Advanced Solar Domestic Hot Water Systems

A Report of the Task 14 Advanced Solar Domestic Hot Water Systems Working Group
October 1996

INTERNATIONAL ENERGY AGENCY
Solar Heating and Cooling Program

IEA Solar Heating and Cooling Programme

The International Energy Agency (IEA) was established in 1974 as an autonomous agency within the framework of the Organization for Economic Cooperation and Development (OECD) to carry out a comprehensive program of energy cooperation among its 24 member countries and the Commission of the European Communities.

An important part of the Agency's program involves collaboration in the research, development and demonstration of new energy technologies to reduce excessive reliance on imported oil, increase long-term energy security and reduce greenhouse gas emissions. The IEA's R&D activities are headed by the Committee on Energy Research and Technology (CERT) and supported by a small Secretariat staff, headquartered in Paris. In addition, three Working Parties are charged with monitoring the various collaborative energy agreements, identifying new areas for cooperation and advising the CERT on policy matters.

Collaborative programs in the various energy technology areas are conducted under Implementing Agreements, which are signed by contracting parties (government agencies or entities designated by them). There are currently 41 Implementing Agreements covering fossil fuel technologies, renewable energy technologies, efficient energy end-use technologies, fusion technology and energy technology information centers.

The Solar Heating and Cooling Programme was one of the first IEA Implementing Agreements to be established. Since 1977, its 21 members have been collaborating to advance active solar, passive solar and photovoltaic technologies and their application in buildings.

Australia	Finland	Netherlands	Turkey
Austria	France	New Zealand	United Kingdom
Belgium	Germany	Norway	United States
Canada	Greece	Spain	
Denmark	Italy	Sweden	
European Commission	Japan	Switzerland	

A total of 22 Tasks have been initiated, 17 of which have been completed. Each Task is managed by an Operating Agent from one of the participating countries. Overall control of the program rests with an Executive Committee comprised of one representative from each contracting party to the Implementing Agreement. In addition, a number of special ad hoc activities--working groups, conferences and workshops--have been organized.

The Tasks of the IEA Solar Heating and Cooling Programme, both completed and current, are as follows:

Completed Tasks:

Task 1	*Investigation of the Performance of Solar Heating and Cooling Systems*
Task 2	*Coordination of Solar Heating and Cooling R&D*
Task 3	*Performance Testing of Solar Collectors*
Task 4	*Development of an Insolation Handbook and Instrument Package*
Task 5	*Use of Existing Meteorological Information for Solar Energy Application*
Task 6	*Performance of Solar Systems Using Evacuated Collectors*
Task 7	*Central Solar Heating Plants with Seasonal Storage*
Task 8	*Passive and Hybrid Solar Low Energy Buildings*
Task 9	*Solar Radiation and Pyranometry Studies*
Task 10	*Solar Materials R&D*
Task 11	*Passive and Hybrid Solar Commercial Buildings*
Task 12	*Building Energy Analysis and Design Tools for Solar Applications*
Task 13	*Advance Solar Low Energy Buildings*
Task 14	*Advance Active Solar Energy Systems*
Task 16	*Photovoltaics in Buildings*
Task 17	*Measuring and Modeling Spectral Radiation*
Task 20	*Solar Energy in Building Renovation*

Current Tasks and Working Groups

Task 18	*Advanced Glazing Materials for Solar Applications*
Task 19	*Solar Air Systems*
Task 21	*Daylight in Buildings*
Task 22	*Solar Building Energy Analysis Tools*
Task 23	*Sustainable Solar Buildings: The Optimization of Solar Energy Use in Larger Buildings (Project Definition Phase)*
Working Group	*Materials for Solar Thermal Collectors*

Task reports and ordering information can be found in the IEA Solar Heating and Cooling Programme publications list. For additional information contact the SHC Executive Secretary, Pamela Murphy Kunz, Morse Associates Inc., 1808 Corcoran Street, NW, Washington, DC 20009, USA, Telephone : +1/202/483-2393, Fax: +1/2021265-2248, E-mail: 103116.1530@compuserve.com.

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This report documents work performed within the IEA Solar Heating and Cooling Program
Task 14: Advanced Solar Energy Systems
Working Group: Advanced Solar Domestic Hot Water Systems

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Task 14 - Advanced Active Solar Systems

Task 14 was initiated to advance the state-of-the-art in active solar energy systems. Many features developed during the few years before the start of the Task, when used alone or in combination, had the potential to significantly improve the performance of these systems. It was the objective of Task 14 to analyze, design, evaluate and, in some cases, construct and monitor a number of different systems incorporating one or more of these features.

The work of the Task was divided into three Working Groups, based on the type of systems studied, and one Sub Task dealing with dynamic testing. The goal of the Working Groups was to facilitate interaction between participants with similar projects. Participants were able to identify and address issues of common interest, exchange knowledge and experience and coordinate collaborative activities.

Domestic Hot Water (DHW) Systems - Working Group

The focus of this Working Group was the development of advanced DHW systems using the "low flow" concept. Participating countries contributed expertise related to different system components. The collaborative work in the Task brought this expertise together to allow participants from each country to design systems which show a significant cost/performance improvement (as high as 48%) over systems on the market in their respective countries when the Task began.

Air Systems - Working Group

Task work concentrated on further development of a commercially available concept for the preheating of ventilation air in industrial and commercial buildings. This concept is a specially designed cladding system to capture the air heated by solar radiation on the south wall of a building. Four projects, two in Canada, one in the USA and one in Germany, were constructed using a perforated version of the wall. The German project adapted the concept to preheat combustion air for a district heating plant. The practical work of these projects was complemented by theoretical work conducted at the University of Waterloo in Canada and the National Renewable Energy Laboratory (NREL) in the United States. Task work demonstrated that the cost/performance of the perforated wall is over 35% greater than earlier versions of the design.

Large Systems - Working Group

The Task also examined large scale heating systems involving temperatures under 200°C. Five large systems were studied. They were all very different but each represented important applications of active solar systems. District heating, the subject of the Swedish project, can be used in most IEA member countries to provide space and water heating for communities. The

German project also involved district heating but with no storage. A tulip bulb drying installation in The Netherlands explored the staggered charging and discharging of long term storage, a strategy which may fmd many uses, especially in agricultural applications. Solar desalination, the subject of the Spanish project, has wide application in water starved areas of the world and could represent a major export opportunity for IEA countries. Industrial process heat was represented by a project in Switzerland. Since virtually all large systems are custom designed, cost/performance improvements for this Group was not a meaningful measure of achievement. Documentation of lessons learned is the most important product of the work.

Dynamic System Testing Sub Task

The work of this Sub Task within Task 14 provided a continuation of work completed earlier by the IEA Dynamic Systems Testing Group. That Group established that dynamic fitting was a suitable tool in processing laboratory tests and in-situ monitoring of solar domestic hot water systems. The objective of the new sub-task in Task 14 is the continued development and evaluation of dynamic testing of solar energy systems, subsystems and components for prediction of long term system performance from short term tests.

Task 14 activities began in 1989 and were completed in 1995.

The following countries participated in this Task:

Canada	The Netherlands	Switzerland
Denmark	Spain	United States
Germany	Sweden	

1. EXECUTIVE SUMMARY

The Task 14 Advanced Solar DHW Working Group set a goal of a greater than 15 percent increase in the cost and performance of solar DHW systems over current practice. This goal is interpreted as achieving designs that have an initial cost to annual energy delivered ratio improvement (dollars/GJ) greater than 15 percent.

Actual cost performance gains ranged from 20-48 percent. These gains were a result of multiple improvements in heat exchangers, storage design, modularization, absorbers and piping.

Because regulations and practices regarding the design and construction of solar DHW systems differed markedly from country to country, it was not possible to propose one universal Task 14 system. Instead, each country developed its own individual "Dream System." In order to measure how well the goal was achieved, one of the most commonly available systems being sold in each country at the time the Task began was selected as a comparative "Base Case."

Despite this lack of commonality, most specific system design features and components could still be made applicable to each country's improved designs. Thus the Working Group's common efforts were focused on compiling and developing design features and components which would improve solar DHW system performance and lower system cost. In this regard, a system design approach termed "low or matched flow," was determined to be the most promising direction for improvements. Thus, from the beginning, Task Working Group efforts were directed primarily toward low-flow design elements.

Many Working Group developments have been implemented by solar industry in several countries. The Dream System of Switzerland and Denmark are currently being commercialized.

Before discussing the Dream System of each country and comparisons with the Base Cases, this summary will address design features and components that were identified by the Working Group to provide improvements in either cost, performance, or both.

1.1. Collector and Load

Often in comparing high- and low-flow designs it was found that good practice in a low-flow design was good practice in a high-flow design. For example: 1) The use of current improvements in top insulation was not cost-effective in either low- or high-flow collectors. 2) When a typical daily load profile was used to size the system for the load, both the low- and high-flow systems showed about the same degree of sensitivity to variations in both daily load profile and day-to-day loads. Variations in the daily load profile had only a small effect on system performance. Task investigations indicate a somewhat greater, but still small, effect for day-to-day load variations. There was some evidence that a larger solar storage would increase annual performance somewhat. Further study in this area is warranted.

For low-flow systems, the following load matching principles should be followed:

- The flow in the collector loop should be approximately 2 to 4 grams/sec-m2.

  Flow into the solar storage or integral heat exchanger design should be such that optimal stratification is maintained.

  Total flow volume through the collector for an average day should be matched to the volume supplied to the load for an average day.

  The collector and load flow rates should be optimally matched.

Since loads and ambient conditions of Task 14 countries are different, application of these principles will result in different optimized designs for each country.

The Task found that absorber design improvement was one area where collector costs can be reduced. And, low flow provides some of the opportunities for absorber cost reduction. Though most current well designed high-flow collectors also perform well in low-flow systems, lower collector cost can be obtained by an absorber optimized for low flow. Costs of low-flow fm-tube absorbers can be reduced substantially by reducing the amount of material that is necessary for the tubes and fins.

Serpentine flow configurations are desirable for low-flow systems since there is a potential for uneven flow distribution in riser/header configurations. Riser/header configurations can be used, but care needs to be exercised in design and construction, especially with horizontal risers, to insure even flow distribution.

Both drainback and glycol/water closed-loop systems can be used for low-flow collector freeze protection. In serpentine drainback systems, a five degree minimum slope, in piping is needed to assure complete drainback.

1.2. Solar Storage, Heat Exchanger, and Auxiliary

The main performance advantage of low-flow systems is due to extensive thermal stratification in solar storage. Solar storage design and the design and interaction with storage by heat exchangers and auxiliary system can effect stratification. Therefore, all three of these components are key components in low-flow systems and they are often considered together as a solar storage system.

These three components in combination with the fluids used are the elements most profoundly affected by differences in regulatory issues and design practices among different countries. For example, some countries have only small manufacturers of DHW tanks and therefore these tanks are relatively expensive as solar storages. In these countries it is more likely that you will find a built-to-order optimized solar storage in a DHW system, rather than a solar storage made by incorporating less than optimum modifications into a standard available DHW tank. In countries with a few large manufacturers of DHW tanks, the opposite is true.

It is likely that less expensive solar storages will be developed based on standard DHW tanks in more countries or that new storages based on system designs that can make use of inexpensive materials, like a cheap unpressurized plastic tank for a drainback system, will eventually emerge.

An optimum solar storage system should have the following characteristics:

- The volume of a tank reserved for solar storage (not auxiliary) should be sufficiently large, depending on solar fraction and economics.
- Temperature differences in the tank should be equalized as slowly as possible.
- The capacitance of the collector side heat exchanger should be sufficiently large, about 50 W/K-m2.
- The storage should be carefully insulated and thermal bridges, such as pipe connections, should be avoided in the upper part of the tank.

Several solar storage systems were evaluated including a mantle tank, side arm heat exchangers, built in helical heat exchangers, stratification manifolds, tank in tanks, two tank systems, internal auxiliaries, and external auxiliaries. Of the several low-flow system storages experimentally evaluated, there was little difference in thermal performance at high solar fractions. Therefore, cost considerations should predominate in selection of storage system type. Only at lower solar fractions, on the order of 20-30 percent, did performance differences become significant.

1.3. Pump and Controller

Though many solar DHW systems take advantage of thermosyphoning in various ways, most require a collector circulation pump. Several classes of small pumps (centrifugal, positive displacement, and thermal self-pumping) were investigated. None of these had a thoroughly acceptable blend of cost, performance, and durability.

A small light weight high speed electronically driven centrifugal pump with the requisite characteristics (called the Task 14 pump in the Dream Systems specifications) is being developed by a Task participant. High durability was gained by keeping the pump simple and shifting most of the pump complexity to the silicon chip. The pump provides the required flow rates for low-flow systems and sufficient start-up pressure for operating drainback systems. The design provides low operating cost with a target power consumption of five watts and can potentially be manufactured, given sufficient sales volume, at a cost lower than that of current competing pumps.

To optimize storage stratification, proportional control of collector flow rate is needed to provide low-flow systems with a fixed delivery temperature equal to the load temperature. Photovoltaic powering of the pump is highly desirable as it can provide a proportional control that can be integrated into the pump itself. However, cost needs to also be considered.

Overheat prevention and, in the case of drainback, freeze protection are other functions of the solar energy system controller.

1.4. Piping

Low flow makes possible compact all-in-one solutions to piping choice, such as having both collector supply and return tubes and control sensor wiring in one envelope. The smaller diameter piping that can be used in low flow also opens possibilities for use of flexible non-metallic materials or easy to bend copper tubing.

Long material lifetime is required in a solar energy installation and therefore the following durability requirements should be noted:

- Piping and insulation must be resistant to temperatures up to 200°C and pressures up 4 bars.
- Piping must be resistant to deterioration by a water-glycol mixture.
- The envelope, insulation, and/or piping must be resistant to ultraviolet radiation.

This approach has many cost and performance benefits, such as:

- Installation of piping and electrical wiring is fast and easy, lowering installation costs.
- Heat losses from the smaller diameter piping and insulated envelope are reduced by a factor of two or more.
- Cost of piping and insulation materials can be reduced by minimizing piping diameter and wall thickness.
- Delivery and handling costs are reduced.

Disadvantages of this approach can be:

- The piping bundles can only be used for smaller solar low-flow DHW installations.
- Some bundle designs have shown a tendency to be damaged during installation.
- There may be a higher pressure drop with the smaller piping diameters.

- There may be additional increases in pressure drop if the piping is bent in a tight radius during installation.

- Too small piping diameters may prevent proper draining in drain-down systems. Problems may occur for inner diameters less than 10 mm.

  There may be a greater risk of a blockage in the collector loop with the small piping diameters.

1.5. Other Low-Flow Considerations

In current practice, lowered cost is the most apparent benefit of the low-flow approach. Performance increases of two to nine percent which were due solely to low flow were measured in two Working Group systems that were not specifically designed for low flow. Over the long term, larger performance increases seem probable for low-flow systems by properly integrating components that have been optimized for maximum system performance in low-flow use. Additional work is warranted here.

1.6. Dream Systems

The Dream Systems of the six Working Group countries are shown in Figures 1-1 through 1-6. As may be seen, there are many common elements, such as piping and sensor wire bundles, combined solar and auxiliary storages, and tank-in-tank storages. Many of the systems use the Task 14 pump. There are also differences which reflect both local regulations and practice, as well as individual preferences.

Table 1-1 provides a summary of Base Case and Dream System cost, performance, and cost to annual energy delivery ratio for each country, as well as the location and ambient conditions on which each country's performance estimates are based. Cost reductions, performance increases, and improvements in the cost to annual energy delivery ratio are also shown. As can be seen, each country has exceed the 15 percent goal.

Significantly, two of the Dream Systems will be introduced as commercial products by the time the Advanced Solar DHW Working Group activities are complete.

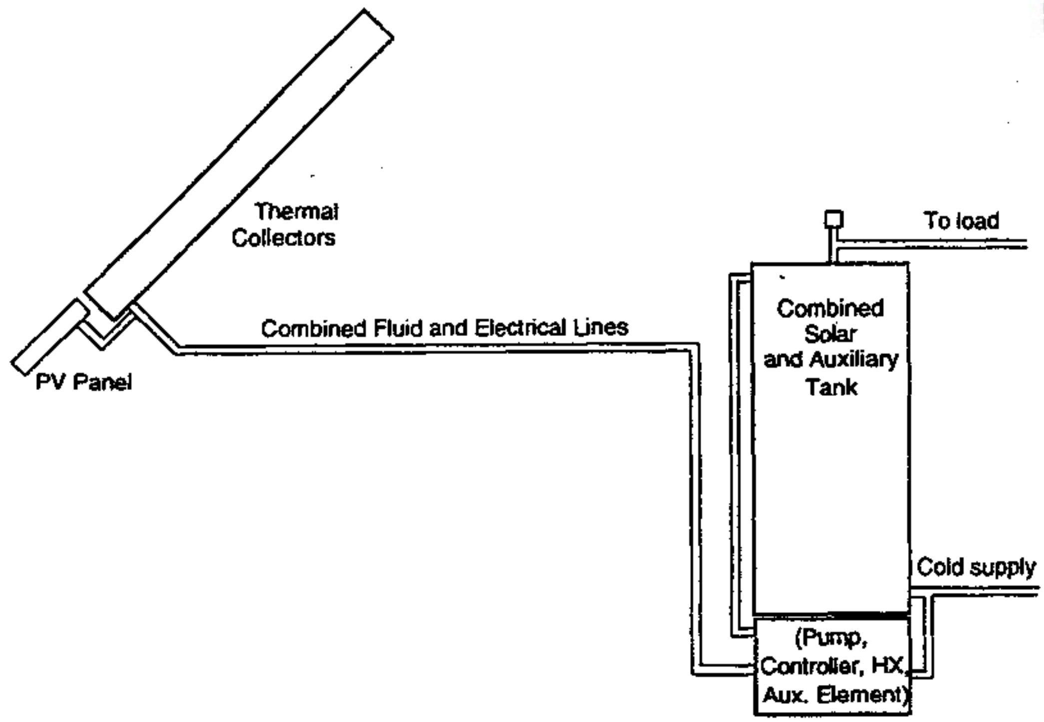


Figure 1-1. Canadian Dream System Diagram.

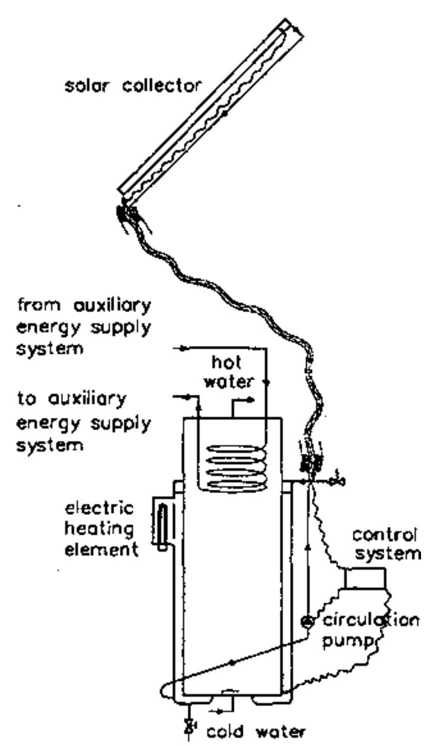


Figure 1-2. Danish Dream System Diagram.

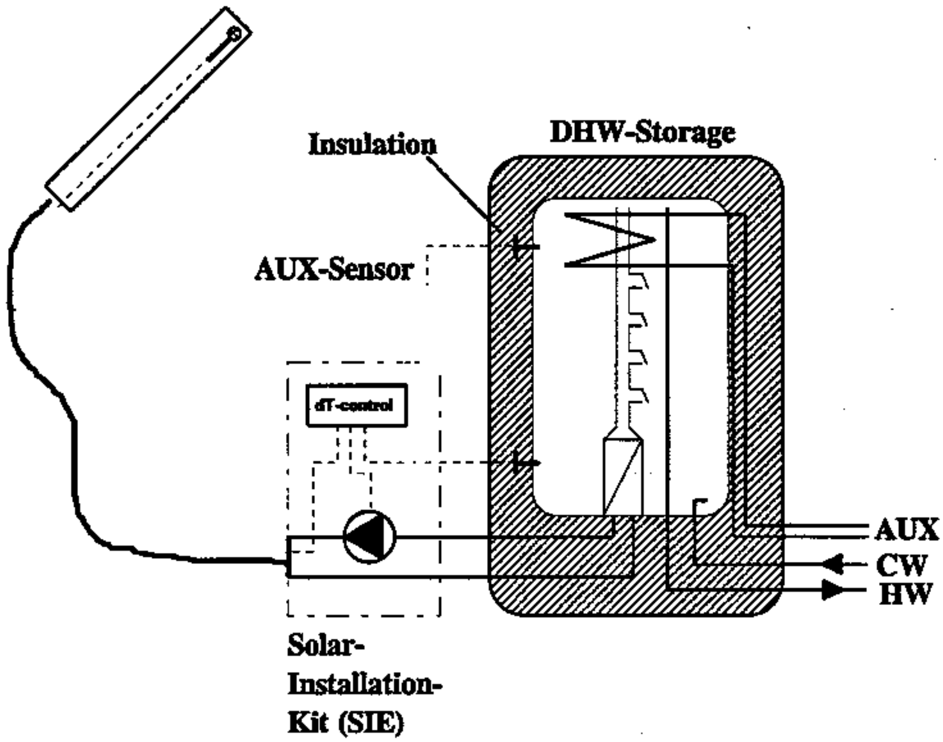


Figure 1-3. German Dream System Diagram.

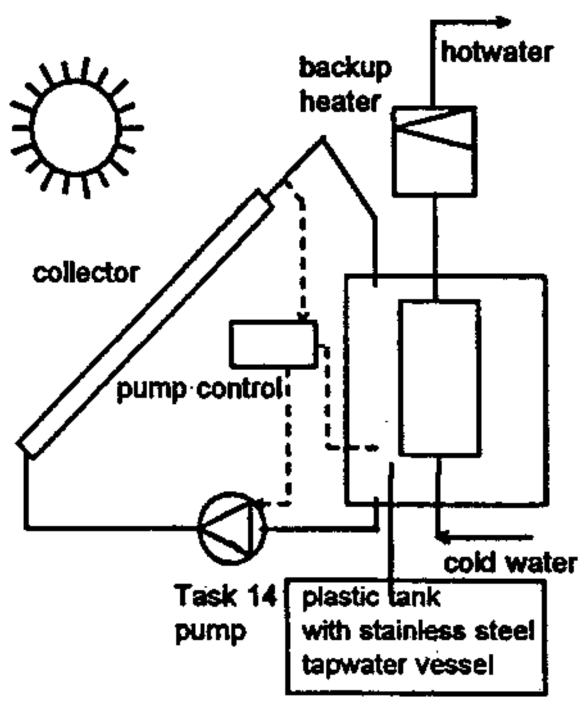


Figure 1-4. The Netherlands Dream System Diagram.

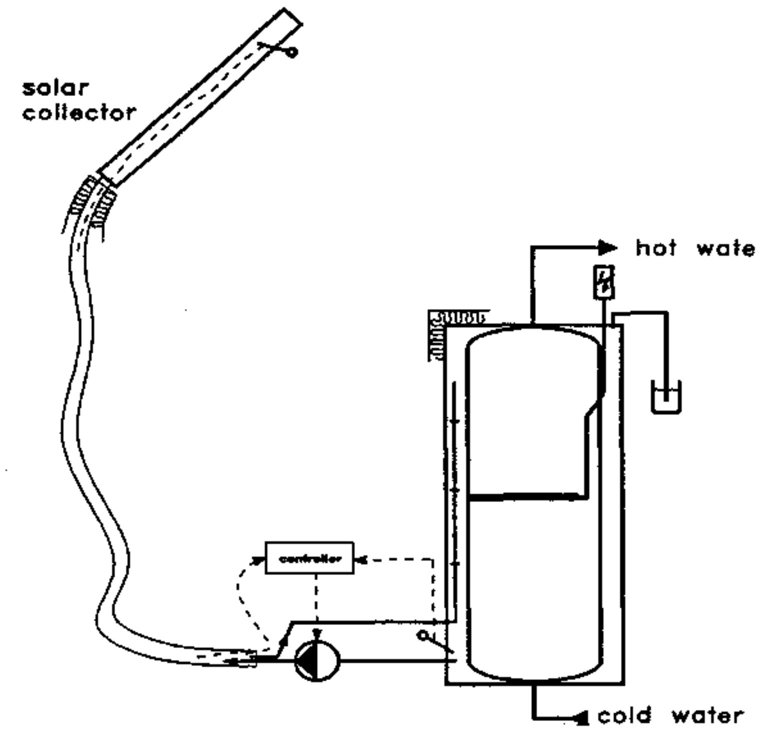


Figure 1-5. Swiss Dream System SOLKIT®.

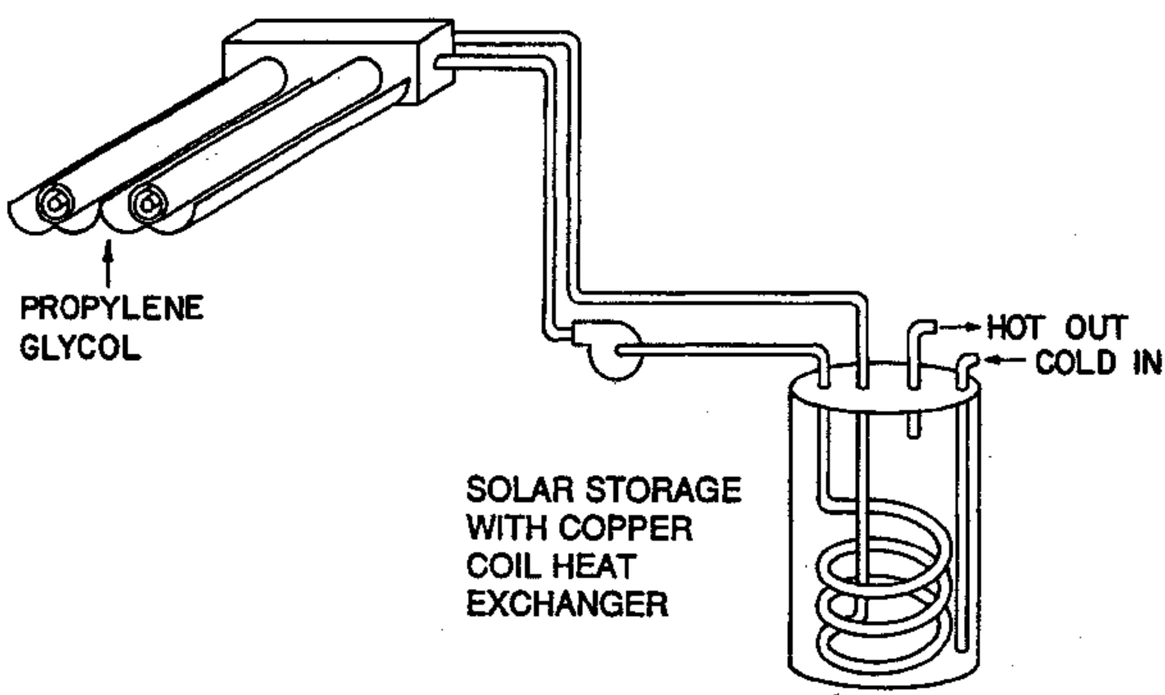


Figure 1-6. United States Dream System for Freezing Climates.

Table 1-1. Costs, Performance, and Comparisons.

Subject	Canada	Denmark	Germany	Netherlands	Switzerland	United States
Reference quantities for calculations						
Location	Toronto	Copenhagen	Hannover	DeBilt	Kloten 1986	Sacramento, CA
Radiation on collector aperture GJ/m²-yr	5.48	4.262	3.808	3.989	4.5	7.497
Monthly average daytime temperature °C	9 (-6 to 22)	8.1	8.7 (0 to 17)	9.5 (2 to 18)	8.6 (-1 to 19)	16 (7 to 24)
Exchange rate in US$ and basis date	0.87$ 1/94	6.70 DKK 3/94	1.7 DM 4/94	1.86 Df 5/94	1.437 sFr 1/94	1.00/$ 12/93
Base Case manufacturing costing approach and Dream System differences	1993 fabrication, market, methods, and prices; design for automation	1994 conditions	1994 conditions	1989 market and fabrication methods at 1994 prices	Base Case costing approach	1989 market and fabrication methods at 1993 prices
Base Case cost (1993 US$)						
Total*	1862	3098	6608	1985	7134	1925
Operating and maintenance $/yr	< 10	15-22	51-131	17	84-150	10
Base Case performance						
Thermal (Q_{load} - Q_{aux}) GJ/yr	8.7	5.07	6.55	3.70	7.2	7.05
Reliability and Durability	good to excellent	no problems	excellent	no significant problems	same as ordinary water heaters	excellent
Dream System cost (1993 US$)						
Total*	1445	1892	5393	1540	4466	1510
Operating and maintenance	< 5	15	37-117	11	74-140	10
Dream System performance						
Thermal (Q_{load} - Q_{aux}) GJ/yr	12.9	5.04	6.65	4.16	7.2	8.51
Reliability	excellent	freezing problems	excellent	improved	improved	excellent
Cost/performance comparisons						
Cost reductions $	417 (22%)	1206 (39%)	1215 (18.3%)	445 (22.6%)	2667 (37%)	415 (21.6%)
Energy delivery increases GJ/yr	4.2 (48%)	-0.03 (-1%)	0.1 (1.5%)	0.46 (12.4%)	0 (0%)	1.46 (20.7%)
O&M improvements $	slight	~5 (~25%)	~14 (~15%)	6 (35%)	10 (10%)	0 (0%)
Base Case $/GJ/yr	214	611	1009	563	990	273
Dream System $/GJ/yr	112	375	811	370	620	177
Cost/energy delivery improvement $(GJ/yr)	102 (48%)	236 (39%)	198 (19.6%)	193 (34%)	370 (37%)	96 (35.2%)

* This is not the end price to the user. Total does not include marketing, selling and distribution costs. The values in this table do not include the past consequences of higher production volumes and improved installation approaches. See Appendix A for further details.

2. INTRODUCTION

This is the final report of the International Energy Agency (IEA) Solar Heating and Cooling Program Task 14 Advanced Solar DHW Systems Working Group. The Working Group is made up of experts from seven countries: Canada, Denmark, the Netherlands, Germany, Spain, Switzerland, and the United States. Since its start in 1989, the Working Group has been led by the United States.

Since participation of the solar industry was an important planned feature of Task 14, each country sent an industry representative and researcher to the Working Group meetings.

The Working Group's goal was a fifteen or greater percent system cost/performance improvement compared to existing state-of-the-art systems in common use in 1989. The Working Group achieved this goal through lowered costs and increased performance of the system and its components as compared to current practice.

This report is designed to make it easy for a solar equipment manufacturer or marketer to locate information on a particular system or component, including associated cost and performance data, and evaluate how that information may be of benefit.

The Solar DHW Systems Working Group chose to focus its activities on low-flow design, since this approach was judged to hold the greatest promise for near-term performance improvements and cost reductions. The Working Group low-flow activities continued the promising low-flow research and development direction started in the late 1970s and early 1980s by a number of researchers, most notably by Terry Hollands [2-1] and Chris van Koppen [5-1].

Canada, Denmark, Germany, the Netherlands, Switzerland, and the United States participated in the low-flow activities. Denmark, Germany, the Netherlands, and Switzerland conducted extensive side-by-side experimentation of state-of-the-art reference and advanced low-flow DHW systems. Results of these activities may be found in [4-1, 4-6, 5-3, 5-8, and 7-1].

The Netherlands, Spain, and the United States also chose to identify a second path and examined the integral collector storage DHW system. The integral collector storage DHW system holds significant promise for cost performance improvements. This alternative was not explored substantively because priority was given to low-flow.

Each of the seven countries followed different paths to accomplishing the Working Group goal. Each followed various mixtures of system modeling, system testing, system improvement, and component improvement.

Prior research and concurrent research from outside the Working Group were incorporated into the systems of the Working Group when appropriate. Much research and development work generated or stimulated by the Advanced Solar DHW Systems Working Group activity is still ongoing.

As Working Group efforts matured, interaction among the participants evolved the concept of an universal "Dream System." The Working Group soon realized that each country's notion of a Dream System was different because each country's interpretation depended on a unique set of national circumstances, involving regulations, market conditions, the structure of the solar industry, energy policy, component prices, solar design approaches, and traditions. Thus, each country evolved its own "Dream System."

Effects of extraneous factors were explicitly avoided when assessing the value of Working Group accomplishments. This was accomplished by having each country define a "Base Case" that could be compared to its Dream System. Each country selected as its Base Case a solar DHW system typical of those that existed in the country in 1989-90 as the work of the Working Group began. A consistent approach was then used to estimate costs and evaluate the performance of both the Base Case and Dream System.

As the work of the Advanced Solar DHW Systems Working Group progressed, a number of heat exchanger/storage designs were identified as promising low-flow components. In the later stages of the Working Group activities, two of the most promising designs were singled out to be experimentally evaluated in the highly controlled environment of Canada's National Test Facility solar simulator. A series of experiments provided a comparison of the two point designs in a low- and a high-flow mode. This experiment substantiated the advantage of using low flow for the given two systems.

3. JUSTIFICATION FOR LOW FLOW

3.1. Introduction

Over the past 10 or 12 years, the designers of small solar systems, primarily domestic water heaters, have come to realize that lowering the collector loop fluid flow rate (hereafter abbreviated to "low flow") can improve system cost effectiveness. A significant part of this understanding has come about through five years of discussion and study within Task 14.

Though the low-flow strategy typically lowers the cost of the system, the degree of performance enhancement depends very much on the base design chosen for comparison. It is generally agreed that tank thermal stratification is the major contributor to better performance. High-flow systems can have varying degrees of stratification, depending on aspects such as whether there is a heat exchanger, and if so, its design and location. Particular types of exchangers, such as the internal, full-height mantle or spiral, generate gentle, natural convection in the tank with minimum mixing (i.e. plume entrainment), and give some stratification even at high collector flow. Side-arm heat exchangers can minimize plume entrainment using particular auxiliary input and pump control strategies. However, there may be further performance benefits to be gained through a fully integrated low-flow system design.

The low-flow regime can be characterized as follows. "Single pass" is a reference to the quantity of fluid flowing through the collector loop being equal to the load. For typical collectors, this will either be a rate in the range of 2 to 4 grams per square meter-second (water equivalent) or that the total of the collector flow (as water) over the day equals the storage tank volume. If the tank volume equals the daily load (draw-off), then these two are equivalent. High-flow rates have been 5 to 10 times higher than this range.

3.2. Low-Flow Cost Impact

Lower collector flow rates have some immediate and longer-term cost advantages. Most directly, the pump can be made smaller and less expensive, and consume less electricity. Also, the piping to the collectors can be of smaller diameter. This makes it more flexible, easier to install, and less expensive. Smaller tubes lower the thickness, and cost, of the insulation because the R-value is dependent only on the ratio of the insulation's outer-to-inner diameters, not the absolute thickness. Of course, the thinner overall diameter further reduces stiffness. All of this adds up to significantly less piping installation time and costs.

In the longer run, new lightweight, low-flow absorber designs could further reduce the system cost. Since they would also improve performance, they are discussed below.

3.3. Low-Flow Performance Impact

It may be possible to develop lighter weight absorbers having somewhat higher thermal performance. In an overall system design emphasizing low-flow and low pump power, the flow in the absorber tubes should be laminar. It is well known that in fully developed laminar flow the heat transfer rate to the fluid in a length of tube is independent of diameter, and so a smaller bore tube and a narrower fin will have a higher fin effectiveness. Alternatively, the fin can be made proportionately thinner while maintaining the original fm effectiveness. If the tube bore is much smaller than, 8 mm, the two-collector serpentine configuration becomes more difficult to manage with low power pumps, because of excessive hydraulic pressure drop. It then may become advantageous to switch to a parallel riser and horizontal header design. The flow velocity in each vertical tube is low enough to allow natural convection to help to assure uniform flow across the collector, assuming of course that the cool fluid inlet is at the bottom header. The vertical risers will also improve the collectors' drainback capability. Of course, the smaller fin-tubes imply a larger number of tubes for a given size of absorber, and increase the amount of labor needed to assemble it, unless the manufacturer is able and willing to invest in some degree of automation. The choice between the larger tube serpentine and smaller tube parallel configurations is thus very dependent on the costs to each manufacturer in his local environment and at a given production volume.

In the near term, low flow allows existing absorber products, such as copper/aluminum fm-tube, to be connected in a serpentine pattern in the collector without significant hydraulic or thermal penalties. Two large serpentine collectors connected in series (doubling collector pressure drop) plus the losses of the connecting piping, could make the total loss too high for a very low power pump, even under low-flow conditions. However, it may be relatively inexpensive to optimize the bores of both the collector and interconnecting tubing to keep the pump power low enough.

Parallel connected collectors with serpentine absorbers would result in a lower pressure drop but with perhaps poorer heat transfer to the slower fluid, unless the absorber tube bore was reduced.

3.4. Low Flow, Tank Stratification, and Performance

Although some high-flow designs give some degree of tank thermal stratification, low flow will further enhance its usefulness via three effects:

First, the charged tank will be stratified more sharply, making more of the energy in the tank available closer to the desired load temperature. This will increase the solar fraction.

Second, starting the day with a partially charged tank, during subsequent hours of charging, low flow will provide higher water temperatures at the top of the tank. Clearly, high flow from the heat exchanger at the bottom of a cold tank will not deliver water to the top of the tank at a sufficiently high temperature. If a draw must be made this early in the charge cycle, water heated by auxiliary energy must be available somewhere in the system. So low flow will lead to faster recovery for small, but hopefully usable, volumes of hot water. Depending upon the high flow draw profile chosen for comparison, low flow might offer a higher solar fraction by minimizing auxiliary input to these early draws. It is to be noted, though, that variations in the low flow draw profile itself have little effect on low-flow system performance.

Third, for storage tanks with internal auxiliary heaters occupying a top fraction of the tank, excessively strong mixing due to high collector flow rates or high local tank velocities may allow auxiliary heat to reach the solar heat exchanger, and hence, pass that heat to the collector inlet and reduce collector efficiency.

3.5. System Design Considerations

Most important, the tank must be thermally stratified, with the top of the solar portion close to the desired load temperature. Whatever mechanism is used to add heat to the tank, there should be as little mixing as possible. As a corollary, the auxiliary input should be provided so as not to interfere with the operation of the solar part of the tank.

The collector flow rate should be such that fluid is always delivered to the tank at temperatures commensurate with the desired load temperature, while considering the current level of insolation. The best algorithm to control this flow is not yet known, but low fixed-flow works quite well if attention is paid to plume entrainment in the tank. (Better combined solar/auxiliary algorithms could almost eliminate entrainment.)

Too small a heat exchanger will raise both the collector supply and return temperatures, even with an adequate level of collector flow. And if there is mixing with colder water in the tank or in a tempering valve installed at its outlet, either the collector must run hotter or more auxiliary energy must be added to achieve the desired water temperature. These last two effects both lose energy at the hotter collector, create entropy by lessening availability, and so demand more auxiliary energy to make up for it.

4. COMPONENT REPORT: COLLECTORS, ABSORBERS, AND LOADS

4.1. Absorber/Collector

4.1.1. Introduction Low-flow collectors will be designed to deliver temperatures close to the delivered load temperatures. The main operating parameters which distinguish low-flow collectors from high-flow collectors are determined by flow configurations and hydraulics in the absorber tubes. There has been much debate over the way these parameters would influence the overall efficiency of a solar system using the low-flow/matched-flow principle. Specially designed low-flow collectors have been introduced in Canada, Denmark, the Netherlands and Switzerland.

Existing solar collectors can be used for low-flow solar heating systems. Danish investigations [4-1, 4-2] show that the efficiency of Danish solar collectors used for traditional high-flow solar heating systems is not significantly influenced by a reduction of the flow rate. Therefore solar collectors currently marketed can be suitable both for low-flow solar heating systems and for traditional high-flow solar heating systems.

Basic information which was available before Task 14 work began was obtained through two studies conducted at the University of Waterloo in Canada. These studies showed the advantages of material reduction in general for absorbers used under low-flow conditions [4-3]. The studies also demonstrated that drastic reductions in absorber material can be made and that absorber fins have an optimal thickness profile of zero at the tip and their maximum thickness at the base [4-4].

Information obtained on solar energy systems by Task 14 and numerous other studies have provided good insights into the effects of the above mentioned parameters. The product development and manufacture of high-performing, low-flow collectors may now result in a lowered product cost compared to the previous generation of collectors.

Computer models to determine the collector efficiency factor F' for sheet and tube solar collectors use the following expression:

$$F' = \frac{1/U_L}{W\left[\dfrac{1}{U_L[D + (W - D)F]} + \dfrac{1}{C_B} + \dfrac{1}{\pi D_i h_{fi}}\right]}$$

(from Duffie and Beckman [4-5, page 2711). The dependencies in this expression on flow rate are a subject of the Task 14 Dynamic Testing Subtask. A simpler analysis can be carried out by considering the dependence of the heat-removal factor F_R on the flow-rate. This analysis can be performed without a complicated computer model.

4.1.2. Design Guidelines The absorber design for low-flow conditions must be optimized for a typical flow rate and heat-removal factor. This will lead to an optimal absorber design. Design options must be evaluated with respect to manufacturing possibilities and material availability and cost. At some point, thinner material may get more expensive than thicker material, while efficiency changes are minimal.

Under steady-state conditions, the dependence of the heat-removal factor on the flow rate is determined by the equation:

$$F_R = \frac{Q_{gain}}{\tau\alpha\ Q_{solar} = U_L(T_i - T_{amb})} = \frac{\dot{m}C_p}{A_c U_L}\left[1 - \exp\left(-\frac{A_c U_L F'}{\dot{m}C_p}\right)\right]$$

(from Duffle and Beckman [4-5, page 277]).

The fin thickness in relation to the inner diameter of the tube is determined by theoretical optimization and technical limitations in the manufacturing technique. Studies conducted at Waterloo University have determined that the fins do not necessarily need to be rectangular in shape. A step change in fin thickness, so that the fin gets thinner as it is farther from the tube, permits a reduction in material content. Roll-form manufacturing processes, like Sunstrip®, can achieve this type of material reduction [4-6]. A Swiss design (2-shaped tube), combines roll-form and welding techniques in order to optimize the material content in the absorber.

The choice of serpentine or header/riser absorber configurations is determined by a number of factors:

- System design (drainback or closed-loop);
- Velocity in the tubes dependent on tube diameter; and,
- Equal flow distribution in the header/riser configuration.

In general, it is believed that horizontal riser/vertical header construction creates a disadvantage in low-flow conditions because of the difficulty in maintaining equal flow distribution for horizontal mounting. Flow distribution in vertical riser/horizontal header constructions is not a problem because of natural convection

The serpentine configuration requires special consideration in drainback systems in order to allow the tubes to drain completely. When designing low-flow absorbers, these conditions need further investigation. A Dutch study [4-7] demonstrated that a low-flow serpentine absorber with 6 mm ID tubes was still able to drain completely, provided the absorber is mounted at least at a 5° angle to the horizontal.

4.1.3. Test Results A Dutch investigation of four different low-flow serpentine absorbers showed comparable results [4-7]. All absorbers performed almost equally, as expected, under high-flow conditions. However, variations occurred at low-flow conditions below 6 percent.

4.1.4. Insulation The effects of top insulation on the collector were investigated by a Canadian group [4-8]. The study showed a slight change in performance if the top (hot side) of the collector is insulated better than the bottom. In general, the change in performance is considered modest and the study results do not favor investment in thicker insulation materials for the top of the collector. Extra insulation is recommended only if it requires minimal time and cost expenditures.

4.1.5. Conclusions If we consider the effects of the collector and the absorber in relation to a low-flow situation, there is very little evidence that improvements in collector design (apart from the absorber) are cost effective. On the other hand, an absorber designed especially for low-flow conditions is highly advantageous. Drastic material reductions can be accomplished with the absorber. Fin and tube absorbers are preferable for low-flow applications due to their strong potential in reducing the material content. It is believed, from a practical point of view, that serpentine configurations are more reliable than header/riser constructions, since the flow distribution pattern in the absorber is critical under low-flow conditions.

Horizontal mounted serpentine absorbers, used for drainback systems, should allow a slope of the tubes of a minimum 5° angle to drain the tubes completely.

4.2. Load Influence

4.2.1. Introduction The principles involved when using systems with a low-flow collector loop to a heat exchanger/tank are:

- Low flow in the collector loop (approximately 2-4 grams/sec-m2);
- Optimal stratification in the tank;
- Total volume through-put for the collector on an average day equals the total average load in such a day; and,
- Optimization of the flow rate for a specific collector.

Variations in the load and the effects on the system efficiency have been the subject of several previous studies.

One problem is the lack of consistency in the daily load. It is unknown how the individual loads in a household will differ from the original design specifications for a system. Since systems will be designed for the "average" load, variations in each individual household will exist. There is a need to gather more information on the effects of load variations on system performance.

Since the basic principle assumes a match between the load and the total flow through a collector, one can understand that variations in the load on a day-to-day basis would affect the efficiency of the system if flow is kept constant.

These effects were studied by TNO-NL and the United States in [4-9] and [4-10]. The TNO-NL study indicated that variations in load pattern over the day, with a constant collector flow, showed no significant difference between the thermal performance of low-flow and high-flow systems.

The reference load pattern throughout all of the countries involved in Task 14 are different. This implies a system design which will be optimized on the specific average load pattern in each country.

4.2.2. Load Profiles In the studies, three types of analyses have been carried out:

- Variations in the yearly draw with a constant daily load and profile;
- Variations in the daily draw with a constant profile, obtained with a random generator so that the yearly load is comparable with that for a constant daily load; and
- Variations in the daily draw by fixed typical loads for different days so that the load for the week is equal to the average.

4.2.3. Rationale The effects on the yearly system efficiency will be limited to certain periods throughout the year. Typical solar hot water systems are designed to supply enough hot water for a household during the summer. In many cases, the yearly solar fraction will be between 50 and 75 percent. This means that there will be a need for auxiliary heating in the winter. The most critical periods, therefore, are the spring and autumn when the system could on some days meet a 100 percent solar fraction (like in the summer), and on others require auxiliary heating.

Since the effects of load profile on system efficiency are primarily of concern during the autumn and spring, one can rationalize that the effects of load variation are limited to roughly half the year. This, of course, will limit the effects on a yearly basis.

4.2.4. Results The Task 14 studies demonstrate that variations in the load have an effect on the daily efficiency of the system compared to the "average" design load. However, varying the flow rate in the collector loop to achieve a better matched flow may not significantly affect performance. In other words, if the collector loop is designed to operate under optimal low-flow conditions, the effect of the load on a day-to-day variation (both in profile and in total draw-off) is likely to be small.

4.2.5. Conclusions This study concludes that variations in load pattern have a minimal effect on the yearly efficiency. However, it is important to choose an optimal flow rate for a specific system and corresponding solar fraction. The solar fraction relates to the storage volume. A storage volume larger than the daily load will make the system less sensitive to the load and will lead to a higher performance. An economic evaluation should be made to match the extra storage cost to the higher performance.

The fact that the optimum collector flow rate is relatively insensitive to variations in the load and profile is very important for practical applications. A solar energy system, once tuned to the optimum collector flow, is unlikely to need adjustment to maintain high performance when the draw changes.

5. COMPONENT REPORT: HEAT STORAGES, HEAT EXCHANGERS, AND AUXILIARIES

5.1. Introduction

Work on low-flow solar heating systems has been carried out at universities and research institutes in various countries since 1979 [5-1].

The main reason for the thermal advantage of low-flow solar heating systems is the extensive thermal stratification inside the heat storage during the operation of the system. The thermal advantage of the system increases with increasing thermal stratification in the heat storage. The mechanism that transfers heat from the solar collector fluid to storage should therefore ensure maximum thermal stratification. Further, the storage design should ensure that temperature differences are equalized as slowly as possible.

The heat storage, the collector side heat exchanger, and the auxiliary energy supply system are therefore key components for low-flow systems.

The suitability of differently designed heat storages, heat exchangers and auxiliary energy supply systems are described in this section.

5.2. Market and Regulatory Issues in Participating Countries

Regulatory issues concerning hot water tanks and design traditions differ between countries. In addition, in some countries few manufacturers of hot water tanks exist while in other countries many manufacturers are marketing hot water tanks.

Therefore, the designs of standard hot water tanks and standard solar tanks vary from one country to another. Short descriptions of market and regulatory issues in the participating countries follow.

5.2.1. Canada The majority of solar water heating systems in Canada consist of a solar preheat tank connected to an electric auxiliary water heater. Electric water heater tanks are widely available at a low cost and are therefore predominantly used for the solar preheat tank.

5.2.1.1 Tank design.

Canada are generally dictated by requirements specified by the Canadian Standards Association (CSA). The following are noted:

- Construction: Tanks are typically of glass-lined steel construction with anodic protection and include thermal insulation and outer metal jacket. Nominal capacities are 175 and 270 liters. A hydrostatic pressure test to 2.1 MPa is required, in addition to other structural tests. Tanks must be installed with a 98°C/1.0 MPa temperature/pressure relief valve.

- Diffusion Ratio: The tank design must provide means to minimize mixing of the inlet water with water stored in the tank. The diffusion ratio, as determined by test, requires at least 90% of the tank capacity to be delivered before the water temperature drops more than 17°C.

- Energy Efficiency Requirement (Standby Loss): The standby energy loss of tanks ranging in sizes from 50 to 270 liters shall not exceed the standby loss as calculated by the following formula:

Standby Loss (Watts) = 61 + 0.20 Volume (liters)

5.2.1.2. Heat exchanger. The use of standard electric water heater tanks for the solar preheat tank dictates the use of an external collector side heat exchanger. The most common external heat exchanger is a copper shell and coil, single-wall design with thermosyphon operation on the potable water side.

5.2.1.3. Heat transfer fluid.
propylene glycol and distilled water. The propylene glycol is typically Dowfrost HD which includes additives for corrosion protection at high temperatures (up to 165°C).

5.2.2. Denmark Two types of hot water tanks are commonly used: A hot water tank with a built-in heat exchanger spiral and a hot water tank with a mantle welded around the surface of the tank. Solar collector fluid is circulated through the heat exchanger spiral or the mantle.

The auxiliary energy supply system, either an electric heating element or a heat exchanger spiral, is normally built into the top of the tank. Therefore, one tank provides storage for the solar heating system and the auxiliary energy system.

For systems with a single separation between the solar collector loop and the public water supply, an approved solar collector fluid must be used. If pure water or BP Termovæ ske S is not used, an approved tracer must be added to the fluid. At present, the following heat transfer fluids and tracers are approved:

Heat transfer fluids: Water and propylene glycol.

Tracers: Brilliant Blue, Green S.

The solar collector loop is normally a pressurized loop with a security valve opening at 2.5 bar.

The minimum material thickness of the tank S_{min} is normally determined by the equation:

$$S_{min} = \frac{0.11 \cdot D_y \cdot \sqrt{k \cdot p}}{100} \text{ mm}$$

where D_y is the outer diameter of the tank in mm, k is a constant determined as the ratio between the modulus of elasticity of steel at 20°C and the modulus of elasticity of the tank material at the maximum tank temperature, and p is the design pressure in bar equal to 16 bar.

Hot water tanks are normally made of steel St 37-2 or stainless steel.

The hot water tank and any heat exchanger spirals in the tank must be protected against corrosion. If St 37-2 steel is used for the tank and the spiral material, both are normally enamelled. Tanks with enamelling are equipped with an anode. Alternatively, steel tanks can also be protected against corrosion by means of coating with an approved synthetic material. At present only rilsan coating is approved.

A shut-off valve, a one-way valve, and a safety valve must be installed on the cold water inlet pipe to the tank.

At present, all marketed heat storages are tested at the Danish Solar Energy Testing Laboratory. Thermal characteristics of the heat storage are measured. A data sheet for each heat storage is prepared. The data sheet includes: The heat storage capacity, the thermal loss coefficient of the heat storage and the heat exchange rate.

5.2.3. The Netherlands Both traditional and solar domestic hot water production must comply with regulations as formulated in Dutch working documents from VEWIN (association of water authorities in the Netherlands).

These working documents are presently being reformulated. The new documents will include a section on solar hot water systems. It is expected that the new working documents will be finished in 1995.[1]

The present working document VEWIN WB 5.4b states the following:

> "Hot water apparatus using indirect heating sources must use a double-wall heat exchanger between the heat transfer medium and the drinking water."

As a result of these regulations, water authorities will generally approve use of drinking water from solar energy systems, using a single-wall heat exchanger if they operate under a pressureless condition.

Any addition to the drinking water is prohibited. Recently one water authority allowed addition of a glycol solution with an ATA approval. However, this is disputed by other water authorities, especially since the pressure in the system is not controlled.

'Available from: KIWA n.v.; Certification and Inspection, Sir Winston Churchill-laan 273, P.O. Box 70, NL-2280 AB Rijswijk, the Netherlands, Phone: + 31 70 395 3477, Fax: + 31 70 395 3420

Present solar systems are developed based on these working documents, resulting in drainback systems filled with potable water, in a closed loop.

The majority of hot water tanks are made of copper. For solar tanks, 316 Ti stainless steel the predominant choice, although a few glass-lined tanks are on the market.

5.2.3.1. Conclusion.

as a circulating fluid are currently unresolved, the potential use of these solutions in the future is uncertain. At the present time, regulations prohibit their use with a single-walled heat exchanger. Therefore, water-filled drainback systems or ICS systems which use potable water in the storage are the only systems allowed on the market.

5.2.4. Spain Solar hot water systems in Spain utilize one of three types of tanks: tanks with an external jacket around a part of the surface (with or without an electric heater inside the mantle), tanks with a built-in heat exchanger spiral, and tanks without any exchanger element.

The tanks must be manufactured in accordance with the Regulations of Pressurized Equipments, Instrucción Téc nica Complementaria MJAP11. They must be tested with a pressure double that of the working pressure of the tank, and must be approved by Ministerio de Industria y Energia.

The technical specifications of collector fluids and tanks are as follows:

Potable water is commonly used in the solar collector loop. In some cases, additives are used depending on climatic conditions and the kind of water. In places without any risk of freezing, only water or demineralized water with anti-corrosives can be used. In places with freezing, demineralized water with antifreeze and nontoxic corrosion inhibitors are used. The commonly used antifreeze is propylene glycol.

Spanish tanks are typically constructed of:

- Galvanized steel for any size
- Stainless steel
- Vitrified steel for small sizes (with anodes for cathodic protection)
- Copper

Tank insulation materials must provide thermal conductivity less than 0.52 W/mK and temperature resistance higher than 80°C. The minimum thicknesses for insulation are 30 mm for less than 300 ℓ and 50 mm for more than 300 In case of outside tanks bigger than 2,000 a minimum thickness of 100 mm is required.

The hot water inlet from the solar loop is located at the top of the tank, except in tanks with an electric element located at the top in which the inlet is always below the auxiliary volume. In systems where the heat exchanger is a built-in helix, the helix is located in the lowest part of the tank.

5.2.5. Switzerland

5.2.5.1. Tank concept.
with a 400 to 500 ℓ volume. The heat exchanger spiral is located in the lower part of the tank and an auxiliary energy system is located in the middle of the tank.

In addition to the SDHW systems, systems are often combined with space heating. More than half of the systems are tank-in-tank designs, where a DHW tank is incorporated into a larger tank for space heating. Typical volumes are 200 to 400 ℓ hot water tanks in 1,500 to 3,000 ℓ
tanks of water for space heating.

5.2.5.2. Tank design.
water tanks. The responsible organization Schweizerischer Verein der Gas und Wasserfachleute (SVGW) has the authority to test new products before they can be sold on the market. The maximum test pressure is 12 bars and the maximum pressure under operation is 6 bars. Corrosion protection is not incorporated into the test procedures. Cold water inlet equipment is similar to that on non-solar tanks, usually consisting of a shut-off valve, a non-return valve, pressure reduction including a filter (from 6 bars mains pressure to 3 bars tank operation pressure), and a safety valve.

5.2.5.3. Auxiliary energy supply.
lower electricity prices during night hours. A number of systems with an oil- or gas-fired furnace have a second heat exchanger spiral in the upper part of the tank, in addition to the electrical heating element, to supply auxiliary heat during the winter.

5.2.5.4. Collector loop.
water-glycol mixtures. All of the components, such as the pump, expansion vessel, security valve (3 bars), etc., are similar to ordinary heating systems.

5.2.5.5. Heat transfer fluid.
by use of water-glycol mixtures. A number of water-glycol products are marketed by different producers such as Hoechst or BASF etc. (Single-walled heat exchangers are allowed and there is no restriction as to the use of either propylene- or ethylene-glycol.)

5.2.6. United States There are two types of solar storage tanks commonly used in the United States. Both tanks are commercially available and are made by one of the country's largest hot water heater manufacturers. The primary reasons for the use of these tanks are cost and immediate availability.

5.2.6.1. Tank design. Commercial tanks are glassline steel with a volume of 200 to 400 ℓ with optional top electrical heating elements. While both tanks appear identical, one tank has a wrap-around heat exchanger made of copper which is 40 to 50 meters long. This tank can be used in either a closed-loop glycol or drainback system. The tank without the heat exchanger is the most common tank found in the United States. Most are open-loop systems located in non-freezing climates. This tank is also used for side-arm heat exchanger systems and drainback systems, which have separate drainback tanks. These storage tanks are tested to 30 bars and have an operating pressure rating of 15 bars.

5.2.6.2. Heat exchangers. The United States uses all types except the mantel and in-tank heat exchangers. The main reason for not using the in-tank or mantel design is that one code listing group, I.A.P.M.O. (International Association of Plumbing and Mechanical Officials), which is strong in the western United States, requires double-wall, vented heat exchangers for any potable-, non-potable transfer. While industry has repeatedly requested allowance of non-toxic fluids, such as propylene glycol, to be used with a single-wall exchanger, I.A.P.M.O. has resisted any change.

5.2.6.3. Heat transfer fluid. United States systems usually use propylene glycol with a closed-loop design and demineralized water with a drainback design.

5.3. Thermal Performance of Low-Flow Systems with Differently Designed Heat Storages

Heat storage types used in small, low-flow systems employ different heat exchange principles for transferring heat from the solar collector fluid to the domestic water. The auxiliary energy supply system, which heats the water to the required temperature, can also be designed in different ways. Consequently, system types with several designs can be used as low-flow DHW systems.

The thermal performance of the various system types depends on the design of the system. Consequently, before the desirability of each system type is judged, the design and operation mode must first be optimized. This process will result in optimum designs which differ between countries, since the system costs are highly influenced by regulatory issues, common practices, and so forth.

Thermal performance of the system is influenced by the design of the auxiliary energy supply. Therefore, the thermal performance of each system is presented both with and without top-heating by an auxiliary energy supply.

In Denmark, the thermal performance of top-heated systems has been investigated at the Thermal Insulation Laboratory [5-2], [5-3]. The results of these investigations are summarized in Section 5.3.1.

In the Netherlands, the thermal performance of systems without auxiliary top-heating has been investigated at Level Energy Technology [5-4], [5-5] and at TNO Building and Construction Research [5-6]. The results of these investigations are summarized in Section 5.3.2.

5.3.1. Heat Storage With Built-In Auxiliary Energy Supply In Denmark, low-flow systems with four different heat storage/heat exchanger designs have been investigated [5-2], [5-

3]. Figure 5-1 shows a schematic of the four low-flow systems. For simplicity, the auxiliary energy supply systems are not included in the figure.

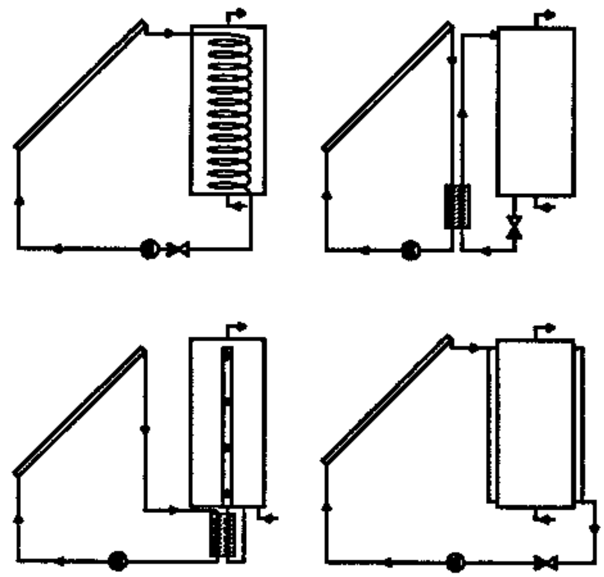


Figure 5-1. Schematic Illustration of Four Low-Flow Solar Heating Systems Investigated in Denmark with Differently Designed Heat Storages.

The first system consists of a hot water tank with a built-in heat exchanger spiral going from the top to the bottom of the tank. Solar collector fluid is circulated through the helical heat exchanger.

The second system consists of a hot water tank and heat exchanger loop with an external heat exchanger placed below the tank. Solar collector fluid is circulated through the heat exchanger. Water from the bottom of the tank is circulated through the heat exchanger to the top of the tank by natural convection.

The design of the third system is similar to the design of the second. This system also makes use of an external heat exchanger. Water is circulated from the bottom of the tank through the heat exchanger back to the hot water tank through a stratification manifold. The stratification manifold ensures thermal stratification inside the hot water tank. Water is circulated by natural convection.

The fourth system uses a mantle hot water tank as the heat storage.

Side-by-side tests with the four systems were carried out under realistic conditions. The hot water tanks of all four systems had electric heating elements located at the top of the tanks as the auxiliary energy sources. Therefore, the design of the auxiliary energy supply system did not influence the results of the investigations.

The detailed designs of the systems and the measured results are provided in [5-2] and [5-3].

The measurements showed little difference at high solar fractions in the thermal performance of the various low-flow systems. The differences between the thermal performance of the four systems are more pronounced during periods of low solar fraction. During these periods, the mantle heat storage system performs better than the other systems.

Differences between the yearly thermal performance of the various systems show up clearly only for relatively small solar fractions.

5.3.2. Heat Storages Without Built-In Auxiliary Energy Supply

5.3.2.1. A Dutch storage concept study. [illegible]

storage concepts provided a basis for the development of the future Dutch advanced solar energy DHW systems. The subject of this study was short-term thermal storage, the central component in a solar energy DHW system. The work embodied the selection of promising storage concepts, testing them at low-flow condition and analyzing the measurements. To support the results of the measurements, numerical simulations for both low-flow conditions and standard-flow conditions were carried out.

5.3.2.2. Storage selection and description. Storage selection was based on a number of conditions:

- The selection was made from currently marketed storage systems, as well as more experimental storage systems. As a reference case, a marketed storage system was used.
- Both collector circuit heat exchange and potable water heat exchange were considered.
- Collector circuit heat exchangers were located at the storage bottom or, for superior performance, run from bottom to top of the storage, being either a mantle or a helix.
- Potable water heat exchange by means of a finned helix or by means of a small potable water tank was considered.

Five storage types that satisfied these criteria were selected. (See Figure 5-2.) Storage 1 and 2 were currently marketed systems. Storages 3, 4 and 5 were experimental systems.

5.3.2.3. The storage tests. [illegible]

collector circuit charge step test was carried out. This test was followed by a mix, or diffusor, test. After reheating, a heat-loss test was carried out, followed by a discharge step test. Finally a simulation of "realistic" operating conditions was achieved with a 50% "noon" draw and a complete tank draw after 8 hours.

5.3.2.4. Results of the Dutch study.

- The perforated, tube inlet diffuser did not function properly. The absence of flow restrictors inside the tube may have caused poor performance. However, the malfunctioning diffusor had little influence on the system performance.

  Storage 4, with the collector inlet at mid-height, maintained heat in the upper part of the tank if colder collector water entered storage. When colder water entered tilt mantle, this inlet configuration functioned better than the tested inlet diffuser because it kept the upper section hot. However, during charging, the plume inlet caused a uniform temperature rise in the upper part of the storage because the plume of hoi water was mixed before it reached the top of the mantle. In this situation, the inlet diffusor functioned better.

  Storage 1 did not utilize its capacity. The collector heat exchanger should be extender to the bottom of the storage.

  Quality of storage insulation varied. One of the marketed storages had the largest heal loss of 1.86 W/K, although all connections were located at the bottom, whereas one experimental storage had the lowest heat loss, 0.93 W/K, although all connections were at the top of the storage.

  Storage 3, with the potable water heat exchanger, was a typical standard flow system The entire heat exchange area of the helix should be utilized for maximum performance. The dynamic test showed that the bottom part of the storage was still al a low temperature. Consequently the heat exchange performance was poor. Standard collector flow would have provided a more uniform storage temperature and consequently, a larger useful heat exchange area for the helix.

  One general conclusion is that the numerical simulations showed the draw pattern hac a much larger influence on storage performance than the primary choice of storage concept.

5.4. Auxiliary Energy Supply System

The auxiliary energy supply system can either be an integrated part of heat storage or i can be separate.

If the system is integrated into heat storage, it is important that the volume of the hea storage reserved for the solar collectors be sufficiently large [5-8]. Also, the auxiliary energy supply system must not heat the water to a temperature higher than required for comfort, health and safety. Finally, it is extremely important that the auxiliary energy supply system be located installed, and insulated in such a way that the extra heat loss from the heat storage caused by the auxiliary energy supply is held to a minimum [5-9].

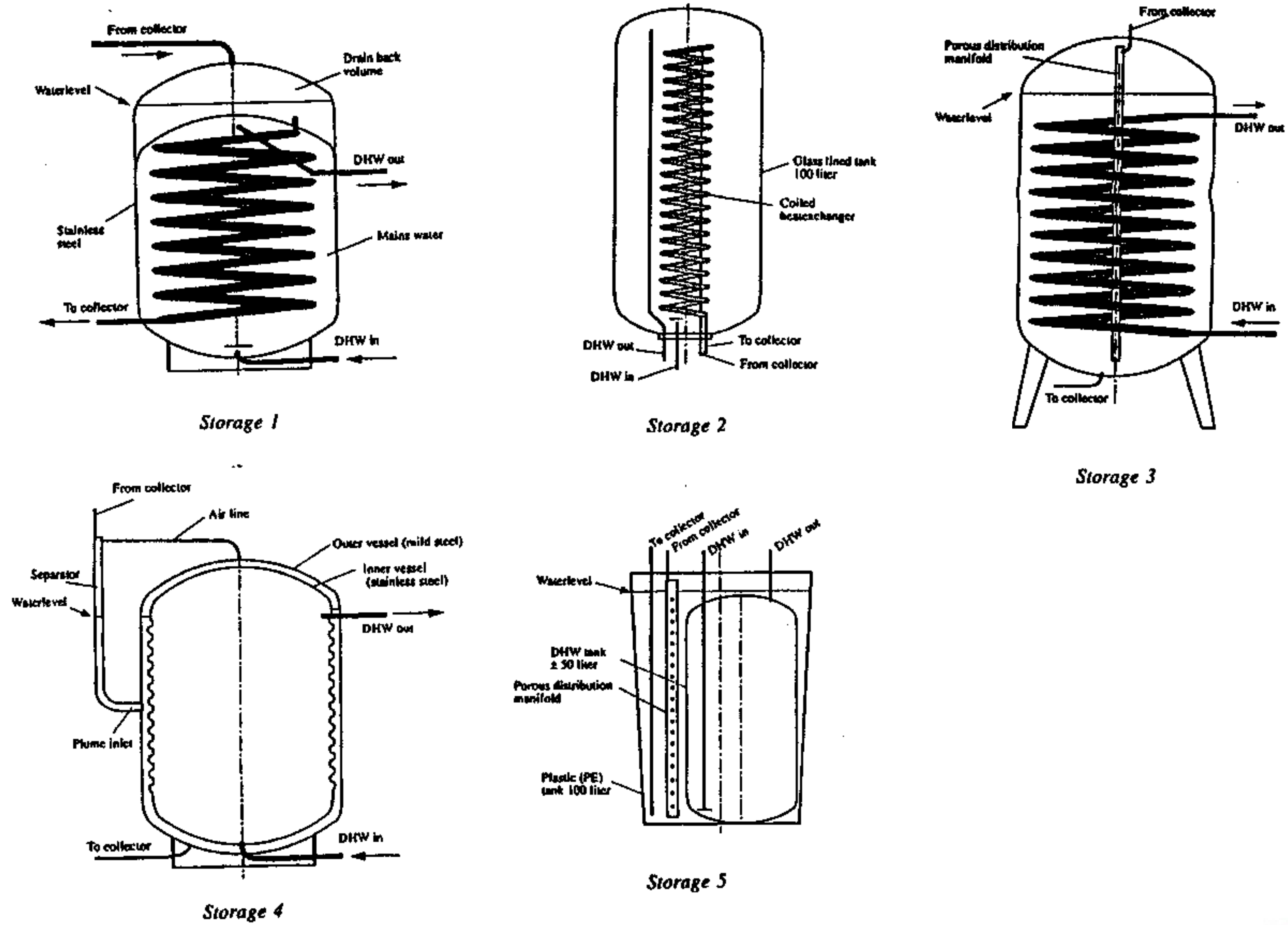


Figure. 5-2. Schematic Illustration of the Five Investigated Heat Storages.

5.5. Conclusions

The suitability of various DHW low-flow systems with different heat storage designs has been investigated, revealing little difference in thermal performance. Only at low solar fractions are there performance differences of importance.

Therefore, cost, rather than performance considerations, is likely to influence decisions on heat storage design.

In order to design an optimum heat storage, the following should be taken into consideration:

- The volume of heat storage reserved for the solar collector should be sufficiently large, depending on solar fraction and economics. A rule of thumb for dwellings is about 50 ℓ/m^2

- The capacity of the heat exchanger used to transfer heat from the solar collector loop to the heat storage should be sufficiently large, about 50 W/K per m^2 solar collector.

- The heat loss of the heat storage should be reduced to a minimum by insulating carefully. Thermal bridges caused by pipe connections should be avoided in the upper part of the heat storage. The total heat loss coefficient should not exceed that corresponding to a perfect insulation with about 5 cm of mineral wool.

In some countries, relatively expensive solar heat storage types are used. These heat storages are manufactured in relatively small numbers. In other countries, inexpensive standard hot water tanks manufactured in large numbers are already used as solar heat storage.

In the future, inexpensive solar heat storage will most likely be developed based on standard hot water tanks and/or utilization of design principles allowing use of inexpensive materials and techniques. For example, the drainback approach makes it possible to use a cheap, unpressurized plastic tank.

6. COMPONENT REPORT: PUMPS AND CONTROLLERS

6.1. Introduction

6.1.1. Overview There are few low-power, low-flow, moderately-high pressure pumps available for "microflow" solar water heaters. Most existing low power and low cost centrifugal pumps do not have a sufficient pressure rating to start drainback systems or to run systems with small bore tubing. Also, some pumps are designed primarily for higher flow and do not easily provide the proper flow rate specified in the system design, necessitating adjustment on site.

Positive displacement pumps readily control flow rate, but tend to be bulky and expensive.

Compared to 100W pumps once in use, a system's net thermal rating could be raised about 10% if 5W pumps were available. Since the start of this Task, 20W to 30W pumps have come into wider use, so the power saving of switching to even lower power pumps is somewhat diminished. However, the capital cost savings could still be substantial. Also, very low power operation would make PV power attractive for off-grid sites now, and for all sites if the price of PV modules dropped sufficiently.

6.1.2. Centrifugal Pumps A number of European participants, including the Dutch, use a Grundfos pump, model UPS 25-40, that uses 30W at its lowest speed and is priced at approximately US$40. Its maximum head of about 1 meter at this speed is marginal for small bore tubing. It can perform a drainback start-up only for systems with up to a 4-m elevation. [illegible] than necessary. (See Figure 6-1.)

Another possibility, being examined by the Dutch team, is a small automotive windshield washer pump, powered by a 12V DC brush-type motor. It has a seal between the motor and pump sections, and motor bushings instead of ball bearings.

At 12V the pump can pump 1 ℓ/minute at an 8 meter head, and consumes 37W of power. The test lifetime at this voltage is 24 hours. By reducing the voltage to 6V, for example, the head drops to 3 meters at 1 ℓ/minute, the power drops to 8.4W, and the test lifetime increases to 2,000 hours (but with significant wear). At 4V, the flow drops to 0.67 ℓ/minute at 1.7 meter head, but the extrapolated lifetime might rise to between 9,000 and 25,000 hours. (See Figure 6-2.)

The Canadian team is developing a small, high-speed centrifugal pump of potentially low cost, with enough pressure to start a drainback system. The design concept emphasizes minimum wear, and hence maximum durability. The variable speed design inherent in the electronic drive allows automatic flow regulation, assuring that the system operates as designed.

Note that closed loop systems (non-drainback, usually with glycol antifreeze) do not need an increased pressure rating at start-up, other than to overcome cold glycol viscosity. This eases the pump ratings and expands the selection available.

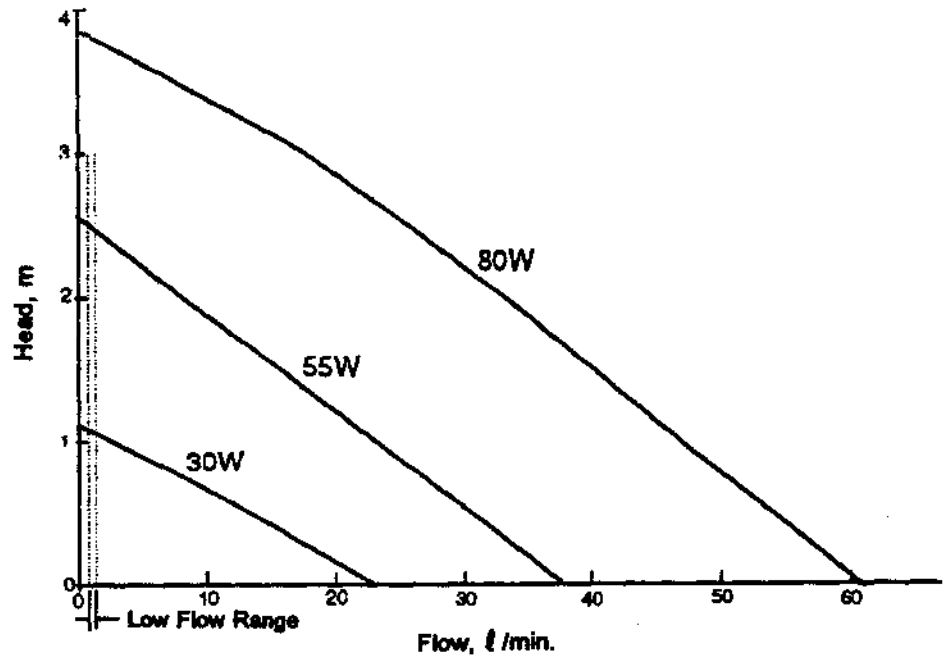


Figure 6-1. Grundfos UPS 25-40 Pump Performance (Source: Grundfos).

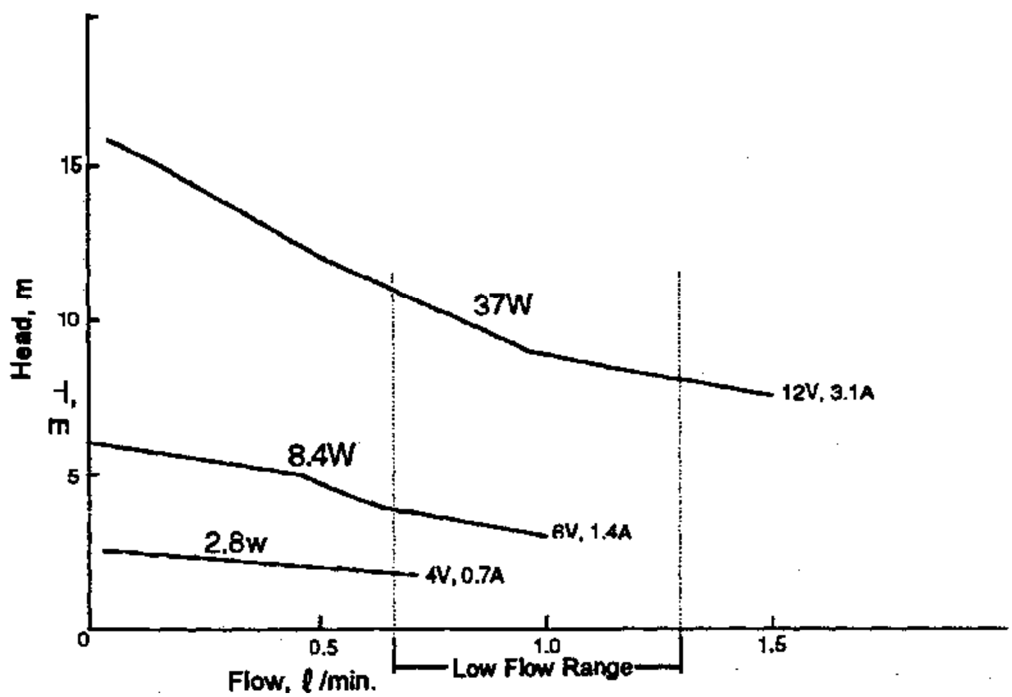


Figure 6-2. Bosch Impeller Pump Performance (Source: TNO, NL).

6.1.3. Positive Displacement Pumps The Swiss participants at SPF/ITR Rappersvil are evaluating a PTFE ("Teflon") diaphragm pump (KNF ND 1.100) requiring 20 to 24W. At one bar it can pump 40 ℓ/hr, or 0.67 ℓ/minute. Start-up pressure can reach 3 bars. Durability is expected to be 20,000 hours. The price in 1000 quantity is about 270 Sfr, or about US$186. (See Figure 6-3.)

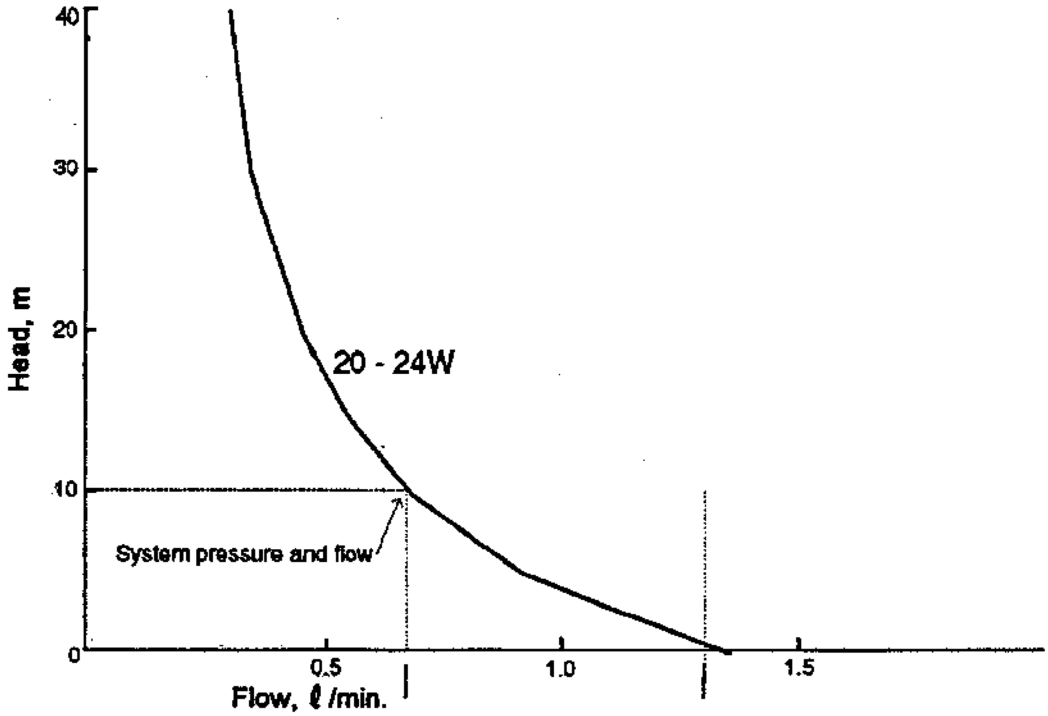


Figure 6-3. PTFE Diaphragm KNF ND 1.100 KT Pump Performance (Source: KNF data sheets via SPF/ITR, Rapperswil).

The pump used in many existing Canadian low-flow systems is a Procon vane pump, powered by an AC motor, or a DC motor with either a transformer/rectifier or a 17 to 20W PV panel. The DC pump is priced at about US$175, and the PV module is about the same. At 1.38 [illegible]
efficiency of 26%. As a relatively high-power capacity, positive displacement pump (about 7 bar and [illegible] (ℓ/minute), a pressure-flow performance curve is not relevant to the present application.

Both of these pumps are much more expensive than desired for solar DHW systems.

6.2. Comparison of Different Concepts

The nature of centrifugal pumps is such that low speed pumps (i.e. 3,000-3,600 RPM) will have either too little pressure or too much flow, or both, for single-family, SDHW low-flow systems.

Positive displacement pumps have piston, gear, vane, and/or valve wear points that make the pump sensitive to fine debris in the fluid. Diaphragm pumps eliminate sliding contact, but still have valves which are subject to wear.

Two of the known "thermal pumps" are quite complex, both to build and to install, and are expensive. This type of pump was judged by Dutch evaluators to offer no cost/performance benefit

6.3. Design Criteria for Low-Flow Pumps and Controls

<u>6.3.1 Pumps</u> The desired operating characteristics for a pump for solar collector loops depend on the system design. Lower latitude sites may not need a pump at all, the tank being installed on a roof above the collectors to allow the heated water to thermosyphon to the tank.

For the higher latitudes where freeze protection is required, a closed loop can be used, with a water-antifreeze mixture. This approach can use a low-pressure, centrifugal pump because the supply and return fluid columns are always filled and the only pressure demand comes from fluid viscosity. This can be high during a cold start, but a small flow will occur and eventually warm up the loop. Thus operating pump pressure, and hence power, can be fairly low.

Drainback systems can use plain water without antifreeze if the piping and pump design ensures that the water can drain from the outdoor loop. Because the liquid is replaced with air, the pump must refill the loop, starting with the supply side. This results in a static head that may well exceed the running pressure. Centrifugal pumps must run at higher speeds (or have larger impellers) to reach higher pressures. This usually produces flow rate too high for a cost effective system design.

Development of easy-to-install, flexible, pre-insulated tubing bundles has made it desirable to use smaller tubes, raising the pumping pressure even at low-flow rates. If a higher pressure pump can be made inexpensive enough not to use up all the cost savings on the tubing, a better system results.

<u>6.3.2. Controls</u> The most common controller is the fixed delta-T type that turns on the pump (at either a fixed speed, or two speeds including a high speed for drainback start-up) when the collector is warmer than the bottom of the tank. For stability, there is hysteresis between the "on" and "off' temperature differences, the "off' being lower. The pump is turned off if the tank temperature rises too high. Such a device is simple and readily available.

Photovoltaic power for the pump can provide an alternative control strategy. The pump will run only when the sun is shining, and the speed will increase with the level of insolation. All that needs to be added is over-heating protection for the tank. If the pump is already DC powered, the electronics design can be very simple. On the other hand, the cost of electronics is decreasing so complexity (e.g. brushes, commutator) may be shifted from the mechanical pump to an electronic circuit. The present, somewhat high, cost of the PV array can in some cases be offset by the cost savings in not having to install a mains cable and outlet in the vicinity of the pump.

Another control device is the so-called "light switch" (a photo-detector), which, like PV powering, runs the pump when the sun shines, although usually at a fixed speed. Power is brought from the mains, as usual.

As low-flow systems have improved, there appears to be a diminishing additional energy benefit to be gained by using variable flow, probably no more than a few percent. Further studies are required before variable flow could be proposed as a significant energy producer. But if the pump is electronically driven, variable flow will add almost no cost, so the additional benefit might be gained for free. On the other hand, variable flow may significantly enhance tank stratification under fluctuating conditions, by minimizing the strength of, or eliminating, thermal inversions.

6.4. Development of a New Low-Flow Pump

The best pump concept is one that is inherently simple, making it possible to avoid mechanical contact and its attendant wear.

<u>6.4.1. Project</u> The project was to design, build, and test a low-flow centrifugal pump, following the above criteria.

<u>6.4.2. Purpose</u> The purpose was to develop a pump with the special characteristics needed in low-flow solar water heaters, including drainback systems with the collectors 20 meters above the pump.

<u>6.4.3. Description of Work</u> A small high-speed centrifugal pump was designed and built. To achieve the speed necessary (up to 40,000 RPM) to break free of the pressure-flow limits of mains-driven pumps, the motor power was provided with electronic frequencies in the 2 to 3 kHz range. The pump weighs about 30 g, of which the rotor contributed two grams.

The pump was designed to minimize parasitic losses within the constraints of physical size (limited by the precision of the smallest components the present tooling can produce). Dimensional inaccuracies in the first prototype appeared to cause hydraulic losses higher than those predicted.

The motor has no brushes, allowing it to be immersed in the fluid. Because of this, the pump has no shaft penetrations and hence no seals to the outside. Being centrifugal, it has no valves. The result offers low friction, low complexity, and potentially high durability.

The target flow and pressure were 1.3 ℓ/minute at 0.9 atm, with a pressure maximum of 2 atm at start-up. One set of tests at 13.2W of DC input achieved target pressure and flow, with a maximum pressure of 2.2 atm (no flow), and a maximum flow of 1.9 ℓ/minute (no pressure). (See Figure 6-4.)

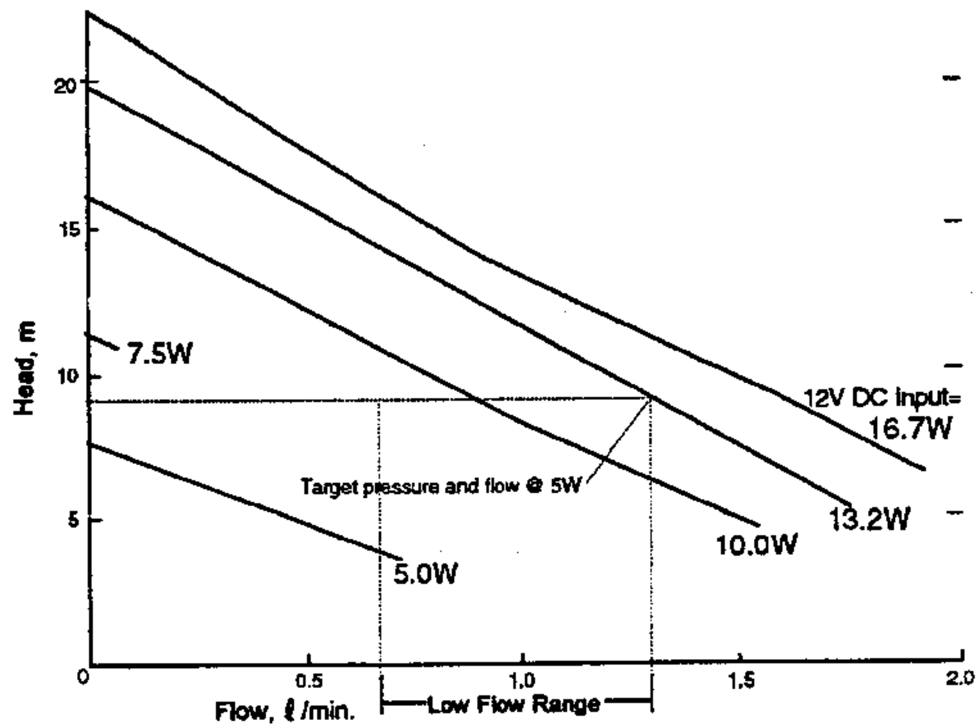


Figure 6-4. Canadian Nanopump First Prototype Performance (Source: Negentropy Inc. in-house tests).

The target power consumption was 5W. The pump was tested at power levels from 0.75W to 16.7W, and at speeds from 12,000 to 43,000 RPM. It did not achieve target flow and pressure at 5W.

The pump drawing cannot be supplied until the patent application has been filed.

The pump was initially intended to be grid powered (by means of a small AC adapter), with PV power as an option.

The bearings are hydrodynamically lubricated with water. There was momentary low speed rubbing contact at a two gram load on these bearings as the pump started, but none during operation.

No corrosion is expected. Exposed parts will be of ceramic, plastic, or stainless steel. Since the total weight is only 30 grams, corrosion resistant materials do not appreciably affect cost.

The pump is not intended for applications involving a continuous throughput of hard water. Small internal clearances have little tolerance for lime build-up. The normal use of the pump is in closed systems only, where the sole source of lime is the initial charge of water, plus any replacement over the life of the system. These charges normally consist of an antifreeze mixture, and can economically be made with distilled water since the total fluid volume is only a few liters in the full microflow design.

How much filtering the pump inlet will require is unknown as yet. Small internal clearances would suggest a 25-micron filter.

There are no restrictions on the location of the pump. It is designed for the start-up of drainback systems up to 20 meters in height.

The system connections are as simple as possible for a pump having inlet and outlet connections. A simpler, slightly less expensive option is to use the pump in its submersible version installed in the fluid reservoir, saving a reservoir-to-pump connection.

6.4.4 Control The pump is inherently capable of variable flow, but the initial control algorithm will incorporate fixed flow for grid-connected systems. The PV-powered option will naturally exhibit variable flow.

Integrated auxiliary control is planned for the controller but is not part of the present prototype power driver. When in-line-auxiliary control is implemented, the tank stratification can be guaranteed, and more electricity consumption can be shifted to off-peak.

Boiling protection will be the responsibility of the controller, by draining the collectors when they cannot be cooled. It is suggested that microflow systems use antifreeze due to small-bore pipe (Life-Line®) draining considerations under freezing conditions, so the controller is not strictly necessary for freeze protection. The collector loop would still be drained during freezing conditions, or when there is no energy to collect, but mostly to save energy and reduce viscosity during initial fluid heating on re-start.

6.5. Future Developments

Further developments could reduce the size and increase the efficiency of the pump. The motor electromagnetic efficiency is currently 85%, including electronics, but the overall efficiency is only 14%. All of the parasitic hydraulic losses are surface area dependent, and both the motor and the impeller could be made smaller with the right manufacturing equipment. Future bearing designs could eliminate all wear. The present 12V control chip uses bipolar transistor technology, and about 0.7W could be saved using a CMOS chip.

6.6. Conclusions

6.6.1. Common Conclusions Some low cost centrifugal pumps may prove to have enough durability and pressure rating to be competitive. Usually, though, they have too much flow capacity and are not easily regulated to provide predictable flow at lesser rates.

Most existing positive displacement pumps are too expensive, although some are low power. Their durability may not always be adequate. They do, however, provide good flow rate control.

6.6.2. Specific Conclusions (Canadian Pump) It appears possible to develop and build efficient, low cost, high reliability pumps weighing 30 grams or less. A major cost is in the electronics, and is amenable to dramatic reduction with volume production.

The basic motor design works well. The pump needs further technical development in the fabrication of hydraulic components to achieve maximum efficiency. The commercial electronic 12V control chip works reasonably well, once the original circuit design was severely modified to overcome chip limitations, but that chip consumes nearly 3/4W. A new (and smaller) circuit board with a lower power 5V chip is in final development, and with minor hydraulic improvements, lOW power consumption is anticipated. Ultimately, a more sophisticated control chip is needed.

There is no real manufactured cost data yet, but the pump is expected to be priced at perhaps US$50 or less in large volume, more likely US$150 in preliminary low volume. The initial cost of parts and materials is about US$15.

The pump is expected to save between 0.15 GJ/an (compared to a 20W alternative) and 0.95 GJ/an (compared to 100W). The former figure is a bit more than 1% of system output. At an estimated median price of US$100, plus US$50 for the PV panel, the pump lowers the Canadian base system cost by about US$140, for a system cost/performance reduction of 9%. If PV costs do not drop soon, there may be further interim cost savings in using a small 12V AC adapter instead of the panel. For these non-PV systems, the main benefits of the pump may be in higher pressure for drainback operation and for small-bore piping, and ultimately an even lower price due to its small size.

6.6.3. Direct Comparison with Other Pump Designs Compared to positive displacement pumps, this pump is expected to be more durable, have lower power consumption, and be less expensive.

Compared to more conventional centrifugal pumps, it should be no more expensive, consume less power, and have a higher pressure rating relative to the flow rating.

Compared to thermally driven pumps, it will offer more net system output and, be less expensive, less complex, and easier to install.

7. COMPONENT REPORT: PIPING

7.1. Introduction

Today the piping of the solar domestic hot water system collector loop is usually built with copper, steel or stainless steel tubes. Return and feeding pipe from and to the storage are separately insulated. The expense of materials and the installation costs of the rigid pipes and insulation are substantial.

The low-flow principle makes it possible to reduce the system flow rate by a factor of 5-10. Therefore, much smaller tubes with inner diameters in the range of 5 to 8 mm can be used. To optimize the advantages of smaller diameter piping we can introduce more compact all-in-one solutions, such as both tubes and the electrical wiring for temperature sensors in one envelope. The use of smaller diameter piping also lends itself to the use of flexible non-metallic materials or easy-to-bend copper tubing.

To ensure a long material lifetime, the following requirements for tubing materials should be considered:

- Durability at temperature and pressures up to 200°C and 4 bars
- Durability using water-glycol mixture
- Durability when exposed to UV-radiation

An overview of the different concepts concerning the use of flexible tubing for "low-flow" DHW systems is presented in this chapter. This chapter also includes discussions on the types of piping materials, pressure drop and heat losses.

The potential benefits of the use of flexible tubing are:

- Reduction of installation cost by the use of flexible tubing (fast and easy installation) including electrical wiring
- Reduction of heat losses by the use of smaller tube diameters and combining the insulation of the "hot" and "cold" tubing
- Reduction of used row materials by minimizing tube diameter and wall thickness
- Easy handling and delivery of the complete tubing

The disadvantages of using flexible tubing are:

- To be used only for "low-flow" concepts in small DHW-installations

- Some designs of flexible tubing with factory fitted insulation show a tendency toward mechanical damaging of the insulation during installation

- Small bend radius might lead to additional, undesired pressure drop due to reduction of the diameter

- Small diameter may cause problems for proper draining of drain-down systems (inner diameters below 10 mm are critical)

- Larger risk of blocking up the solar collector loop

7.2. Comparison of Flexible Piping for Low-Flow DHW Systems to Fixed Piping in Traditional DHW Systems

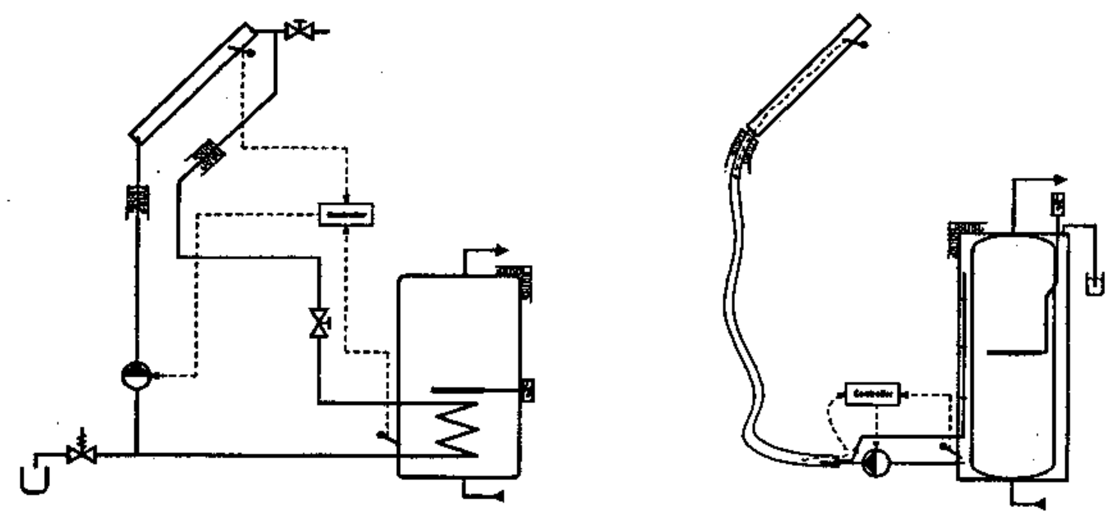


Figure 7-1. Scheme of the Traditional and Low-Flow Systems.

Figure 7-1 illustrates the differences in fixed and flexible tubing. In the flexible tubing the return and feeding pipes from and to the storage are both in one envelope. The tubes are separately insulated to reduce heat transfer between them. In a low-flow system the temperature difference between in and outlet of the solar collector could be as high as 40 - 50°C when maximum insolation occurs. In the improved design shown in Figures 7-3 and 7-5, the "hot" tube is more heavily insulated than the cold. The flexible system also includes a wire for the temperature sensor and/or photovoltaic module. The flexible tubing is easily connected to the collector via special fittings.

7.3. Design of Different Concepts

The most obvious difference between the fixed and flexible tubing in these examples is the diameter of the tubes. The typical inner diameter for traditional systems of 12 to 15 mm relates to the necessary flow rate of 200 to 400 ℓ/hr. By using the low-flow principle,
the flow rate is reduced to 30 to 60 ℓ/hr,
mm. The use of rigid tubing in low-flow systems seems as unlikely as using flexible tubing in traditional systems. Nevertheless, a Dutch approach provides an interesting compromise: a semi-flexible copper tube with an inner diameter of 10 mm leads to an acceptable flow rate range of 60 to 200 9/hr regarding pressure drop. For this diameter, pipes need to be correctly installed (sloped) to drain properly.

7.3.1 Fixed Tubing

Design:

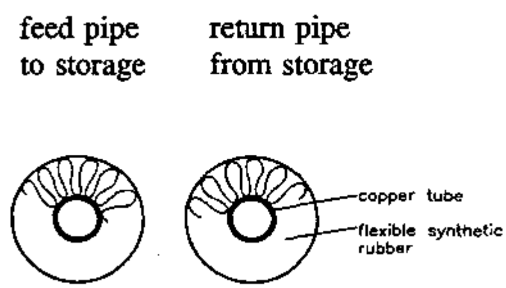


Figure 7-2. Cross Section of the Typical Fixed Tubing.

• Tube: Material: copper Dimensions: d_o = 15 mm d_i = 13 mm	**• Insulation:** Material: flexible synthetic rubber Dimensions: d_i = 15 mm d_o = 33 mm
• Cost: approx. 10-12 US$/m	**• Installation time:** typically 6 hours for 15 m

7.3.2 Flexible Tubing Swiss Flextube®

Design:

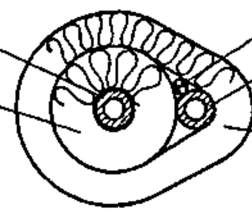


Figure 7-3. Cross Section of the Swiss Flextube®.

• **Tube:** Material: silicone Dimensions: d_o = 9 mm d_i = 5 mm	• **Insulation:** Material: flexible synthetic rubber Dimension inner insulation: d_i = 10 mm; d_o = 28 mm Dimension outer insulation: d_i = 35 mm; d_o = 53 mm
• **Cost:** - approx. 12 US$/m for purchase of 1000 m or more - additional costs for four fittings of approx. 5 US$	• **Installation time:** typically 3 hours for 15 m

Comment:

The advantage of the Swiss Flextube® is the high degree of flexibility of the tubing. Fast and easy installation is possible. Additionally, the "hot" pipe is red and the "cold" pipe is grey, so mistakes during installation are unlikely. The "hot" pipe has more insulation than the cold pipe, and the bundle includes a wire temperature sensor. The disadvantage is the sensitivity to insulation damages during installation. A protective jacket is recommended. Furthermore, if installed outdoors, the Flextube® should be protected against weathering.

Fittings:

The fittings shown in Figure 7-4 are part of both the collector and the storage tank (refer to Figure 7-1). Rubber tubing can be mounted by sliding the tube onto the nipple portion of the fitting and securing it with the spring loaded clip shown.

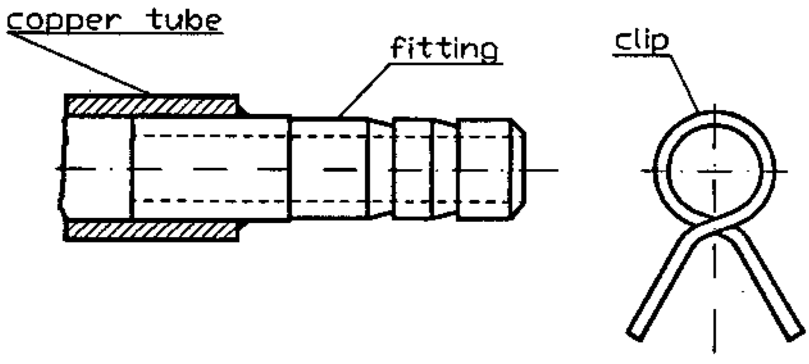


Figure 7-4. A Diagram of the Fittings Connecting the Collector / Flextube® and the Storage / Flextube®.

7.3.3. Flexible Tubing Canadian Life-Line®

Design:

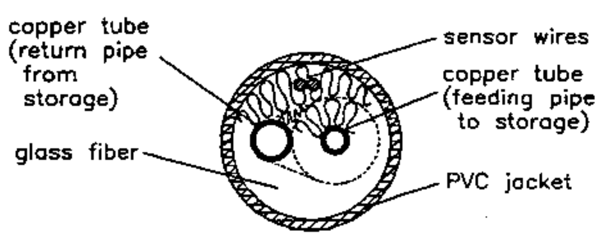


Figure 7-5. Cross Section of the Canadian Life-Line®.

• **Tubes:**	• **Insulation:**
Material: copper Dimension feeding pipe: d_i = 4.83 mm; d_o = 6.35 mm Dimension return pipe: d_i = 7.90 mm; d_o = 9.53 mm	Material: non-hygroscopic glass fibre with exterior PVC jacket Dimension: d_i = 6.35 mm; d_o = 40-42 mm thickness inner insulation s_1 = 7 mm thickness outer insulation s_2 = 7 mm thickness of PVC jacket s_3 = 2.5 mm
• **Cost:** - approx. 20 US$/m (purchase of 2000 m) - additional cost for fittings approx. 5 US$	• **Installation time:** Typically 1-2 hours for 15 m Life-Line®

Comment

The Canadian Life-Line® consists of a 1/4 in. copper feeding pipe, a 3/8 in. copper supply pipe, two sensor wires, non-hygroscopic glass fibre insulation, and an exterior PVC jacket. Hot solar collector fluid flows through the small pipe towards the centre of the Life-Line® while the cold solar collector fluid from the heat storage flows in the larger pipe located closer to the outside of the Life-Line®.

Fittings:

Compression fittings (3/8 in. and 1/4 in.) on soldered copper couplings

7.4. Pressure Drop of Different Concepts

Figure 7-6 shows a comparison of calculated pressure drop curves with the fixed tubing, Swiss Flextube®, and Canadian Life-Line® at volume flow rates of 20, 40, and 80 Whr. Calculation of pressure drop based on a mixture of 1/3-vol.% Ethyleneglycol and 2/3-vol.% water and one meter length of either supply or feeding pipe.

7.4.1. Discussion of Results Figure 7-6 shows large differences in the pressure drop between the fixed and the flexible tubings.

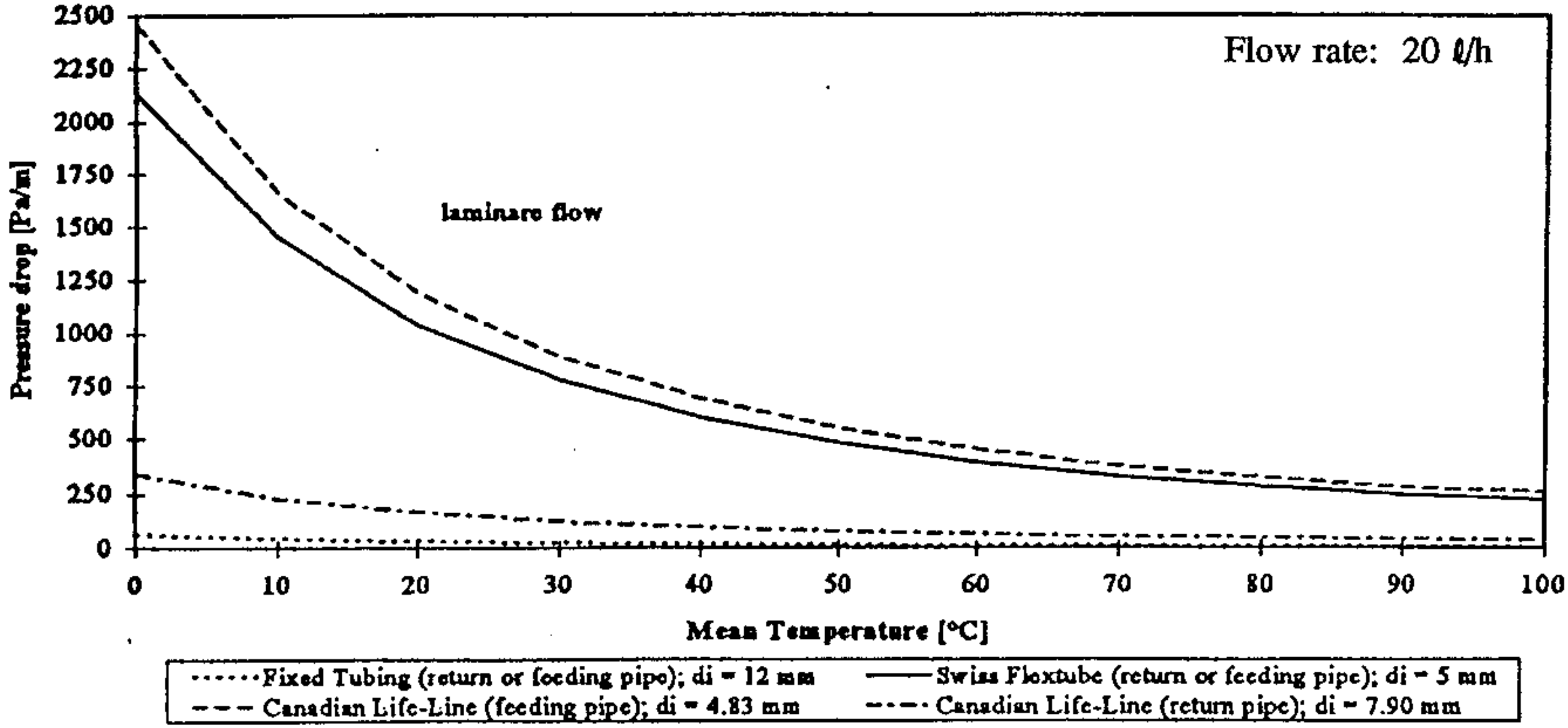


Figure 7-6. Comparison of the Pressure Drop for Different Tubing Designs and Flow Rates.

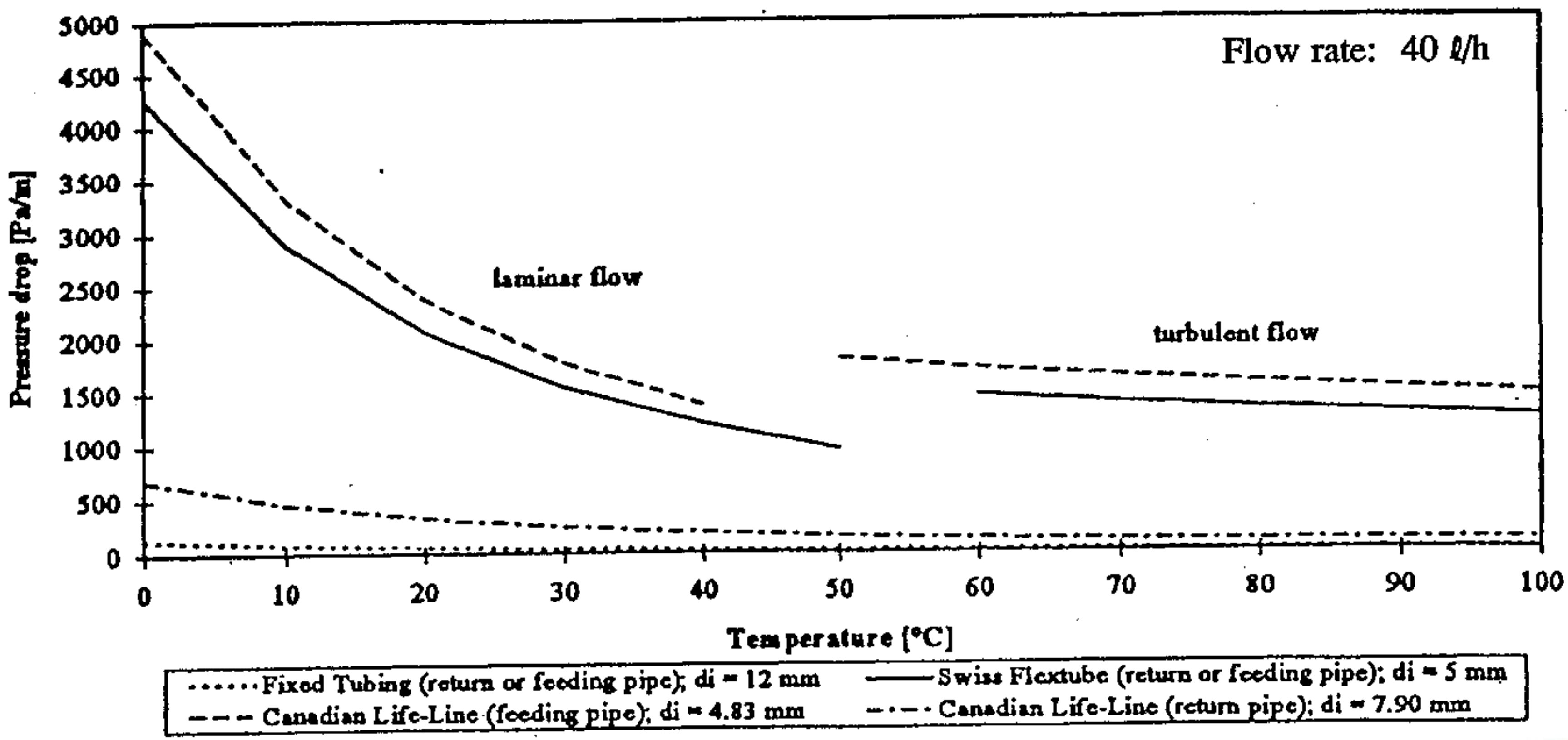


Figure 7-6 (cont.). Comparison of the Pressure Drop for Different Tubing Designs and Flow Rates.

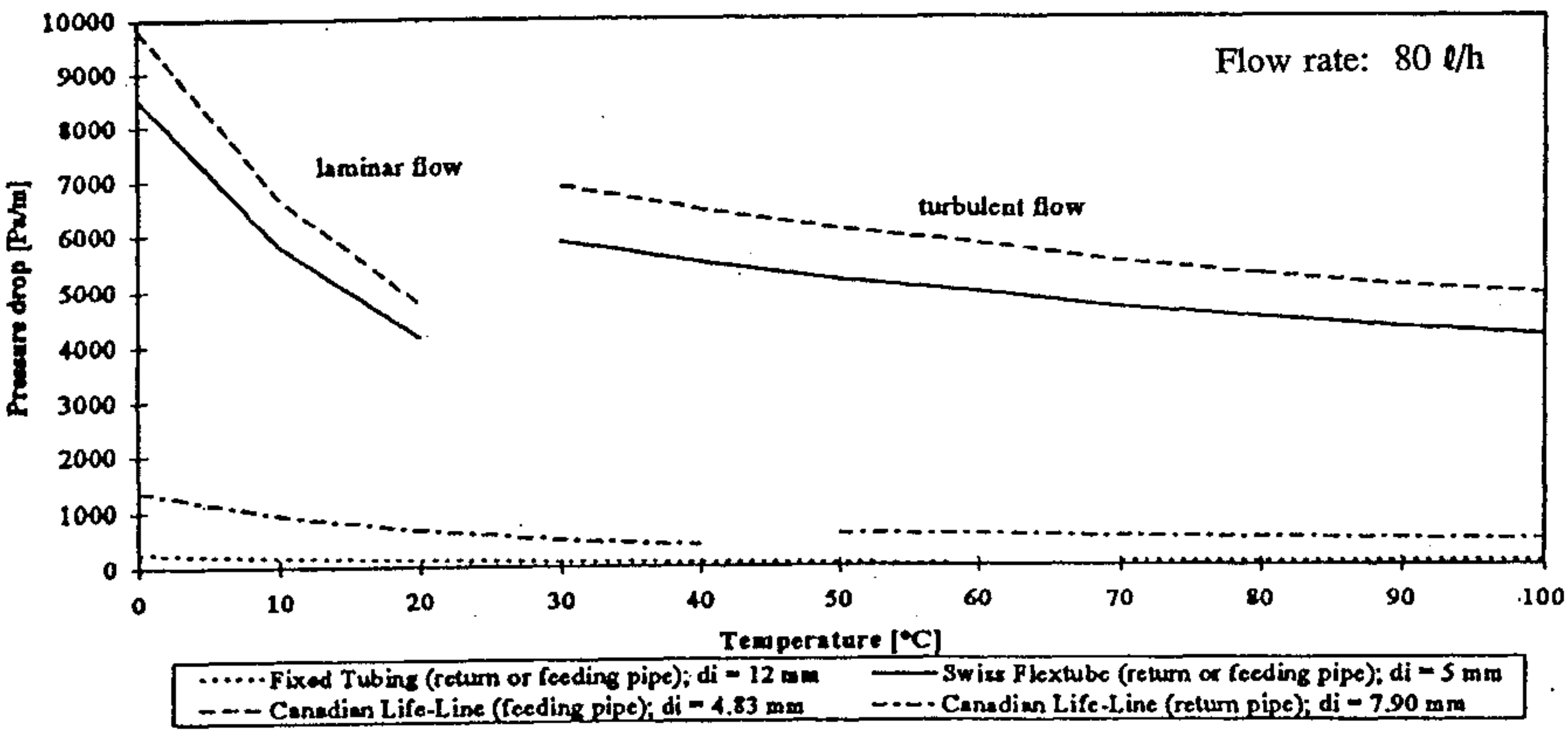


Figure 7-6 (cont.). Comparison of the Pressure Drop for Different Tubing Designs and Flow Rates.

During system operation, the maximum pressure drop occurs when the fluid in the cold tube is near cold water temperature of the storage and the pump begins to run. At this moment the temperature of the hot tube is nearly the same as the cold tube.

The following example shows values for a flow rate of 40 Or, a mean temperature of 10°C for the return pipe, a mean temperature of 12°C for the feeding pipe and 15 m length of piping:

	Pressure drop of Fixed Tubing	Pressure drop of Swiss Flextube®	Pressure drop of Canadian Life-Line®
Return pipe from storage	1311 Pa	43500 Pa	6980 Pa
Feeding pipe to storage	1237 Pa	41030 Pa	47120 Pa
Total	**2548 Pa**	**84530 Pa**	**54100 Pa**

On a sunny day, the cold tube in a low-flow system operates near cold water temperature of the storage while the hot tube is often in the range of 50 to 70°C. Therefore, the pressure drop of the return pipe from the storage is higher than the pressure drop of the hot feeding pipe.

The following example shows values for a flow rate of 40 9/hr, a mean temperature of 10°C for the return pipe, a mean temperature of 60°C for the feeding pipe and 15 m length of piping:

	Pressure drop of Fixed Tubing	Pressure drop of Swiss Flextube®	Pressure drop of Canadian Life-Line®
Return pipe from storage	1311 Pa	43500 Pa	6980 Pa
Feeding pipe to storage	357 Pa	21875 Pa	25781 Pa
Total	**1668 Pa**	**65375 Pa**	**32761 Pa**

These two examples show pressure drop for typical operating conditions. Large differences in the pressure drop between the three concepts could be seen.

7.5. Heat Loss

There is a large difference between the operation of a low-flow system and a traditional system. Figure 7-7 shows the radiation and temperatures over time for a typical sunny day for a low-flow system and a traditional system.

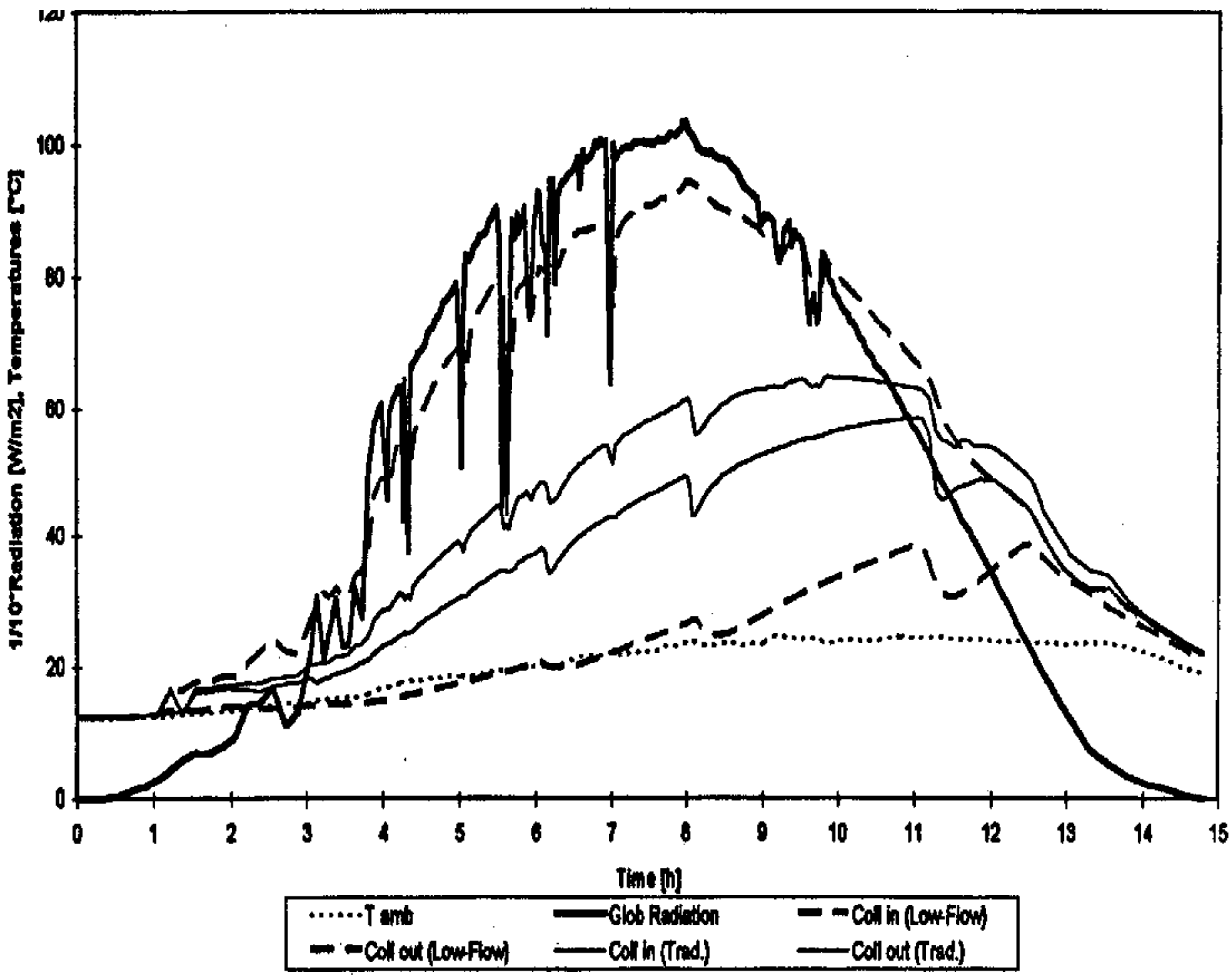


Figure 7-7. Typical Operation of a Low-Flow and a Traditional System on a Sunny Day.

The cold tube in the low-flow system operates near ambient air temperatures while the hot tube is often in the range of 50 to 70°C. Therefore, the use of the Canadian and the Swiss flexible tubing can result in losses from the hot tube to the environment or heat transfer from the hot to the cold tube. In the traditional system, both tubes are operating at the same temperature range (temperature differences from 5 to 15 K). Both are above the ambient air temperature, and therefore both have heat losses to the environment.

7.5.1. Results of Heat Loss Measurements — Measurements of different tubing under different conditions were taken at various laboratories. The fixed piping and the Swiss Flextube® were measured at the Solar Energy Laboratory in Rapperswil/CH while the Canadian Life-Line® and the Swiss Flextube® were measured at the Thermal Insulation Laboratory at the Technical University of Denmark.

The Swiss measurements were taken during the testing of complete systems. The inlet and outlet tubing temperatures (return and supply) were measured during the operation of the systems. Therefore, dynamic and system operation effects, as well as changing weather conditions, might influence the results. Nevertheless, the values presented show how the different concepts perform under realistic conditions. The mean piping losses for traditional systems range from 0.5 to 0.9 WK ; however, most of the values vary over a wide range in the order of 0.8 $WK^{-}m^{-1}$. The values include losses for both return and supply tubes. Losses for the Swiss Flextube® in a low-flow system are much lower, in the range of 0.35 to 0.5 .

The investigations done by Denmark were conducted in the laboratory. Values are much lower than shown by the Swiss measurements. The losses for either the Canadian Life-Line® or the Swiss Flextube® are in the range of 0.2 to 0.3 $WK^{-1}m^{-1}$. The results from the simulations done by Canada [7-2], [7-4] and Denmark [7-1] show a very good agreement to the laboratory results from Denmark [7-1].

Due to the different testing conditions of the Swiss and the Danish investigations, the results are difficult to compare. Nevertheless, the Swiss results show for the flexible tubing in a low-flow system lower losses by a factor of two compared to ordinary piping in a conventional system.

However, the heat loss and the heat exchange between the cold and the hot tube can be calculated. These calculations can be used for the optimization of insulation of flexible tubing. In the reports [7-1] from Denmark and [7-2] to [7-4] from Canada, calculations of heat loss coefficient are presented and compared with measurements performed in Denmark [7-1].

7.5.2. Analysis of Heat Losses in Flexible Piping Bundles — An analysis of the heat transfer taking place between the components of flexible tubing bundles has been carried out in Canada [7-2, 7-4, 7-5]. In DHW systems employing such tubing bundles, there is a thermal performance penalty caused by heat transfer from the hot tube to the cold tube. The penalty in system performance happens because the cross heat transfer results in higher collector inlet temperatures, lower collector efficiency, and lower solar energy being delivered to the DHW storage system. The Canadian work has shown that the standard Hottel-Whillier-Bliss (HWB) equation can be modified to simultaneously take into account both the pipe heat losses to the ambient environment and the cross heat transfer between the hot and cold streams. Parameters in these equations are the thermal resistances between the fluids in the two tubes, and between each fluid and the ambient air. Methods are presented in reference [7-5] for both calculating and measuring these thermal resistances.

The parameters in the modified HWB equation were calculated for the case of a representative solar D11W system (delivering about 50% of the energy required to supply 300 liters/day of water at 60°C) equipped with either of two different flexible tubing bundles: Life-Line-C®, and one consisting of two Nylon-11 tubes inside a PVC cover that contained no thermal insulation. The thermal effects of the tube bundles reduce the net delivered solar energy by 6 to 14%. The loss in system performance due to cross heat transfer was found to be practically independent of the loss in performance due to heat losses from the tubes to the ambient air. Moreover, heat loss to the ambient air was found to be more detrimental to system performance than is heat transfer from the hot to the cold conduit.

7.6. Materials and Requirements

Materials used and requirements for their use are given in Table 7-1.

Table 7-1. Materials.

Type	Material	Dimension d_i/d_o [mm]	Bend radius [mm]	Density [kg/dm^3]	T_{max} [°C]	P_{max} [bar]	UV-stability yes/no	Component costs * [US$/m]
Fixed:								
tube	copper	13/15	40	8.9	>200	>20	yes	3
insulation	synthetic rubber	15/41		0.067	120	-	no**	3
fitting	-	-		-	-	-		-
Swiss Flextube®:								
tube	silicone	5/9	50	1.61	>200	4	yes	2
insulation	synthetic rubber	10/28		0.067	120	-	no**	1.5
insulation	synthetic rubber	35/53		0.067	120	-	no**	1.5
fitting	brass	5/7.6		8.5	>200	>20	yes	1
Canadian Life-Line®:								
tube	copper	4.8/6.4 + 7.9/9.5	230	8.9	>200	>20	yes	2
insulation	fiberglass	6.4/30.4		0.15	200	-	-	1.50
insulation	fiberglass	30.4/34.4		0.15	200	-	-	1.50
jacket	PVC	35/40		1.38	100	-	yes	-
fitting	brass	3/8 in. + 1/4 in.		8.5	>200	>20	yes	-

*Component costs only includes raw material costs without marketing, selling and distribution costs (not the end price to the user)

**To install outdoors UV-protection is needed

7.6.1. Results of Aging Tests, Experiences in the Field The most important aspect regarding new tubing materials such as plastics or rubber is their durability! Short lifetimes have been reported in Canada with Nylon-11 tubing. Besides the lifetime of the tubing material itself, the fittings and connecting tubing material are of great importance. Compression fitting leaks in combination with some tubing materials (e.g. Teflon or Nylon) have been reported in Canada [7-3], and 0-ring fittings rather than compression fittings should be used on all plastic tubing connections.

The use of silicone rubber hoses for automotive application is well known. Also, silicone hoses have been used in solar applications for collector couplings for more than 15 years. The fittings and clips used to connect the silicone tubing to the collector and storage tank have also been used for many years in similar applications without any problems.

More work is needed to find other suitable materials which are cheaper than the present silicone rubber tubes.

7.7. Conclusion

Integrated flexible tubing is of great interest to the development of better domestic solar water heaters for low-flow applications for the following reasons:

The heat losses are lower by a factor of two or more compared to fixed piping.

Installation time is shorter and therefore cost of installation is lower.

In addition, the cost of silicone hoses with an inner diameter of 10 mm or more for traditional high-flow systems is very high and the copper alternative for these diameters is more difficult to install because of its poor flexibility. Therefore, the advantage of flexible tubing is mainly realized in combination with low-flow systems, where smaller diameters are needed.

Further developments are required to achieve the ideal tubing including:

Finding more appropriate production techniques for lower cost products.

Finding new non-metallic materials with lower prices for the tubes, as well as for the insulation.

8. LOW-/HIGH-FLOW TEST

8.1. Introduction

Both experimental and theoretical investigations which compare the thermal performance of low-flow systems and of traditional high-flow systems have been carried out [8-1], [8-2], [8-3], [8-4].

The designs of the systems, as well as the test conditions, influence the relative performances of the systems in experiments. The relative performances of the systems determined by means of simulation programs are influenced by the suitability of the programs and by the input data. Therefore, it is extremely difficult to draw general conclusions about the thermal performance of low-flow and high-flow solar heating systems that are valid for a large variety of solar heating systems under many operating conditions from a small number of tests or calculations.

In an experimental comparison of low-flow and high-flow solar heating systems two approaches can be followed:

1. Detailed tests of the systems are conducted and the results used for verification of calculation models. That is, information about the system properties is collected, followed by performance calculations with the models. In this case, tests will be performed under extreme conditions in order to characterize the various system properties. Subsequent calculation then produces the desired annual system performance under normal operating conditions.

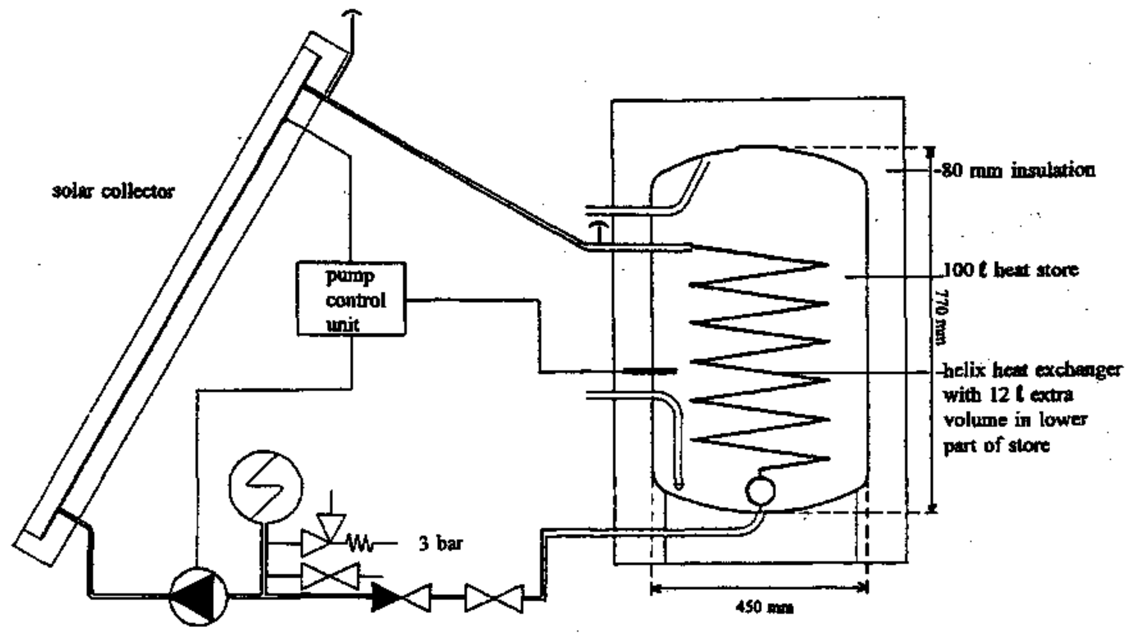
2. A less thorough investigation of the systems is conducted that does not reveal detailed information on system properties or long-term system performance. In this case, tests are performed for normal conditions. A comparison is made only for those conditions and no extrapolation to yearly performances is made.

Inevitably, the first approach is more extensive and it can reveal much more information than the second approach.

In this study the second approach was followed to make results more quickly available. Two different solar DHW systems were tested under the same conditions in an indoor solar simulator. Both systems were preheating systems. A high- and a low-flow rate was used in the solar collector loop.

8.2. Description of the Tested Systems

There were two solar pre-heat systems with remote heat storages tested. One used a helix and the other a mantle heat exchanger. For both systems, the solar collector, collector pump and pump control system were the same. The collector circuit was always filled with water, also during periods without sufficient solar irradiation. A check valve prevented undesirable backwards thermosiphoning. Both systems are made as suitable as possible for low flow. However, piping and pump are conventional. Figure 8-1 shows both systems and Table 8-1 lists their main characteristics.



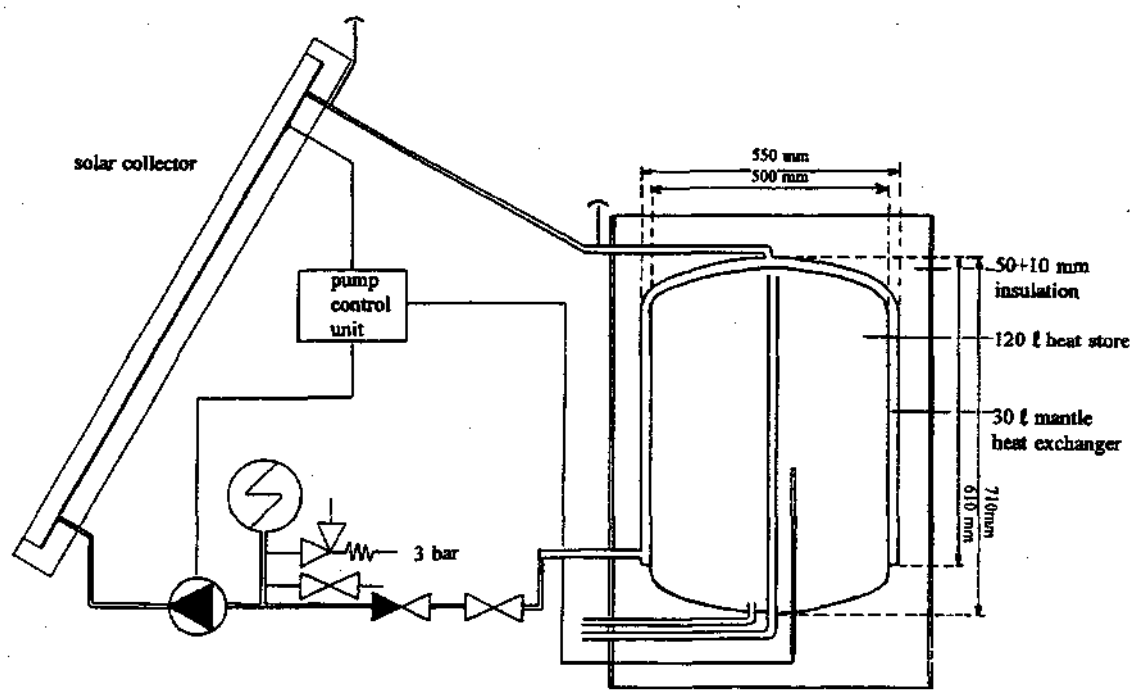


Figure 8-1. Scheme of the Tested DHW Pre-heat Systems.

Table 8-1. Main Characteristics of the Tested DHW Pre-Heat Systems.

<u>Solar collector circuit</u>	
•collector area	2.67 m^2
•collector efficiency [8-5]	$\eta = 0.815 - 3.5 \cdot T^* - 0.018 \cdot G \cdot (T^*)^2$
•insulated collector piping	20 m
•pump control sensor on collector	on absorber back side near the top
•pump control sensor in heat storage	helix store: In tap water part, 30 cm from bottom mantle store: In tap water part, 18 cm from bottom
•pump control	$\Delta T_{on} = 10$ K, $\Delta T_{off} = 2$ K
•pump power	30 W
<u>Helix heat storage</u>	
•tap water volume	100 ℓ
•helix plus extra volume in lower part of storage	12 ℓ
•insulation	80 mm polyethylene
•storage and helix material	stainless steel
<u>Mantle heat storage</u>	
•tap water volume	120 ℓ
•effective heat storage volume in tap water part	105 ℓ
•mantle volume	30 ℓ
•insulation	50 mm caril + 10 mm polyethylene
•storage and mantle material	stainless steel

8.3. Test Procedure

The two systems were tested in an indoor solar simulator test facility, [8-6].

Each system was tested at a high-flow rate of about 2.3 //minute corresponding to about 0.9 //minute per m^2 solar collector and at a low-flow rate of about 0.5 //minute corresponding to about 0.2 //minute per m^2 solar collector in the solar collector loop.

The duration of each test was about 3 days. Figure 8-2 shows the total irradiance on the solar collector and the ambient air temperature of the collector during the 3-day test. The weather data were changed every half hour. The irradiance profile was derived from the Test Reference Year for De Bilt, the Netherlands.

The ambient air temperature of the heat storage was about 20°C.

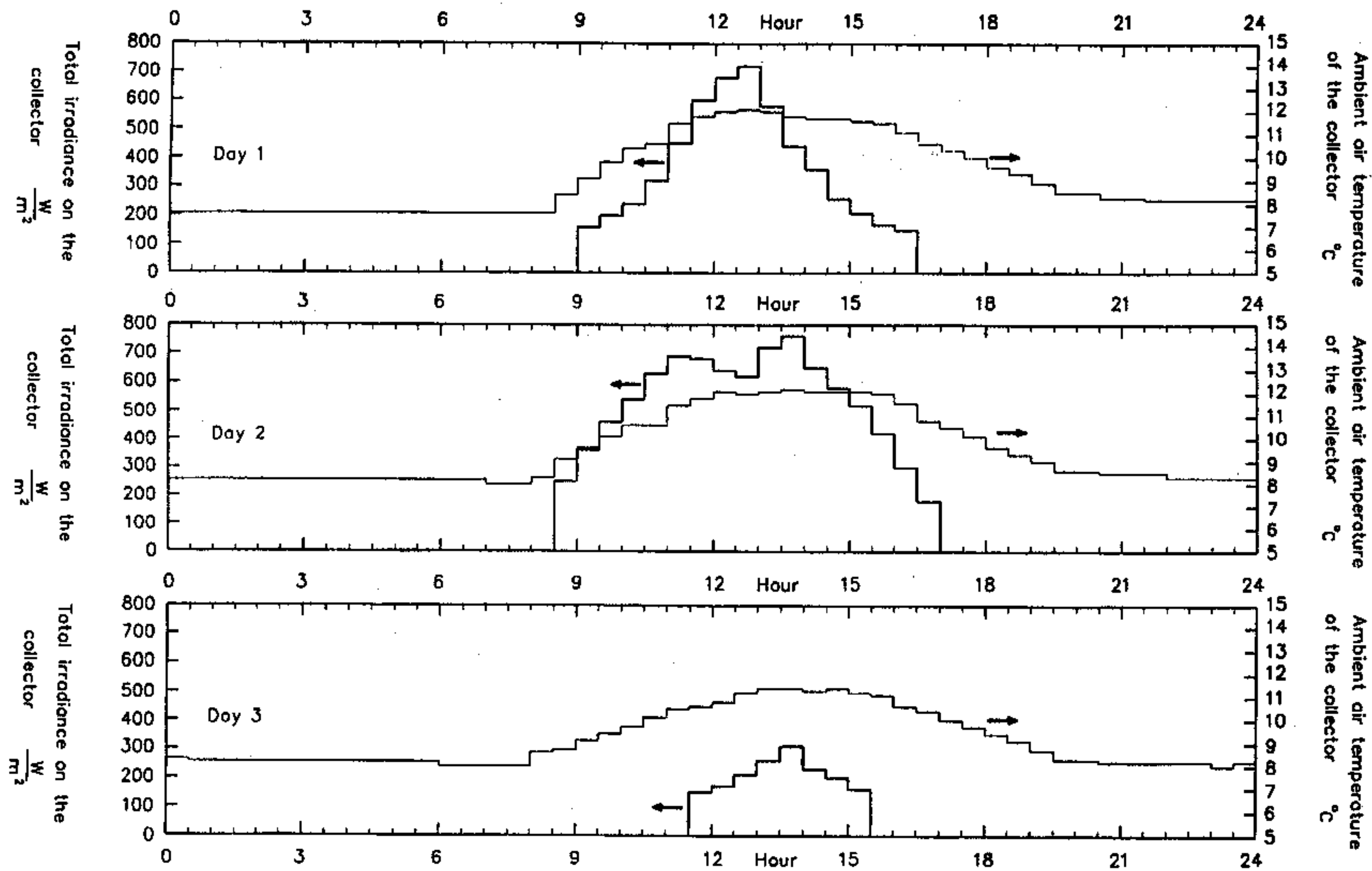


Figure 8-2. Weather Data for the Three-Day Test Period.

Table 8-2 shows the hot water consumption during the test. The cold water temperature was about 15°C and the hot water temperature was 65°C at maximum, *i.e.* if the temperature of the tapped water was above 65°C in the 36.7 ℓ tappings, a smaller water volume was tapped. In this case, an energy quantity corresponding to 36.7 ℓ of water, heated from 15°C to 65°C, was tapped. Tappings no. 11 and 12 reveal the energy left in the storage after the three-day period.

Table 8-2. Hot Water Consumption During the Test Period.

Day no.	Hour	Hot water tapping	
		No.	Volume
1	8^{30}	1	tapping until steady state
	12^{00}	2	36.7 ℓ
	18^{00}	3	36.7 ℓ
2	8^{00}	4	36.7 ℓ
	12^{00}	5	36.7 ℓ
	18^{00}	6	36.7 ℓ
3	8^{00}	7	36.7 ℓ
	12^{00}	8	36.7 ℓ
	18^{00}	9	36.7 ℓ
	18^{30}	10	36.7 ℓ
	19^{00}	11	100.0 ℓ
	20^{00}	12	100.0 ℓ

8.4. Test Results

The tests and the test results are described in detail in [8-7]. An overview of the results is given below. Table 8-3 shows the volumes and energy quantities of the various tap water draw-offs for the system with the helix heat exchanger, both for the high and low-flow regime. For every day, subtotals of the volume and energy draw-offs have been made. For the third day, the energy contents in the final large draw-offs have been summed additionally. Moreover, sums have been made for the draw-offs of all days. Table 8-4 presents the same overview for the system with the mantle heat exchanger.

Table 8-3. Tapped Energy Quantities for the Helix System, Both for High and Low Flow.

Day no.	Draw-off no.	Hot water demand		High flow		Low flow	
		Volume [l]	Energy [MJ]	Volume [l]	Energy [MJ]	Volume [l]	Energy [MJ]
1	2	36.7	7.6	37.2	2.2	36.6	2.4
	3	36.7	7.6	36.2	3.7	37.0	3.8
	total day 1	73.3	15.1	73.4	5.9	73.6	6.2
2	4	36.7	7.6	37.0	3.0	37.2	2.9
	5	36.7	7.6	36.6	4.4	36.4	4.8
	6	36.7	7.6	36.4	6.0	36.8	6.4
	total day 2	110.0	22.7	110.0	13.5	110.4	14.0
3	7	36.7	7.6	37.2	5.0	36.8	4.9
	8	36.7	7.6	36.2	3.2	37.0	3.0
	9	36.7	7.6	36.0	2.1	37.0	2.3
	10	36.7	7.6	37.2	2.0	36.2	1.9
	subtotal day 3	146.7	30.3	146.6	12.4	147.0	12.2
	11			100.0	1.7	100.0	1.5
	12			100.4	0.7	100.2	0.6
	total of 11 and 12			200.4	2.4	200.2	2.1
	total day 3			347.0	14.7	347.2	14.3
1-3	total of 2-9	293.3	60.5	292.8	29.7	294.8	30.5
	total of 2-10	330.0	68.1	330.0	31.7	331.0	32.4
	total of 2-12			530.4	34.1	531.2	34.5

Table 8-4. Tapped Energy Quantities for the Mantle System, Both for High and Low Flow.

Day no.	Draw-off no.	Hot water demand		High flow		Low flow	
		Volume [l]	Energy [MJ]	Volume [l]	Energy [MJ]	Volume [l]	Energy [MJ]
1	2	36.7	7.6	36.0	1.9	36.4	2.8
	3	36.7	7.6	37.4	3.4	37.0	3.5
	total day 1	73.3	15.1	73.4	5.4	73.4	6.3
2	4	36.7	7.6	36.8	2.9	37.0	3.0
	5	36.7	7.6	37.0	4.4	35.8	5.4
	6	36.7	7.6	36.0	5.6	36.8	6.0
	total day 2	110.0	22.7	109.8	12.9	109.6	14.3
3	7	36.7	7.6	36.8	5.0	36.8*	4.8*
	8	36.7	7.6	36.4	3.4	36.6	3.6
	9	36.7	7.6	36.6	2.6	36.6	2.7
	10	36.7	7.6	37.2	2.6	36.6	2.6
	subtotal day 3	146.7	30.7	147.0	13.5	146.6	13.7
	11			100.2	3.9	100.2	3.6
	12			99.8	0.8	100.0	0.8
	total of 11 and 12			200.0	4.7	200.2	4.4
	total day 3			347.0	18.2	346.8	18.1
1-3	total of 2-9	293.3	60.5	293.0	29.2	293.0	31.8
	total of 2-10	330.0	68.1	330.2	31.8	329.6	34.4
	total of 2-12			530.2	36.5	529.8	38.8

* A power failure at the test facility on the beginning of Day 3 resulted in a 38-minute delay to the start of the solar irradiance schedule. Test time was extended to accommodate the difference. The first draw for Day 3 scheduled for 8 a.m. was carried out at 11 a.m. The effect on the thermal performance of the system is considered to be minor.

Other test results:

- For the low-flow regime, higher tap water temperatures were measured than with high flow. This result was most pronounced for the mantle system at the midday draw-off, *i.e.*, after a relatively cold start for the top level of the heat storage in the morning. In that case, differences in tap water temperatures of over 10 K were observed.

- During the first day, for low flow, the collector pump was in operation for a longer period, about half an hour. For the other days, it was about the same. This was observed for the helix as well as the mantle system.

Discussion of the results:

- In the discussion of the results below, no comparison is made between the thermal performance of the helix and mantle system as this was not the aim of the tests. The aspect under investigation is the difference in thermal performance between low-flow and high-flow operation. This difference has been determined for two specific solar DHW systems.

- For the helix system, the solar fractions for high- and low-flow operation are 46% and 47%, respectively, for the draw-offs 2-10. For the mantle system, these fractions are 47% and 51%, respectively. These solar fractions correspond well with the annual solar fraction calculated for similar systems in the tests using meteorological data of TRY - De Bilt, Netherlands, for a demand of 110 liters per day, heated from 15°C to 65°C.

- For the helix system, the measured difference in thermal performance between low-flow and high-flow operation is 1 - 3%, depending on whether energy left over in the storage after draw-off of 330 liters is taken into account and whether draw-off no. 10 is considered. Notice the measuring error is about the same.

- For the mantle system, this difference is greater, 6 - 9%.

- Once again, notice that the differences in thermal performance between low-flow and high-flow operation as discussed above are valid for the conditions during the three-day test, and cannot be extrapolated to predict the annual system performance.

8.5. Conclusions

For well-designed, high-flow systems such as the two tested, low-flow operation can obtain slightly greater thermal performance than that of high flow for a choice of realistic meteorological and draw conditions. The difference in thermal performance between high-flow and low-flow operation appears to be larger for the tested mantle system.

8.6. Final Remarks

When designing a system, comparison testing with high and low flow can provide guidance for the choice of flow regime. In this regard, improvement of thermal stratification by changing from high flow to low flow is of major interest. Valuable information can be obtained with respect to the flow regime by these comparative tests without the use of computer models.

In the tests, two different solar pre-heat systems which closely match those on the Dutch market have been investigated for specific meteorological and tap water draw-off conditions. The results are specific to the systems and the test conditions. Broader conclusions cannot be drawn for other solar DHW systems and conditions.

Both tested systems had well-stratified heat storages for high-flow as well as for low-flow operation. Therefore, the thermal advantage of low-flow operation was relatively small.

The difference in thermal performance between low- and high-flow operation is larger if the difference in the thermal stratification in the heat storage is greater. In the investigations, thermal stratification was most improved for the tested mantle system.

Furthermore, if a system is optimally designed for low-flow operation, the extra thermal performance obtained by reducing the flow rate would be greater than found in the tests.

For extensive research on a vast variety of solar DHW system types under different meteorological and tap water draw-off conditions, the first approach mentioned in Section 1 must be used. Through verification of mathematical models and subsequent calculation of system performance annual system performance can be predicted as well. With this approach, models need to be verified on a rather detailed level, which requires considerable effort.

9. COUNTRY INFORMATION AND STATISTICS

9.1. Introduction

This chapter presents a common set of statistics and other information about each country. A tabular presentation makes it easy to contrast country activities and approaches. The information provided illustrates that circumstances vary widely from country to country and provides insight into why each country took a different approach to task activities.

9.2. Tables

The country information is organized into four tables.

Table 9-1 provides information on the climate factors that are most relevant to solar DHW system performance. As shown, these conditions can vary greatly between and within countries. Although conditions vary only slightly within smaller countries, within larger countries they can vary dramatically.

Table 9-2 lists information on government and utility initiatives, regulations, and consumer characteristics that can influence solar DHW system design and development paths.

Table 9-3 gives key statistics about the solar industry, consumers, and the economic environment in which solar must compete.

Some of the Task 14 Solar DHW Systems Working Group meetings have included a solar DHW industry workshop in which the industries of the host country and Task 14 industry representatives made presentations, exchanged information, and discussed issues and common interests. Table 9-4 provides information on these workshops.

For further details on the information presented see Appendix B.

Table 9-1. Climate.

Climate Characteristics	Canada	Denmark	Germany	Netherlands	Spain	Switzerland	United States
Locations	southern regions	all	all	all	southern regions	lower areas	all
Annual radiation on tilt=latitude GJ/m^2	4.4 to 6.4	4.0 to 4.1	3.8 to 5.2	3.9 to 4.4	6.8 to 7.5	4.1 to 5.0	5.3 to 9.1
Locations	three locations	all	all	all	southern regions	Kloten	four locations
Average daytime temperature °C	10 to 15	8.7	8.7	10.2	17.5	8.6	13 to 24

Table 9-2. Infrastructure and Demographics.

Subject	Canada	Denmark	Germany	Netherlands	Spain	Switzerland	United States
Important government initiatives	S-2000 Program to encourage solar DHW utility activities	energy tax CO_2 tax CO_2 reduction water tax up to 30% subsidy	income tax deduction national subsidy state subsidies	subsidies awareness campaigns long-term agreements with industry	national and regional subsidies	Program Energy 2000 subsidies for multi-family	state tax credits agency subsidies
Important utility initiatives	pilot projects market survey long-term funding	demonstrations	utility programs	leasing programs	pilot projects R&D investment	none	DSM subsidies peak penalty pricing
Important regulations	none	subsidized systems must be tested and approved	none	double-walled HX or drain back	technical guidelines that are obligatory for subsidies	none	double-walled heat exchangers
Population	25 million	5.2 million	80.7 million	15.5 million	40 million	7 million	260 million