

Experimental evaluation of stagnation-safe flat plate collectors with heat pipes for domestic hot water preparation in thermosiphon systems

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Abstract

Overheating in thermosiphon systems, which represent the dominating solar thermal technology worldwide and are often installed in regions with water scarcity, can lead to significant water losses and to failures of the safety valve. The paper analyses the behavior of a prototype thermosiphon system, provided with a novel flat plate collector, by comparing it with a conventional system based on experimental investigations. The novel collector is equipped with specially designed heat pipes, offering an intrinsically safe solution to limit the stagnation load in the heat tank up to a desired temperature. The results of the performance tests according to ISO 9459-5 show, that the prototype system can achieve similar solar yields as the standard system (-2% to -4%, depending on the location). Stagnation measurements report a maximum temperature of 96 °C in the heat tank of the prototype. The conventional system reaches temperatures above 100 °C, which leads to a pressure increase of 3.4 bar.

Keywords: thermosiphon systems; flat plate collectors; heat pipes; stagnation, overheating protection

1. Introduction

Thermosiphon systems (TSS) dominate the solar thermal market, representing more than 75 % of the total and about 90 % of the new installations worldwide (Weiss et al. 2019). Thus, the development of efficient and cost-effective thermosiphon systems based on heat pipes has a huge potential. Heat pipes are already well-established in evacuated tubular collectors as devices to transfer the heat from the absorber plate to the solar circuit or to the hot water tank. Flat plate collectors with heat pipes, on the contrary, have only been realized within research projects so far (Jack et al. 2014).

In TSS, heat pipes offer a simple system hydraulics and enable the thermal decoupling of the absorber from the heat tank. On the basis of the so-called dry-out limit - an inherent physical effect of heat pipes - the cyclic process and respectively the heat transfer can be suppressed above a certain temperature, by choosing suitable design parameters (type and quantity of the heat carrier). For TSS the reference temperature is set below 100 °C in order to prevent boiling and the loss of water by the safety valve in case of continued overheating. It must be considered that these systems are often operated at locations where drinking water is a scarce resource. Moreover, in regions with calcareous water, a frequent opening of the safety valve at high temperatures often leads to its failure (Zörner et al. 2010). In contrast to commercially available TSS, which are already offered with cooling and safety solutions (Wagner 2014), the use of heat pipes offers a simple, cost-effective and at the same time intrinsically safe way of limiting the temperature.

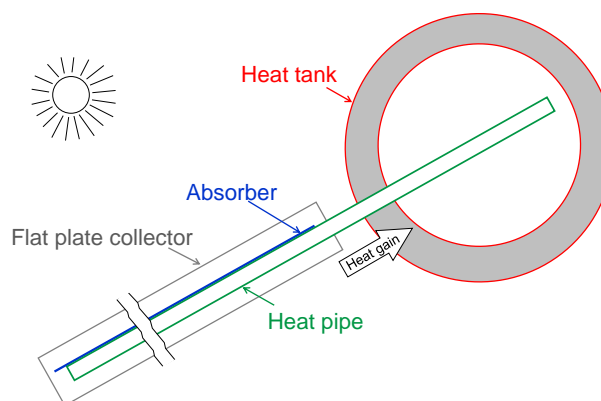


Figure 1: Section of a TSS with heat pipe in the heat flow path between the absorber plate and the domestic hot water tank

2. Experimental investigation of the system behavior

2.1. Experimental setup

For a holistic assessment, we carry out comparative experimental investigations on a prototype and a standard system. The standard system is used as reference and features a classical double circuit (closed loop System) with a common flat plate collector and a double shell tank (see Figure 2, left). Thus, the natural circulation of the collector circuit fluid only runs through the outer jacket of the heat tank. The prototype system exhibits a flat plate collector with novel aluminum heat pipes. These are metallically connected to the absorber along the evaporator section and their condenser project into the heat tank (see Figure 2, right). A water-glycol mixture as well as a complex double shell heat tank are not necessary in this case. Both TSS have a collector gross area of 2 m² and a tank volume of 200 l, which represents a usual dimension for such solar thermal water heaters in Mediterranean locations. Further specifications of collectors and heat pipes are given in Table 2 (see Appendix). Figure 2 shows the standard system (a) and the heat pipe system (b) during tests at ISFH.

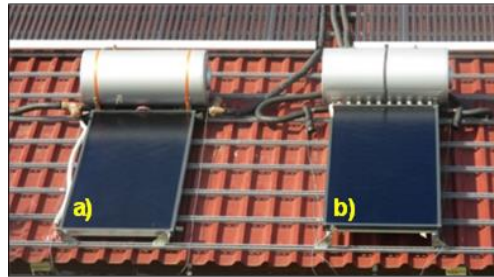


Figure 2: Experimental outdoor facility: a) standard and b) heat pipe TSS

To carry out dynamic system tests (DST), we installed a test setup according to (ISO 9459-5 2007), which can emulate hot water taps under software control (see Figure 3, left). Using suitable sensors, the energy balance around the heat tank can be determined and the system yield as well as the system performance can be specified. In addition, we installed further sensors for temperature and pressure measurement at the heat tank to evaluate the thermal loads during stagnation periods. Furthermore, we use a bypass with a pump for conditioning the heat tank by an electric heater up to high temperatures as shown in Figure 3 right. This procedure is exclusively intended to initiate stagnation sequences.

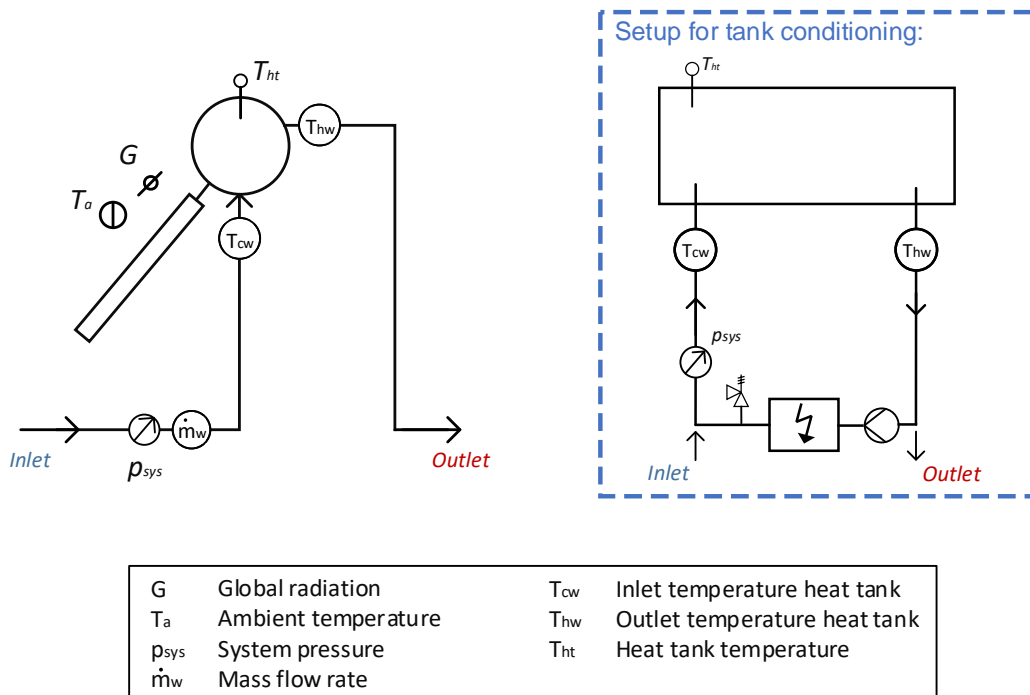


Figure 3: Test setup and measurement devices for dynamic system tests with TSS according to ISO 9459-5 (left) as well as for conditioning the heat tank before stagnation sequences (right)

2.2. DST System parameters

The aim of the DST procedure is to determine the performance of complete solar thermal systems under real conditions. The system is investigated under characteristic operating conditions by setting defined test sequences. Each sequence represents a separate period of measurement and needs several days. These data can be used in a system simulation to determine the annual yield. The identified model parameters of both systems are shown in Table 1.

Tab. 1: Identified model parameters of the dynamic system testing procedure for both the standard system (a) and the heat pipe system (b)

Parameters	Symbol	Unit	Standard (a)	Heat Pipe (b)
Effective collector loop area	A_C^*	m ²	1.05	1.04
Heat-loss coefficient of the collector loop	u_C	Wm ⁻² K ⁻¹	4.76	5.43
Heat-loss rate of the store	U_S	WK ⁻¹	2.21	1.94
Heat capacity of the heat tank	C_S	MJK ⁻¹	0.79	0.77
Fraction of the store heated by the auxiliary heater	f_{aux}	-	-	-
Draw-off mixing parameter	D_L	-	0.055	0.036
Collector loop stratification parameter	S_C	-	0.015	0.019

We can see that the effective collector area A_C^* of both systems differs only slightly. The real collector area and the optical properties of both collectors are identical, so that the values for A_C^* are directly comparable. According to (Spirkl 1990), A_C^* can be used to draw conclusions about the conversion factor η_0 and thus the quality of the thermal connection of the solar absorber to the heat tank. The value for the heat pipe system (b) is approx. 1 % lower, so that we can conclude that the heat transport capacity of both systems is almost equal. The effective heat loss coefficient u_C describes the sum of the thermal losses occurring in the collector circuit. Based on the measurement results, we found a loss coefficient of 5.4 W/m²K for the heat pipe system (b), which is about 0.7 W/m²K higher compared to the standard system (a). This effect is probably caused by the heat pipe-based temperature limitation, which can minimally affect the collector performance at high operating temperatures. However, we cannot clearly analyze these effects due to its compact design as a thermosiphon system. In addition, we can see that the heat loss rate of the heat tank U_S is correspondingly higher in the standard system (a). Both heat tanks are identical in volume and dimensions of the drinking water part (ratio of diameter and width), whereby a double shell tank is used in the standard system (a). Thus, the heat input due to natural circulation of the collector circuit fluid through the outer jacket. In the prototype system (b), the heat is transferred directly to the inner part of tank by the heat pipes, whereby the outer jacket is not used. The air filled outer jacket of (b) can lead to lower heat losses and thus to lower values of U_S . The heat tank capacity C_S of both systems differ only minimally, but this may also be due to the uncertainty of the procedure of parameter identification based on individual measurement data sets. The proportion of the auxiliary volume f_{aux} is typically > 75 % for thermosiphon systems, which are usually equipped with an electric heating rod. According to (ISO 9459-5 2007), the auxiliary heating must be deactivated during the tests and the parameter f_{aux} is not considered. The parameter D_L describes the demixing of the heat tank during tapping and S_C the stratification at the heat tank due to the input by the solar loop. Both parameters are in a similar range so that it will not be further discussed here. Because some parameters, especially u_C and U_S , are not completely independent of each other, it is advisable to consider the entire parameters in the context of a long-term prediction and quantify the total system performance instead.

2.3. System performance in operation mode

The system performance is expressed by the annual yield and was determined by operating the two systems according to the DST procedure described in (ISO 9459-5 2007). The results depend on the tapping rates and on the climatic conditions of the considered locations. In general, the differences between standard (a) and heat pipe (b) system are almost negligible for small tapping rates and increase with higher heat demands, as shown in Figure 4. The solar yield of the heat pipe system is reduced by a maximum of 5 % compared to the standard

system. In Athens, representing a typical climate for TSS, the deviations in annual yield ranges between 2 and 3 %. This confirms the result of the parameters consideration in section 2.2, that both systems are almost on the same performance level.

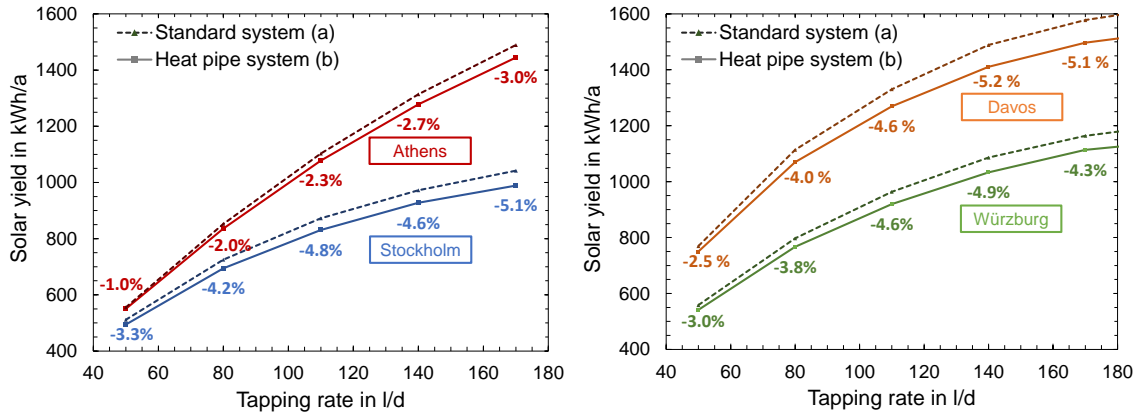


Figure 4: Solar yield of the standard system (a) and the heat pipe system (b) for all considered locations against the tapping rate

Figure 5 shows the energy balances of both systems for all considered locations at a typical tapping rate of 80 l/d. The annual hot water demand represents the sum of solar yield and auxiliary energy demand. Depending on the location, the solar yield of the heat pipe system (b) is between 17 kWh/a (Athens) and 44 kWh/a (Davos) lower, which leads to a corresponding increase in the auxiliary energy demand. Compared to the standard system (a), the maximum relative reduction of solar yield occurs in Stockholm with 4.2 %. In Athens, on the other hand, the relative deviation in solar yield is only 2 %. The deviation of the system results may be mainly due to the difference of the parameters A_C^* and u_C . However, under consideration of the uncertainty of the DST method of about 5 % (Visser et al. 1997) the deviations of the prototype system can be negligible.

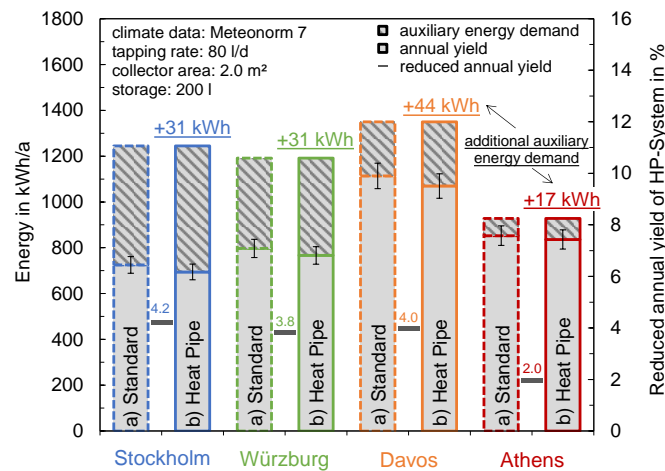


Figure 5: Annual energy balances of the standard system (a) and the heat pipes system (b) for all considered locations at a tapping rate of 80 l/d

2.4. Stagnation mode

In addition to the behavior in operation mode, we also investigate the case of stagnation using a suitable test procedure. For that, we electrically warm up both heat tanks to a temperature level of $> 95^\circ\text{C}$ on a sunny day and mixed up the heat tank volume by the pump. After conditioning the heat tank, we adjust the system overpressure to about the same level (1 bar) and leave both systems exposed to the solar irradiation. This procedure should imitate several days of standstill, i.e. no tapping from the heat tank, at full solar irradiation.

Figure 6 shows the corresponding temperature and pressure curves for such a stagnation sequence. The heat tank temperature of the heat pipe system (b) remains almost constant after the conditioning procedure and amounts to a maximum temperature of 96°C , while the absorber temperature rises to 188°C . This results in a temperature

gradient of approx. 90 K between the solar absorber and the heat tank due to the heat pipe-based power shut-off. In the standard system (a), the heat tank temperature increases to a maximum of 109 °C, which initially only means a 13 K higher thermal load compared to the heat pipe system (b). However, when looking at the system over pressure, there is a significantly pressure increase in the standard system of approx. 3.7 bar as consequence of the further increasing heat tank temperature. In contrast, the pressure in the heat pipe system remains almost unchanged.

Beforehand we expected a higher thermal load in the standard system. This can be explained by the meteorological conditions of the outdoor measurements at the ISFH. We further assumed that higher stagnation loads occur at southern European or North African locations with higher solar irradiation and higher ambient temperature. However, it should be noted that with a 3 bar safety valve or a higher outlet pressure (2 – 4 bar), which often corresponds to the pressure in drinking water systems, the safety valve would have opened several times and drinking water would have been lost.

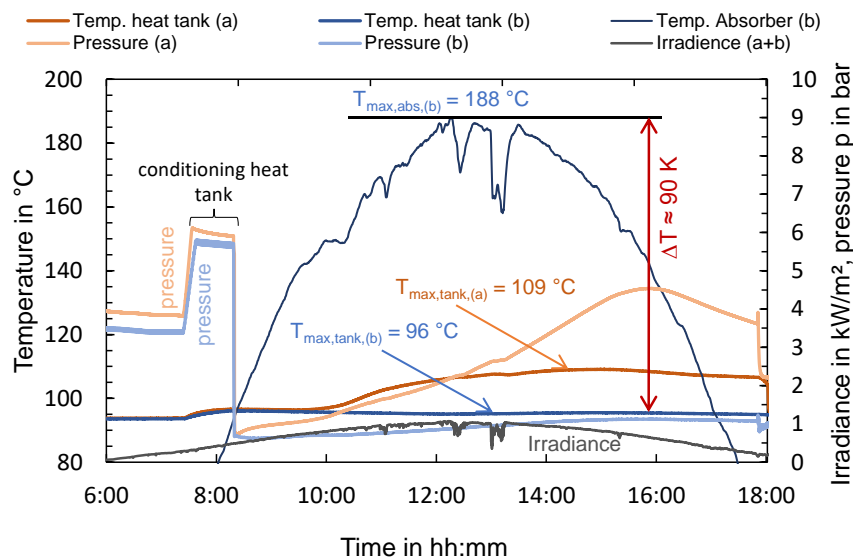


Figure 6: Temperatures and system overpressure of the standard system (a) and the heat pipe system (b) during a stagnation period at the ISFH test field

3. Cost reductions

Figure 7 shows the expected investment costs of a heat pipe system in relation to a standard system (without safety solution). The heat pipe system corresponds as far as possible to the prototype-tested version in the previous sections. We initially assumed that the costs for the collector with absorber-heat pipe solution do not differ from the standard case. According to the statement of the manufacturer, it can be expected that the additional effort for the aluminum heat pipes production can be compensated by the complete substitution of the expensive copper piping of the standard collector. A significant saving compared to state-of-the-art systems is estimated at the collector hydraulic, because the solar circuit is no longer necessary due to the use of heat pipes. Thus, a hydraulic circuit between collector and heat tank are not required. This also eliminates the costs of the solar fluid (water-glycol mixture). In addition, the costs of the drinking water tank can decrease, as a double shell tank can be replaced by a simple tank with connections for the heat pipes, which is easy to manufacture for the tank producer. Due to the elimination of collector piping, the installation procedure will be much easier. However, the additional effort to install the collector to the heat tank (inserting and sealing the heat pipes in the heat tank) will compensate this advantage. Thus, no change in the installation procedure is supposed. Overall, a cost-saving potential of about 15 % can be expected for the heat pipe-based system in terms of investment costs.

In practice, TSS with overheating protection leads to a significantly lower frequency of faults and failures. This is the reason for systems with appropriate safety solutions, which are already being offered today (Wagner 2014). The heat pipe collector automatically leads to a temperature limitation because of the thermo-physical effects inside the heat pipe, which are independent of additional mechanical components and completely intrinsically safe. In contrast, conventional safety solutions are usually based on complex cooling or compensation systems.

Compared to such systems, heat pipe based TSS can be much more cost-effective. Compared to TSS without temperature limitation, further advantages are possible due to the possible longer service life of some components as well as lower maintenance and service costs. Currently, neither any concrete data nor experience reports are available. Within the framework of planned demonstration systems, these expectations are to be checked in practice.

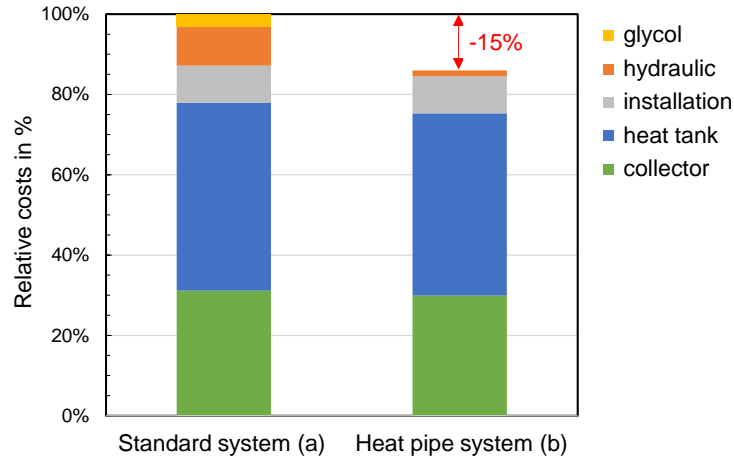


Figure 7: Expected investment costs (components and installation) of a novel heat pipe based TSS in comparison to a comparable standard system

4. Conclusion and outlook

We investigated the performance of a novel thermosiphon system with a heat pipe-based flat plate collector using dynamic system tests (DST). That prototype system achieved almost the same system performance as a comparable standard system. Depending on the location, the solar yield of the heat pipe system is less than 5 % for common domestic hot water loads. For the climate of Athens, representative for most TSS-typical locations, the solar yield is only 2 % below that of the standard system.

In addition to the system performance, we investigated the stagnation behavior using appropriate test sequences. For this, we emulated the standstill of the systems (no hot water tapping for several days) at the ISFH test field. The results show that the maximum heat tank temperature in the heat pipe system can be limited intrinsically safe to 96 °C, whereas the maximum heat tank temperature in the standard system was measured at 109 °C. We had previously been expected higher temperatures for the standard system. However, the temperature increase of 13 K results in a corresponding pressure increase. The maximum system pressure is about 3.4 bar higher than in the heat pipe system. Depending on the mains pressure, such an increase can lead to continuous triggering of the safety valve. Depending on the availability of drinking water, this means on the one hand the loss of a valuable resource. On the other hand, a permanent opening of the safety valve at high temperatures leads to leaks and ultimately to its failure (especially in regions with calcareous water). It is to be expected that this problem can be reduced by limiting the further overheating of the heat tank, even if it is only 10 – 20 K. Unfortunately, concrete data and experience reports from thermosiphon systems at locations with high solar irradiation are difficult to obtain.

An estimation of the component costs shows that the main advantages of heat pipe based thermosiphon systems result from the much simpler system hydraulic. Overall, the considered heat-pipe system is expected to have 15 % lower investment costs compared to the state of the art. In prospective activities, the heat pipe systems need to prove in real system operation so that statements on lifetime and maintenance costs are determined.

5. Acknowledgments

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7. Appendix

7.1. Specifications of the collectors and heat pipes

Tab. 2: Specific collector / heat pipe parameter for the standard system (a) and the heat pipe system (b)

Parameter	Standard (a)	Heat Pipe (b)
Heat pipe evaporator length	-	1.850 m
Heat pipe transport zone length	-	0.172 m
Heat pipe condenser length	-	0.287 m
Heat pipe diameter (evaporator)	-	0.008 m
Heat pipe diameter (condenser)	-	0.008 m
Number of heat pipes per collector	-	10
Heat pipe heat carrier fluid	-	Butane (3.74 g)
Maximum temperature heat pipe process	-	107 °C
Aperture area collector	1.83 m ²	1.83 m ²
Cross area collector	2.02 m ²	2.02 m ²
Heat tank volume	200 l	200 l