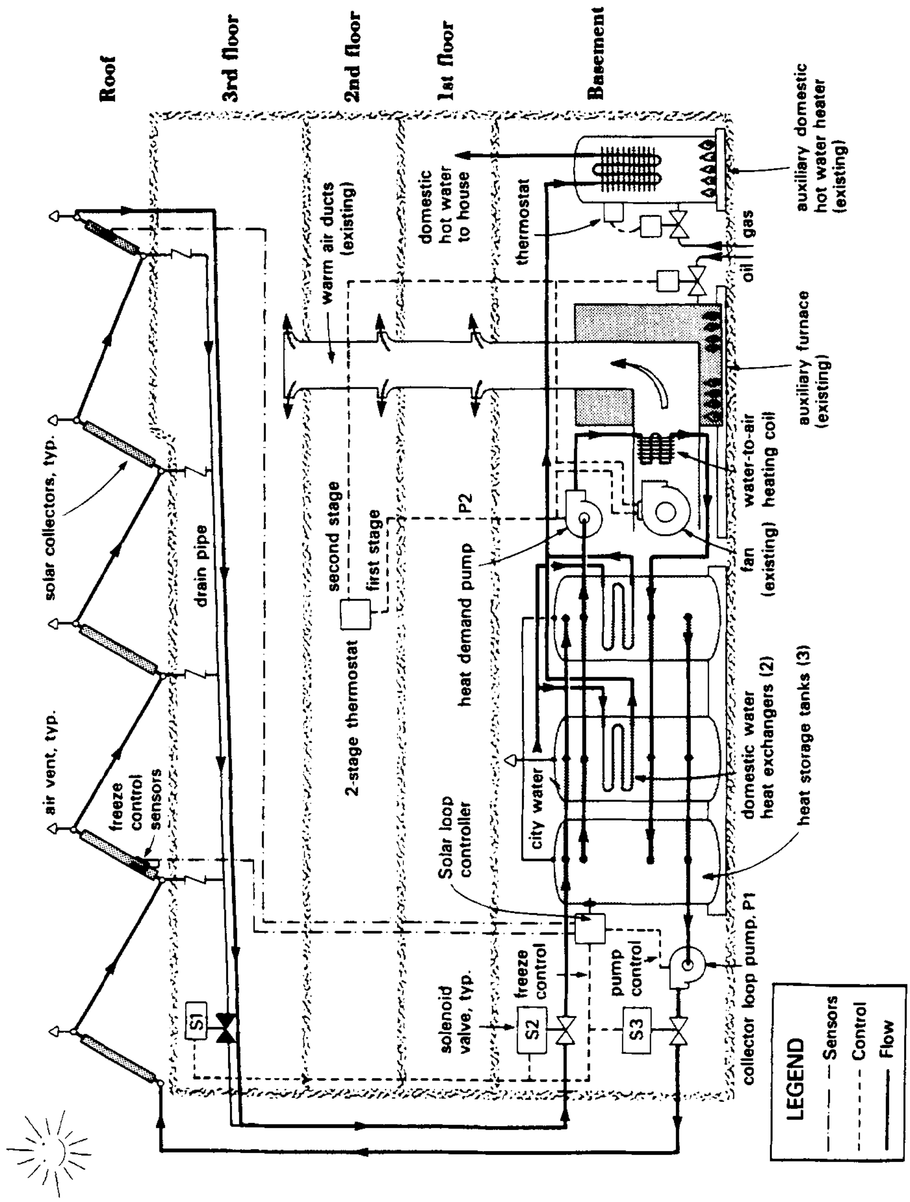

Fig. 3. The SolaRow solar and domestic water heating system.

Table 1. Ambient temperature (T_e) and insolation (I) data for a characteristic day in each season.

Hour of Day	Characteristic Day of the Month of:							
	January		April		July		October	
	T_e, °C	I, Btu/hr ft^2	T_e, °C	I, Btu/hr ft^2	T_e, °C	I, Btu/hr ft^2	T_e, °C	I, Btu/hr ft^2
1	11.6	0	23.0	0	9.6	0	3.6	0
2	12.9	0	24.0	0	9.7	0	5.1	0
3	14.2	0	25.2	0	10.3	0	5.3	0
4	15.2	0	26.5	0	12.0	0	6.2	0
5	16.3	0	27.5	0	13.2	0	8.3	0
6	17.2	0	29.1	10	14.7	41	8.8	0
7	17.9	0	29.5	45	15.4	93	9.5	5
8	17.9	21	30.4	79	16.2	142	10.7	53
9	18.0	61	29.7	107	16.7	184	11.7	98
10	18.5	94	30.2	130	16.9	217	11.8	133
11	18.3	114	30.0	144	16.0	243	11.7	155
12	18.4	121	29.0	149	14.8	244	10.6	162
13	17.7	114	27.7	144	14.1	243	9.7	155
14	16.5	94	27.3	130	13.4	217	9.2	133
15	15.9	61	26.1	107	13.5	184	8.5	98
16	14.8	21	25.5	79	13.0	142	7.8	53
17	14.2	0	24.9	45	12.1	93	6.7	5
18	13.6	0	24.4	10	11.4	41	6.1	0
19	12.9	0	23.9	0	10.7	0	5.1	0
20	12.5	0	23.7	0	10.4	0	5.5	0
21	12.0	0	23.5	0	9.7	0	4.7	0
22	11.7	0	23.3	0	9.5	0	4.8	0
23	11.7	0	23.3	0	9.1	0	4.4	0
24	11.3	0	22.9	0	9.0	0	4.8	0

determining their influence on the performance and economic results. In that context, the thermal storage tank temperature was varied around 35.5°C within ± 10°C, the temperature change pattern was varied from constant to stepwise changing between 20.5 and 50.5°C, the service hot water demand schedule was changed between the extremes of uniform rate and one where the whole demand occurs during one hour, and the total hot water demand was varied between 50 and 150% of the U.S. average value. As a consequence, this study presents results which are more general than any published on this topic so far.

2. THE HEAT TRANSFER AND ENERGY MODEL

2.1 *General information*

To determine the operation of the system and its energy performance, the computer program TRNYS[6] was used and separate programs were written to (1) compute the main storage tank temperature T_c, which is one of the time-dependent forcing functions used as input to TRNSYS, (2) link the input data set in a way which is easy to change for the optimization and sensitivity analysis, and (3) link the outputs to the economic analysis developed for this purpose and described in Section 3 below. TRNSYS[6] was chosen here because a National Bureau of Standards study[10] has shown that it can conform to the experimentally observed performance of solar hot water systems within 6%. The TRNSYS quantity integrator was used to integrate the 6-min energy quantities over the simulation period.

2.2 *The straight-through system*

The straight-through system was described by a TRNSYS "Type 5: Heat Exchanger",† linked to a "Type 4: Stratified Fluid Storage Tank with Internal Heater", corresponding to the components described in Fig. 2.

†All Type No. statements refer to TRNSYS nomenclature.[6]

The heat exhanger, consisting of two parallel spiral copper tube coils immersed in the storage tank water, was assumed in the base-case to be of constant effectiveness. That effectiveness, as determined from actual operating data was 0.96. The length l of the copper pipe needed for this submerged coil type heat exchanger was related to the effectiveness ϵ by

$$\epsilon = 1 - \exp(kl/L), \tag{1}$$

where k is the experimentally-determined heat exchange coefficient ($= -0.3107$), and L is the base-case length of 286 ft (1/2" copper).

Water is the fluid both inside and outside the heat exchanger tubes. To simulate this observed constant high effectiveness in the immersed heat exchanger, a hot-side flow rate of 1000 kg/hr was assumed. The cold-side input temperature was that of the city-water supply. A "Type 14: Time Dependent Forcing Function" was used for the cold water input flow rate, for the base-case obtained from the typical demand shown in Ref. 8.

Another "Type 14", for the hot-side input temperature to the heat exchanger, i.e., the temperature of the water in the storage tank, was obtained from a computer program written to determine the storage tank temperature changes from an energy balance affected by heat capacity of the storage, the solar-collected heat input, and the space-heat demand dictated by the house thermostat and ambient temperature. The data for the heat-loss values from the house, and for the average tank temperature, were obtained from the actual measurement of space and water temperatures in SolaRow. The ambient temperatures and insolation values used are those shown in Table 1.

The "Type 4" gas-fired hot water tank has a volume of 75 gallons (0.315 m^3), height of 1.7 m, and diameter of 0.48 m. The heat loss coefficient from the insulation is 14.4 kJ/hr°C m^2. The net maximal heat input from the gas is 55.4 MJ/hr and the tank thermostat setting is 65.5°C (as needed for washing clothes and dishes[8]). It is assumed that it operates at the national standard efficiency of 70%. Since a stratified tank is modelled, it is assumed that the warm water input from the storage tank enters at the top (TRNSYS "segment No. 1"), and that the tank thermostat is located in the top tank segment.

Amongst the output parameters, the amounts of solar heat supplied to the service hot water, and the auxiliary (gas) energy used, serve as input to the economic analysis model.

2.3 *The recirculating system*

In addition to the components used in modelling the straight-through system, the model of the recirculating system uses a "Type 3: Pump", a "Type 2: Controller", and an additional "Type 4: Stratified Fluid Storage Tank", the latter to represent the auxiliary storage tank in Fig. 1, without an internal heater. "Type 14" forcing functions are obtained and used in the same way as described in Section 2.2 above.

The "Type 4" Auxilliary Tank was assumed for the base-case to have a volume of 0.21 m^3, height of 1 m, and overall heat loss coefficient to the ambient of 2.52 kJ/m^2 hr°C. This tank allows the preheat of city water below the required delivery temperature.

The "Type 3" pump was assumed to have a maximum flow rate of 253 kg/hr ($\sim$ 1 gpm). The "Type 2" pump controller was assumed to have a "NSTK"[6] value of 4, a 2°C controller deadband (ΔT_2, sum of the offwidth and flopwidth) and an offwidth (ΔT_1) value of 1.00°C. The high temperature water input T_1 is T_c, the hot water temperature in the main storage tank, and the low temperature input T_2 is T_a, the water temperature in the auxiliary tank.

In addition to the amounts of solar heat supplied and the auxiliary gas energy used, this model also calculates the amount of electric power consumed by the pump which is rated at 200 W.

3. THE ECONOMIC MODEL

The optimization of the systems and their comparison was based on their life-cycle present value cost. In addition, the total amounts of heat delivered by the hot water system, energy supplied by the auxiliary gas-fired heater, and energy consumed by the circulation pump (in the recirculation system), as well as the annual costs of these two auxiliary energies, were computed for each case.

The present-value life-cycle model is the same as used previously by Jones and Lior.[11] The

Table 2. Capital and labor costs of modification and annual maintenance cost.

Type of Cost	System: Recirculation		System: Straight-Through	
C_i, $	Heat exchanger, 286 ft,at $0.25/ft:	71.50	Heat exchanger, 286 ft, at $0.25/ft:	71.50
	210ℓ Auxiliary thermal storage tank & piping	150		
	200W pump:	100		
	Differential temperature controller:	70		
	Two temperature sensors:	50		
	Wiring:	50		
	Total cost above that of Straight-Through System	$420		
C_ℓ,$		100		$100
C_m,$/year		20		20

[+]This length for the base-case only.

present-value life-cycle average cost per year is

$$C = \frac{1}{n}[E_1(C_i + C_l) + E_2C_m + E_{3e}C_e + E_{3f}C_f], \tag{2}$$

where n = life of system for economic analysis (years), C_i = cost of added system components, C_l = cost of labor for the installation, C_m = first-year cost of maintanence, C_e = first-year cost of electrical energy used for recirculation pump, C_f = first-year cost of gas used in backup heater. The E-terms are the appropriate economic coefficients and are calculated as shown in Ref. 11.

The costs C_i, C_l, and C_m are only for additions to the existing solar space heating system and backup gas heater, as shown in Table 2. The values of the economic parameters needed to calculate the E-coefficients in Eq. (2) are shown in Table 3. The cost of electricity was assumed to be \$0.075/kWh and of gas \$3.3/10^6 Btu. Since modification cost (Table 2) is relatively small, it was assumed that no loan was taken to finance it.

In addition to the cost C, the solar energy cost fraction, SF, is

$$SF = (C_{fl} - C_{fel})/C_{fl}, \tag{3}$$

where C_{fl} = first-year energy cost of hot water supplied by the gas-fired heater without any solar system, C_{fel} = first-year energy cost of auxiliary gas and electricity (where applicable), with the solar system installed.

Table 3. Values of the parameters used for calculating the economic coefficients E in Eq. (2). It was assumed that no loan was taken.

Life of system = 20 years	Fractional salvage value at end of equipment life = 5%
Down-payment fraction of first cost = 1.0	Annual general inflation rate = 15%
Investment tax credit fraction = 0.0	Annual discount rate (cost of money) = 15.3%
Annual property tax rate = 0.5%	Annual rate of increase of maintenance costs = 15.4%
Annual insurance cost = 0.3%	Annual rate of increase in gas costs = 15.2%
Depreciation lifetime = 10 years	Annual rate of increase of electric costs = 15.1%
	Annual incremental income tax rate = 18%

Table 4. A sample of optimization runs.

Type of System	Variable	C, $	Annual Total Fossil Energy Consumption, GJ/year	First Year Auxiliary Energy Cost $, of:		
				Gas	Electric	Total
Recirculating	ε = 97%	213.1	45.7	204.7	31.4	236.1
	94	217.8	47.2	211.8	32.2	244.0
	93	212.6	45.8	204.0	32.8	236.8
	92	215.0	46.5	207.0	33.4	240.4
	89	215.3	46.6	206.0	35.5	241.4
	80	223.2	48.7	213.1	40.7	253.8
	50	246.3	53.4	210.3	78.0	288.3
	V = 315ℓ	223.4	47.9	203.2	49.1	252.3
	250	220.4	47.5	206.1	41.9	248.0
	180	215.8	46.6	206.8	34.5	241.3
	160	218.4	47.6	213.8	31.2	245.0
	120	220.1	48.3	220.2	27.2	247.4
	80	221.2	48.9	226.2	22.8	249.0
	ΔT_2 = 5.0°C	217.4	47.2	211.1	32.5	243.6
	4.0	215.3	46.5	207.2	33.4	240.7
	ΔT_1 = 3.5°	217.5	47.4	214.2	29.5	243.7
	2.5	219.4	47.4	208.7	37.8	246.5
Straight-Through	ε = 99%	165.3	40.1	198.4	0	198.4
	96	163.5	40.1	198.3	0	198.3
	93	165.8	40.9	202.4	0	202.4
	92	166.4	41.1	203.6	0	203.6
	90	163.0	40.2	199.1	0	199.1
	88	164.9	40.8	202.2	0	202.2
	85	168.9	42.0	208.1	0	208.1
	80	167.3	41.7	206.3	0	206.3
	50	178.9	45.3	224.2	0	224.2

4. HARDWARE OPTIMIZATION

Optima were sought for 4 hardware variables: the volume V of the auxiliary tank, the turn-on (ΔT_2) and turn-off (ΔT_1) temperature differences for the pump-controller in the recirculation system, and the heat exchanger effectiveness ϵ in both systems. The optimization criterion was the present value life-cycle cost C of Eq. (2). The procedure, essentially one of pattern-search, was to compute C for several values of these variables around the base-case. In the first computation sequence, one variable at a time was changed while the others were held constant at their base-case values. Then the values of the 4 variables which produced the minimal costs C became the new base-case, and the procedure was repeated until marginal changes in the variables always produced higher costs. The initial base-case values were assumed to be as follows: heat-exchanger effectiveness = 96%, auxiliary thermal storage tank volume = 210 l., turn-on temperature difference = 2°C, turn-off temperature difference = 1°C. Table 4 represents a sample of the optimization runs and their results. The third column in the table, annual total fossil energy consumption, is the sum of the gas energy and the heat-equivalent of the electric energy consumed. For the latter, a 30% thermal-to-electric energy conversion and transmission efficiency was assumed.

To insure that local optima which differ by small amounts could be detected, the error tolerance of the computation was reduced to 0.5% wherever needed. The non-monotonic nature of some of the results can be explained by the discrete nature of the controller's operation.

The lowest values of C were obtained for the following hardware configurations: *Recirculating System*—heat-exchanger coil tube length = 72 m (236 ft), auxiliary thermal storage tank volume = 170 l (41 gal), turn-on difference $\Delta T_2 = 4.5$°C (8.1°F), turn-off difference $\Delta T_1 = 3.0$°C (5.4°F), present-value life-cycle cost C = \$212/yr. *Straight-Through System*—heat-exchanger coil tube length = 62.2 m (204 ft), present-value life-cycle cost C = \$163/yr.

Since the cost of the heat exchanger is an important factor in the optimization, which may change with time and circumstances, a separate analysis was made to determine the optimal coil length as a function of the cost of the tube. The results are shown in Fig. 4.

This phase of the optimization indicated that the straight-through system has a cost C that is 77% of that of the recirculation system. That is principally because the electrical power needed for the pump reduces the solar energy cost fraction SF [Eq. (3)] to only about 0.5% for the optimal recirculating system, as opposed to the 17.1% value for the optimal straight-through one,

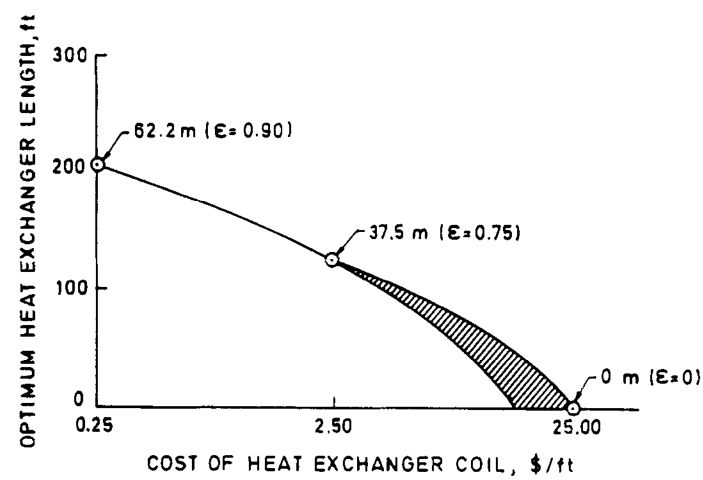

Fig. 4. Optimal length of the heat-exchanger coil.

where the latter does not even require the added $420 capital investment. Furthermore, the relatively low cost of the heat exchanger material allows energy-effective operation of the system even without the recirculation and the associated pump.

In an attempt to implement the recommended optimal hardware sizes in actual installations it is to be recognized that commercial components are sold in standard, discrete, sizes which may not match those recommended here, and that, in a commercial installer's package-deal, some of the components may be priced significantly higher then the industry-average used for estimation. The conclusions of this analysis remain, however, generally true nevertheless: careful observation of the results in Table 4 indicates that the life-cycle costs of straight-through systems are lower than those of recirculation ones, and that the sensitivity to hardware size is small for both types of systems. As for pricing, it is to be anticipated that free competition will eliminate attempts to sell components which are overpriced relative to the materials, labor, and normal profit associated with their construction and sale.

5. SENSITIVITY TO FORCING FUNCTIONS

5.1 *Range of variations*

The forcing functions for the optimization described in Section 4 above were restricted to the base-case conditions specified by Table 1 and patterns S_1 and D_1 in Figs. 5 and 6. It is noteworthy, however, that one of these forcing functions, the temperature variation pattern of the water in the solar-collector-heated thermal storage is in actual systems a complex function of the stochastic demand, weather, and insolation parameters. Another forcing function, the temperature demand schedule.and demand quantity of service hot water could be very arbitrary, but is also to a large extent under the control of the homeowner. Consequently, to provide a comparison between the recirculation and straight-through systems whose results would be more generally applicable, the initial restrictions were relaxed by repeating the cost and performance calculations in a wide range of forcing functions, as follows: (i) The average solar-collector-heated thermal storage tank water temperature was varied ± 10°C around the base-case value of 35.5°C. The calculations were thus performed for 25.5, 35.5 and 45.5°C. (ii) In addition to the base-case temperature variation pattern of the water in the solar-collector-heated thermal storage, obtained for the SolaRow conditions (Fig. 5, called pattern S_1 here), two extreme temperature patterns were used in the analysis: one that varies as a square wave (called S_2), and another where the temperature is constant, 35.5°C (S_3). These three storage temperature patterns are shown in Fig. 6. (iii) The temperature of the delivered service hot water was varied ± 10°C around the base-case value of 65.5°C. The calculations were thus performed for 55.5, 65.5 and 75.5°C†. (iv) In addition to the base-case domestic hot water demand schedule shown in Fig. 6 (called schedule D_1 here), two

†The higher temperature levels may not be applicable for domestic hot water but were used in the analysis to obtain optimization trends.

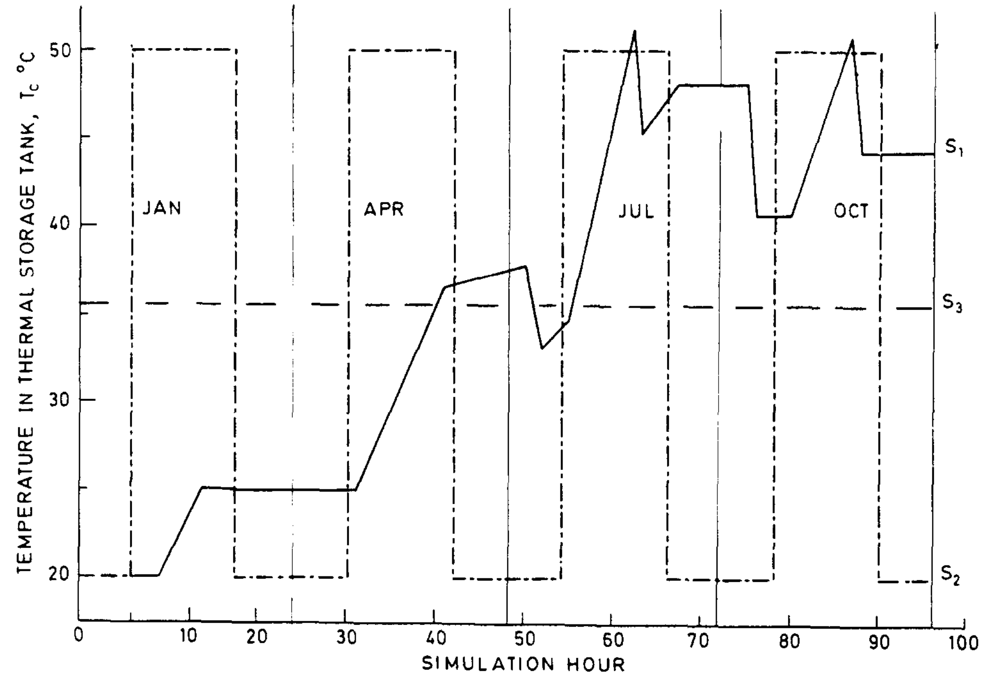

Fig. 5: Variation patterns of the solar-collector-heated thermal storage water temperature: S_1 = computed pattern for SolaRow; S_2 = square wave pattern; S_3 = constant continuous draw.

extreme schedules were used in the analysis: one that lumps the entire demand to one 30-min period at about 6:30 p.m. (D_3), and another where the demand is uniform and continuous (D_2). These schedules are shown in Fig. 6. (v) The quantity of service hot water used was varied ± 50% around the base-case value of 72 gal. The calculations were thus performed for 36, 72, and 107 gal/day.

5.2 *Sensitivity analysis results*

The calculations were performed for the optimal configuration of each of the two systems, described in Section 4, and the principal results are shown in Table 5.

5.3 *Discussion of the results*

(i) As one would expect, higher average water temperatures in the solar-collector-heated thermal storage tank reduce the amount of auxiliary energy used, and thus the present-value

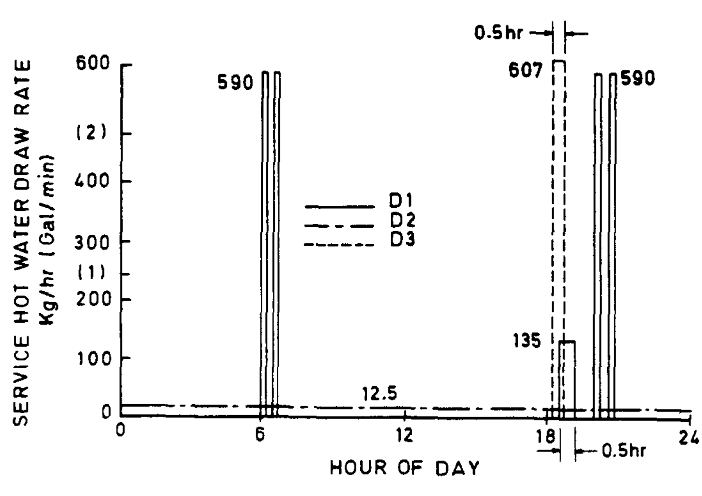

Fig. 6. Daily draw pattern of domestic hot water: D_1 = characteristic national; D_2 = uniform; D_3 = lumped at the end of the day. The duration of each of the 590 kg/hr draws is 0.1 hr.

Table 5. Sensitivity analysis results: a comparison of the effects of the forcing function on the cost and energy consumption of recirculation and straight-through systems.

Forcing Function Related to	Forcing Function	System Type	C, \$/yr.	Annual Total Fossil Energy Consumption (E_f), GJ/year	First Year Auxiliary Cost, \$/year			SF, %	Advantage of the S over the R System,‡ %, in:		
					Gas	Electric	Total (E)		C	E_f	E
Solar-Collector-Heated Storage Tank	Optimal Case†	R	212.6	45.8	204.8	32.8	236.8	0.5	30.4	13.9	18.9
		S	163.0	40.2	199.1	--	199.1	17.1			
	S_2	R	207.5	45.5	215.7	13.8	229.5	4.4	16.2	1.8	3.8
		S	178.5	44.7	221.0	--	221.0	7.9			
	S_3	R	211.0	45.8	209.3	25.1	234.4	2.4	27.6	12.0	15.9
		S	165.3	40.9	202.3	--	202.3	15.7			
	T_c=45.5°C	R	212.0	45.1	193.9	42.2	236.1	1.6	44.5	26.7	34.1
		S	146.7	35.6	176.0	--	176.0	26.7			
	T_c=25.5°C	R	213.3	46.7	216.6	21.2	237.8	0.9	16.6	1.7	4.6
		S	182.9	45.9	227.3	--	227.3	5.4			
Hot Water Demand	D_3	R	216.2	47.5	221.5	19.3	240.8	6.7	26.7	12.0	14.7
		S	170.7	42.4	209.9	--	209.9	18.7			
	D_2	R	240.5	52.4	223.5	52.1	275.6	-15.5	32.0	14.7	21.8
		S	182.2	45.7	226.3	--	226.3	5.2			
	S_2, D_3	R	222.6	49.8	238.4	11.5	249.9	3.2	10.9	-2.3	-1.1
		S	200.8	51.0	252.6	--	252.6	2.2			
	T_d=75.5°C	R	255.9	58.0	264.4	32.8	297.2	-2.5	22.8	9.0	12.8
		S	208.4	53.2	263.5	--	263.5	9.1			
	T_d=55.5°C	R	172.0	34.2	146.4	32.8	179.3	5.7	42.3	21.3	28.6
		S	120.9	28.2	139.4	--	139.4	26.7			
	Q_d=107 gpd	R	259.0	58.6	263.7	38.0	301.7	-1.1	32.3	18.1	22.8
		S	195.8	49.6	245.6	--	245.6	17.7			
	Q_d=36 gpd	R	181.7	37.3	167.7	24.3	192.0	-6.2	30.9	12.0	14.4
		S	138.8	33.3	167.8	--	167.8	6.8			

†As shown in Section 4 above.
‡R: Recirculation System; S: Straight-Through System

life-cycle cost C. The auxiliary energy used by the recirculation system is, however, reduced very slightly as the temperature increases (and significantly less than the larger decrease in the straight-through system), since the pumping power increases almost as much as the gas consumption decreases. (ii) The square-wave pattern (S_2) of storage tank temperature reduces the advantage of the straight-through system over the recirculating one, from 30.4 to 16.2%. This is principally because the recirculating system can obtain energy for service hot water heating when the collector-heated tank temperatures are highest, before that energy is lost either to space-heating or to the ambient through insulation. The case S_3, with constant tank temperature, reduced the pump energy consumption in the recirculation system, and has little effect on the straight-through system as compared to the base-case. (iii) The costs C of both types of systems rise with the service hot water delivery temperature, and the cost-advantage of the straight-through system decreases (from an advantage of 42.3% for 55.5°C, to 22.8% for 75.5°C). The reasons for the latter phenomenon are that the temperature-controlled recirculation system maintains the service hot water temperature at a higher level as the required demand temperature is increased, and that the gas consumption in both systems increases due to higher heat losses. (iv) When comparing hot water demand patterns, the lumped demand (D_3) gives the lowest advantage of the straight-through system. This is primarily because the recirculation pump has the shortest operating time. At the other extreme, the continuous hot water draw (D_2) requires the longest operation time of the pump, and results in the fact that the recirculation system's energy use costs 15% more than if the hot water supply system wouldn't have been solar at all. The continuous draw also results in the highest cost (C) straight-through system. (v) As expected, the cost of both systems increases with the daily quantity of hot water used. The advantage of the straight-through system increases with this quantity, particularly as far as energy use is concerned. That system can maintain a high solar energy cost fraction (SF) in a high-demand situation both because it doesn't require the pumping energy and because the heat losses from the tank to the environment become a smaller fraction of the total heat delivered. (vi) In search for domains in which the recirculation system may be better than the straight-through one, it was found that this may occur when the variations in the solar-heated thermal storage temperature are such that their minima are in-phase with the maxima of the hot water demand. A case similar to this is the combination S_2, D_3, shown in Table 5. The straight-through system in this case has a 1–2% higher energy cost and consumption, but it has still a cost (C) advantage of 10.9% because of the \$420 added capital investment which the recirculation system requires.

6. CONCLUSIONS

(i) The straight-through solar water heating system was found to be better than the recirculation system with respect to all comparison criteria: present-value life-cycle cost, auxiliary energy consumption, and reliability. (ii) The recirculation system may attain an advantage over the straight-through one if the temperature variations of the solar-heated thermal storage are severely out of phase with the hot water demand pattern. This circumstance, however, indicates poor system design, and should not have been incurred to begin with. (iii) Maintenance of a constant temperature in the solar-heated thermal storage tank doesn't have any significant influence on the cost C, when compared to the typical situation (in a properly designed system) where this temperature varies with heat input and demand. Consequently, no special effort (such an increasing the storage volume further) beyond the conventional practice of solar system design is necessary. (iv) For the same total amount of service hot water used, the U.S. typical hot water demand as shown in Fig. 6, results in lowest costs C as compared to the lumped and uniform demand patterns. The latter, of continuous draw of hot water over the day, is the costliest.

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