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PERFORMANCE EVALUATION OF REFRIGERANT-CHARGED THERMOSYPHON SOLAR DOMESTIC HOT-WATER SYSTEM

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PERFORMANCE EVALUATION OF A REFRIGERANT-CHARGED THERMOSYPHON SOLAR DHW SYSTEM

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ABSTRACT

Refrigerant-charged passive solar domestic hot water (SDHW) systems, which have recently become commercially available, are generating great interest in the solar community. These systems, which can be installed even in freezing climates, may achieve the high performance and reliability of direct thermosyphon systems. The Solar Energy Research Institute (SERI) is testing and analyzing a promising prototype refrigerant-charged thermosyphon system; SERI also plans to evaluate a commercially available system. The prototype was installed in a single-family residence using a stabilized R-11 as the heat transfer fluid. A system analysis was performed based on measured data. This paper discusses the analysis method and preliminary results, which indicate that there is reason to be optimistic about this type of system.

1. INTRODUCTION

Refrigerant-charged passive SDHW systems, which have recently become commercially available, are generating great interest in the solar community. These systems may attain the high performance and reliability of direct thermosyphon systems and can be operated in a freezing climate. Because they are promising, SERI is beginning a testing and analysis program that includes a prototype refrigerant-charged thermosyphon system and a commercially-available system.

The prototype system was designed and installed by the owner in a single-family residence in Golden, Colorado, in May 1980. This paper describes the system analysis and presents some initial performance results for the prototype system. More complete testing and analysis results of the prototype and the commercially available system should be available in October 1981.

2. SYSTEM DESCRIPTION REVIEW

The system, installed as a retrofit using readily available components and materials, is described in detail in Ref. 1 and is summarized here in Table 1. Figure 1 describes the system graphically, showing the heat exchanger in the storage tank located in the attic above the collectors. The system includes pressure and pressure/temperature relief valves, shrader valves for refrigerant filling and draining, and a sight glass for periodically checking the level of the refrigerant. The system is connected to an older, conventional gas water heater and can operate in solar-only, solar-preheat, and gas-only modes. Large-diameter pipe is used for the manifolds and riser pipe is used to facilitate liquid refrigerant collection and refrigerant vapor flow.

For aesthetic reasons, the tilt and orientation of the collectors are the same as those of the existing roof. The low tilt angle of the collectors will affect their output adversely during winter months because of reflective losses resulting from the oblique angle of solar radiation striking the collector glazing. Additionally, the collectors in the prototype are shaded in the late afternoons and early mornings of late fall by two deciduous trees.

R-11 refrigerant was selected for the prototype system because of its high latent heat and low vapor pressure. Although ordinary R-11 has a low thermal stability compared with R-114 and R-12, we anticipate that the stabilized R-11, used in the prototype system, will perform favorably. Samples are periodically tested by the manufacturer for signs of degradation.

3. SYSTEM OPERATION SUMMARY

Like a conventional single-phase thermosyphon system, a refrigerant-charged thermosyphon system operates without pumps, electronic controls, or solenoid valves. The latter system is inherently protected from freezing since most refrigerants freeze only at exceptionally low temperatures. System operation is based on the transfer of heat from vaporization of the refrigerant in the

Table 1. SYSTEM DATA

Collectors		Piping	
Net Area	4.0 m^2 (43 ft^2)	Material	ACR copper
Coating	Black chrome (α = .95, ϵ = .08)	Diameters	
		Top manifold	54.0 mm (2-1/8 in.)
Glazing	Low-iron glass (τ = .89)	Riser	28.6 mm (1-1/8 in.)
		Downcomer	15.9 mm (5/8 in.)
Sealant	High-T silicone	Bottom manifold	28.6 mm (1-1/8 in.)
Tubes	ACR copper		
Connections	Silver brazed	Storage Tank	
Insulation	102 mm (4 in.)	Capacity	303 L (80 gal)
Container	Galvanized steel	Construction	Glass-lined
Pressure test	2.07 MPa (300 psi) for one day	Mounted	Horizontally
		Insulation	Fiberglass, 152 mm (6 in.)
Tilt	17° from horiz.	Heat Exchanger	
Orientation	15° east of true south	Type	Coil in tank
		Material	ACR copper tubing
Refrigerant		Tubing OD	12.7 mm (1/2 in.)
Type	Stabilized R-11	Tubing length	18.3 m (60 ft)

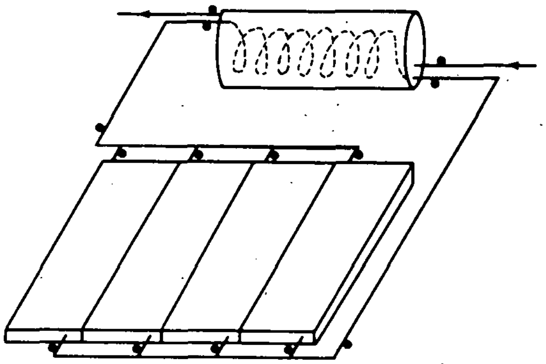

Fig. 1. System Configuration (• marks thermocouple location)

collector (thereby removing heat from the collector absorber) and the condensation of refrigerant in the storage heat exchanger (thereby giving up heat to domestic water). The refrigerant circulates because of the density difference between refrigerant vapor in the riser and condensed refrigerant liquid in the downcomer, and this circulation is probably enhanced by the pressure created in the refrigerant loop. The system is self-regulating because pressure in the refrigerant line during collection is set by and increases with storage temperature. Thus, efficiency and refrigerant circulation are probably greatest at conditions of high solar radiation and low storage temperatures.

4. SYSTEM ANALYSIS

Twenty-two copper-constantan thermocouples, with thermocouple extension wire, were used. The thermocouple ends were twisted, wrapped several times around the pipes, and attached with aluminum tape to the pipe; then the entire pipe was insulated. The thermocouples attached to the tank were also affixed with aluminum tape. See Figs. 1 and 2 for the thermocouple locations.

Our analysis is concerned with temperature trends and differences; hence, the absolute reading of the thermocouple is not critical. Temperatures were measured and recorded every thirty minutes on a data logger. A pyranometer was placed in the plane of the collectors and the total insolation was recorded at the end of each day along with daily weather conditions and the daytime high ambient temperature.

Our approach involved filling the tank with

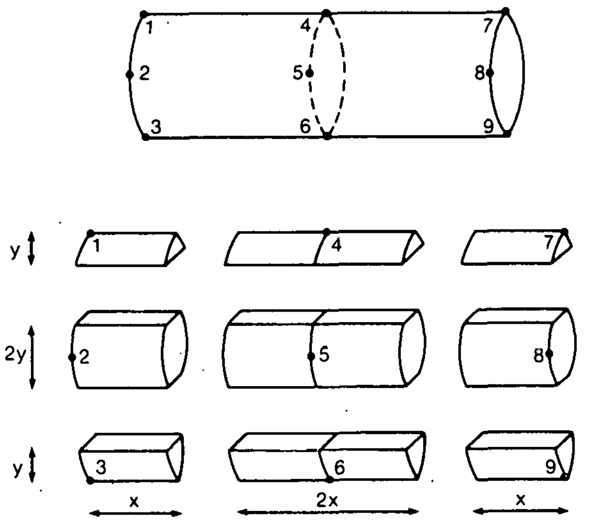

Fig. 2. Weighted Mass for Each Thermocouple

cold water and measuring the change in its temperature as a function of time without drawing water from the tank. This method allowed us to determine the change in energy in the tank over thirty-minute intervals resulting from solar gain and nighttime storage losses. Since insolation was measured on a daily basis, only a daily system efficiency was calculated.

The change in internal energy in the tank, with no load, results from solar gain and storage losses. By estimating storage losses, solar energy delivered to the tank could be estimated. First, each thermocouple was assumed to represent a range of X in the horizontal direction and a range of Y in the vertical direction from its position (see Fig. 2). Then, the equation for the change in internal energy based on the tank division in Fig. 2 and the recorded temperature is:

$$\Delta Q = C_p \rho \phi \{ \alpha [(\Delta T_1 + \Delta T_3 + \Delta T_7 + \Delta T_9)/4 + (\Delta T_4 + \Delta T_6)/2] + \beta [(\Delta T_2 + \Delta T_8)/4 + \Delta T_5/2] \}$$

where:

ΔQ = Change in internal energy of the tank during a time interval Δt

C_p = 4.187 kJ/kg°C (1 Btu/lb • F), specific heat of water

ρ = 1 gm/mℓ (62.4 lb/ft^3), density of water

α = $(-\sqrt{3} + 4\pi/3)$

β = $(2\sqrt{3} + 4\pi/3)$

ϕ = $(4x)(y^2)$, m^3 (ft^3)

T_i = Temperature at location i (i = 1 to 9)

ΔT_i = $T_i(t + \Delta t) - T_i(t)$

t = Time

Δt = 30 minutes.

Note that T_1 will represent the highest temperature in volume 1 and not the average. However, T_9 will read the coldest temperature in volume 9. These errors tend to cancel, but nevertheless will result in overestimating solar energy delivered to the tank, since temperature stratification will be greater in the top section of the tank. Temperatures T_2, T_5, and T_8 at the center of the tank represent almost 61% of the total volume, and they weight the result heavily. T_5 is the single most influential node, representing 30% of the tank volume. Temperatures T_1, T_3, T_7, and T_9 each account for less than 5% of the volume.

Storage losses could be calculated from

$$\frac{dQ}{dt} = UA \frac{dT}{dt}$$

if the temperature difference between the water volume and attic were known as a function of time and if a reliable UA (area heat loss coefficient) were known. Since both of these were unknown, the storage losses were based on the change in internal energy of the tank as a function of time during periods of no solar gain in the tank.

The heat loss over three consecutive typical cool nights averaged 325 kJ/hr (308 Btu/hr). On one particularly cloudy, rainy day (3.25 MJ/m^2•day, 286 Btu/ft^2•day) there was negligible solar gain. The storage losses during that day averaged 219 kJ/hr (208 Btu/hr), and provided an estimate of daytime storage losses. Heat loss is a function of both tank temperature and attic temperature. With increased insolation, both temperatures will increase. Therefore, our analysis assumes a daytime storage loss of 211 kJ/hr (200 Btu/hr).

In this particular system, liquid refrigerant can be trapped in the heat exchanger because of its horizontal installation. Unlike systems with vertical heat exchangers, this system can lose heat from storage at night because of the coiled heat exchanger design. As the collector temperature falls below the storage tank temperature, so does the system pressure; liquid refrigerant trapped in the heat exchanger revaporizes, drawing energy from the storage tank. The vapor then condenses in the cool collector, giving up its energy to the environment. However, the effect is slight. Assuming the heat exchanger is filled with liquid refrigerant, calculations show that the energy lost would be on the order of 1% of the total energy collected during the day.

5. RESULTS

Our analysis covered several days with no load on the tank. However, this paper will concentrate on one particular day of operation, day 61 (Mar. 2, 1981). Day 61 was hazy and partly cloudy with an insolation of 15.2 MJ/m^2•day (1337 Btu/ft^2•day) and a daytime high ambient temperature of 12.2°C (54F).

The tank temperatures shown in Fig. 3 follow a typical profile; the temperatures of the colder parts of the tank lag behind those of the warmer parts. Note that the temperature in the upper part of the tank reaches a pla-

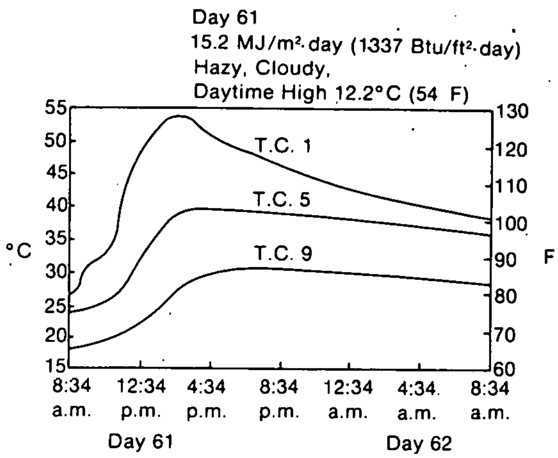

Fig. 3. Tank Temperatures

teau between 9:34 and 10:04 a.m. This is caused by a passing cloud, as shown in Fig. 4, where the collector outlet temperature drops 7°C (13F) in thirty minutes. The gain in internal energy of the tank reflects a drop for that period as well. In general, the hottest part of the storage tank (T_1) is subject to much greater swings than the coldest part (T_9). This is primarily the result of variations in the collector outlet temperatures.

A collector operating on the principle of phase change behaves much differently than does a conventional single-phase collector. On day 61, with a high ambient temperature of 12.2°C (54F), the collector operated for 3.5 hours at around 60°C (140F). Temperature differences across the collector of 30°C (54F) were not uncommon. The collector itself appears to stratify significantly as a result of the low flow rates (estimated to be less than .028 kg/s (.3 gpm)). However, the large temperature difference across the collector in this case may also be due to a low level (below the top of the absorber) of refrigerant in the collectors. The system does not operate isothermally as do heat pipe systems because of the large temperature difference across the storage tank heat exchanger and resultant sensible cooling of the fluorocarbon heat transfer fluid.

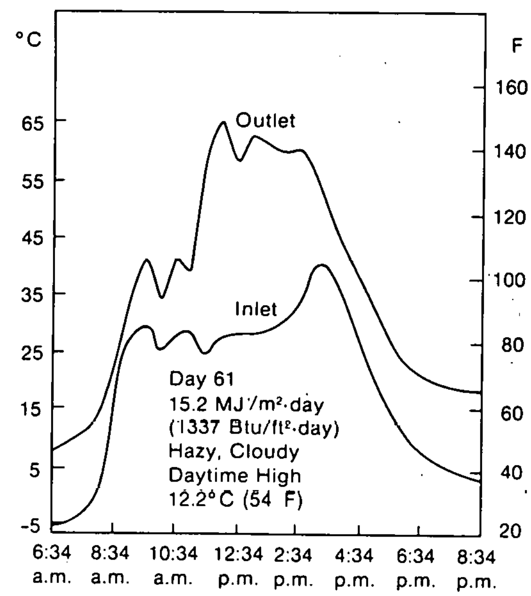

Fig. 4. Collector Temperature (West-Most Collector)

The gain in internal energy of the tank is represented by the area under the curve and above the dashed line in Fig. 5. The storage thermal loss is represented by the area under the dashed line and above the curve. The solar energy delivered to the storage tank is the change in internal energy and energy lost from the tank during the daytime.

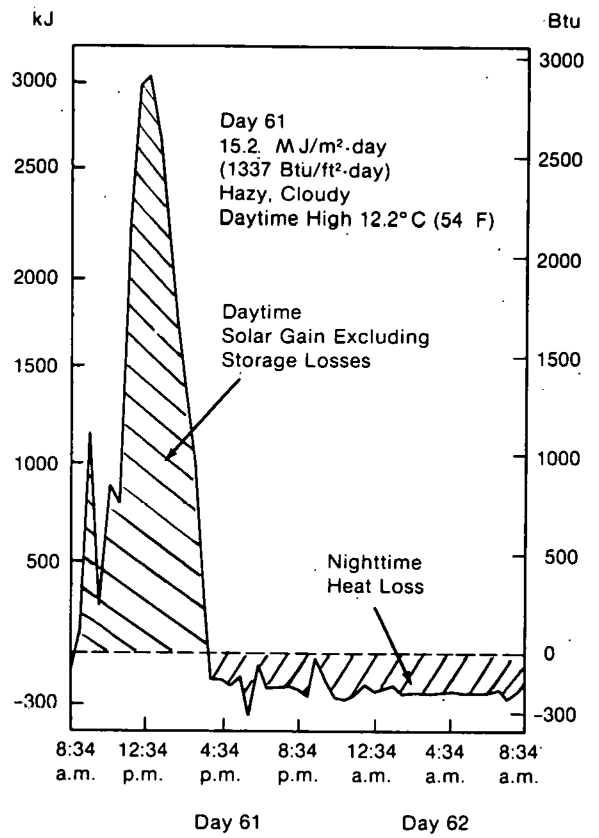

Fig. 5. Change in Internal Energy of Tank Due to Solar Gain and Heat Loss over Thirty-Minute Time Intervals

The net energy gain for day 61 amounted to 21.2 MJ (20,095 Btu), excluding storage losses, with an insolation of 15.2 MJ/m^2 (1337 Btu/ft^2), resulting in a daily system efficiency of 35%. With tank losses assumed to be 211 kJ/hr (200 Btu/hr) during the seven hours of operation, the solar energy delivered to the tank is 22.7 MJ (21,495 Btu) and results in a system efficiency of 37%. The storage loss during this period is about 7% of the solar energy delivered to the tank. Storage losses from the following night totaled 6642 kJ (6273.3 Btu) or almost 30% of the collected energy. A daytime or evening load profile would prevent much of this nighttime loss.

6. CONCLUSIONS

This type of system performs favorably compared with conventional SDHW systems which often have efficiencies of less than 3% (2). However, more detailed thermal analysis is required to better understand the operation of the system and to determine design parameters that would optimize its performance. More knowledge about boiling in the collector, vapor transport, and the heat-exchange mechanism is required. Questions need to be addressed regarding the material compatibility and refrigerant stability, as well as long-term reliability and maintenance, of refrigerant systems.

Finally, we need to know more about the cost per unit energy provided during the system's lifetime. Presently there is much reason to be optimistic about these systems, since capital and operating costs are lower than for active SDHW systems; no pumps, controls, or solenoid valves (and, hence, no parasitic electrical energy use) are required. Because fewer components are required, the system should be highly reliable.

7. ACKNOWLEDGEMENT

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