Experimental evaluation of evacuated tube collectors with heat pipes to avoid stagnation loads in a domestic hot water system

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Abstract

Heat pipes in solar thermal collectors can reduce thermal loads in the solar circuit by using the physical effect of dry out limitation. By avoiding high temperatures and vapor formation, simplified, more reliable and cost effective solar thermal systems can be designed. The paper investigates a newly developed evacuated tube collector based on optimized heat pipes able to limit the temperature loads at the manifold up to a desired value. On the basis of efficiency measurements on a prototype heat pipe collector, we determine the annual yield of a collector field in a domestic hot water system compared to an identical system with direct flow collector according to ISO 9459-5. The results are validated by simulations with TRNSYS and show no significant difference between the performances of the two systems. By means of extensive outdoor stagnation tests on the same fields, we report a maximum temperature of 125 °C in the solar circuit of the heat pipe system, which is 95 K lower compared to temperature measured in the reference system.

Keywords: heat pipe, stagnation temperature, overheating protection, evacuated tube collector, dynamic system testing

1. Introduction

Heat pipes in solar thermal collectors are state-of-the-art devices for the heat transfer from the absorber plate to the solar circuit. In comparison to direct flow collectors the heat pipe represents an additional thermal resistance in the heat flow path. Therefore, a high thermal performance of the heat pipes is essential to achieve a high collector efficiency. Beside the simplified collector hydraulic the use of heat pipes provides the advantage of decoupling the absorber plate from the fluid circuit. If the two-phase flow inside the heat pipe is suppressed beginning from a certain temperature, no more heat is transferred from the absorber plate to the manifold. By taking advantage of this physical effect, which is called the dry-out limit of heat pipes, the maximum temperature in the solar fluid can be limited to reduce thermal loads. The shut-off behavior and the maximum temperature are mainly determined by the type and the mass of working fluid in the heat pipe. Above this temperature, the fluid exists only in the vapor phase, so that the heat pipe heat transport ability is disabled. The maximum temperature in the connected solar circuit can be limited to such an extent that the evaporation of the solar circuit fluid can be completely avoided. In contrast to other technologies, the temperature limitation by heat pipes is inherently safe and independent from thermomechanical devices. The suppression of vapor formation in the solar circuit leads to simplified and more reliable system configurations and to a reduction of the overall system costs.

2. Evacuated tube collector prototype with novel heat pipes

We developed and investigated an evacuated tube collector prototype with novel heat pipes by means of indoor performance measurements with a sun simulator according to EN ISO 9806. The evacuated tubes with pentane-filled heat pipes were manufactured in cooperation with the German company NARVA Lichtquellen GmbH & Co. KG. The prototype collector consists of ten single tubes, which are connected with the solar circuit by a typical manifold. The collector power curve was measured at an average wind speed of 3 m/s and an ambient temperature of about 26°C. Figure 1 shows the measurement results and compares the efficiency curve of the heat pipe collector to that of a similar direct flow collector (standard collector). The conversion factor of the new prototype amounts to 75.4 % and is about 2 %-absolute lower than the standard collector, which is a typical difference caused by the additional thermal resistance of the heat pipe in the heat transfer path. Up to an operating temperature of about 75 °C both collectors have nearly the same heat loss coefficients. Above this

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temperature the heat loss coefficient of the heat pipe collector changes significantly as a result of the starting dry-out process, as represented by the characteristic kink point of the efficiency curve. As consequence of this, the stagnation temperature in the solar circuit takes place at 125 °C (measured without wind), which is significantly lower compared to the standard collector.

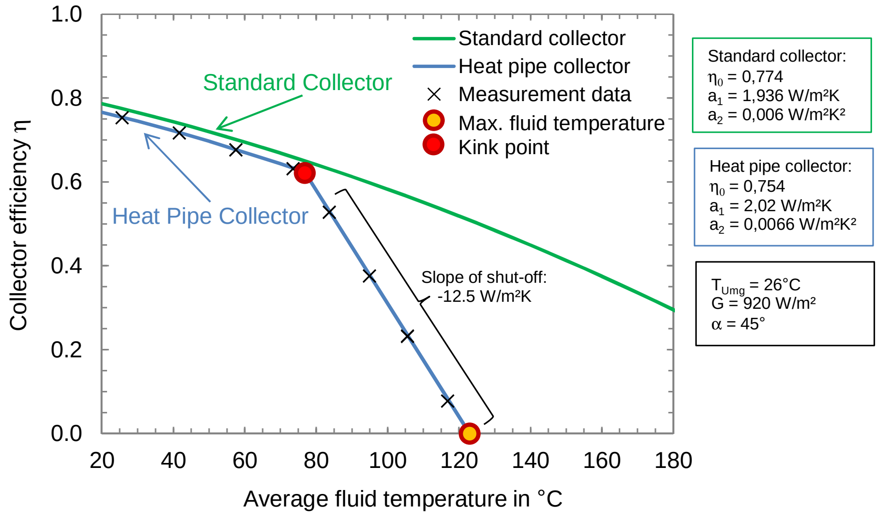


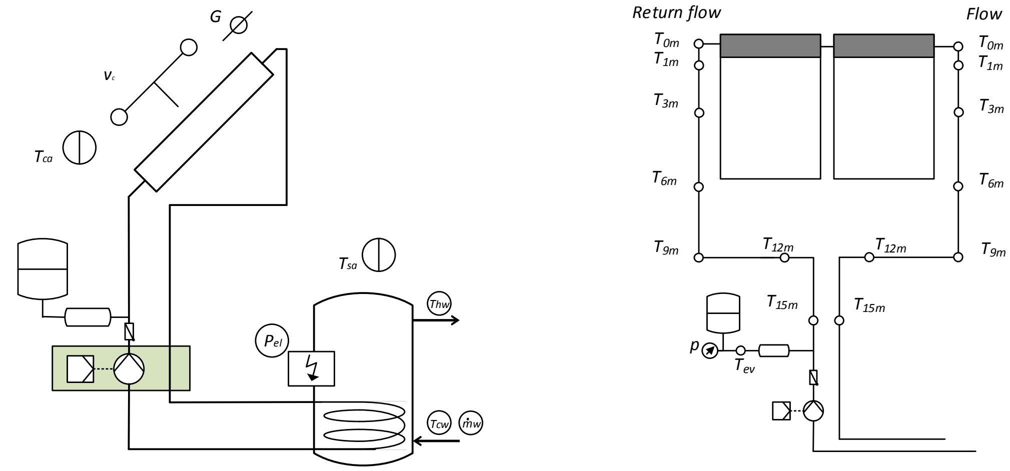
Fig. 1: Efficiency curve and coefficients of the prototype (measurement) and of a similar standard collector (TÜV-Rheinland 2014)

3. Experimental investigation of the system behavior

3.1. Experimental setup

For a comprehensive evaluation of the system behavior of heat pipe collectors with temperature limitation two identical experimental setups were installed on our test roof, which differ only in the collector hydraulic. We compared a prototype collector with deactivating heat pipes with an evacuated tube collector with direct flow as reference (Figure 2). Both systems consist of a field with an aperture area of 6 m² and a hot water tank with a volume of 400 l, which represent a common design for domestic hot water systems. The expansion vessel and all other solar circuit components are dimensioned in conformity with common guide lines. The specific system parameters of both systems are listed in Table 2 in the Appendix. Both solar plants are equipped with the necessary measurement and system technology according to ISO 9459-5 (see Figure 3, left). Additional temperature sensors and pressure measurement devices are installed in both solar circuits for investigating the stagnation loads (see Figure 3, right).

Fig. 2: Test roof with both prototype and reference evacuated tube collector fields



G	Global radiation	T_{ca}	Collector ambient temp.
P_{el}	Power of auxiliary heater	T_{st}	Storage ambient temp.
$\dot{m}_W$	Mass flow rate	T_{cw}	Inlet temp. storage
v_c	Wind speed	T_{hw}	Outlet temp. storage

T_{im}	Temperature of solar piping (i = distance from collector)
T_{ev}	Temperature of expansion vessel
p	System overpressure

Fig. 3: System scheme according to ISO EN 9459-5 (left) and additional sensors in the solar circuit (right)

The aim of the DST procedure is to determine the performance of complete solar 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. A complete data set consists of four measurement files, each with one sequence. 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. The effective collector area *Ac** describe not only the collector geometry but also its optical properties. The slight difference can be explained by the lower conversion factor of the heat pipe collector (see Figure 1). The other parameters are almost in the same range.

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

Parameters	Symbol	Unit	Heat pipe system	Standard system
Effective collector loop area	A_C*	m²	4.44	4.61
Heat-loss coefficient of the collector loop	u_C	$Wm^{-2}K^{-1}$	4.47	4.39
Heat-loss rate of the store	U_S	WK^{-1}	2.60	2.62
Heat capacity of the store	C_S	MJK^{-1}	1.54	1.48
Fraction of the store heated by the auxiliary heater	f_{aux}	-	0.573	0.595[1]
Draw-off mixing parameter	D_L	-	0.017	0.019
Collector loop stratification parameter	S_C	-	0.009	0.005

3.2. System performance in operation mode

The system performance is expressed by the annual yield and was determined by operating the two systems in parallel operation under identical conditions 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 the prototype and the reference system are almost negligible, as shown in Figure 4 (left) at a tapping rate of 200 l/d. For Würzburg, representing middle European climate, the deviation in annual yield amounts to 20 kWh/a (0.8 %). Thus, the auxiliary energy demand is increased by 20 kWh/a (2 %). For locations

[1] The annual yield is simulated with the same fraction values f_{aux} = 0.57 for both systems, because identical storages and auxiliary heaters are used.

with higher solar irradiation such as Athens or Davos the annual yield reduction of the heat pipe system further decreases down to 0.4 %. For locations with lower solar irradiation such as Stockholm, on the contrary, it increases up to 1.2 %.

The quotient of the annual yield Q_{sol} and the whole energy demand for domestic hot water Q_{DHW} is defined as the fractional system gain f_{sol} and is expressed by Equation 1. In general, the fractional system gain for both systems decreases with higher daily tapping rates (see Figure 4, right for Würzburg). For small tapping rates (< 100 l/d), the required temperature level, which has to be reached by the solar collector for contributing to the energy supply, has a negative impact on the performance, due to the prolonged standstill time and the increased heat losses. For common tapping rates of 100 to 300 l/d, the fractional system gain of both solar systems ranges between 77 and 60 %.

$$f_{sol} = \frac{Q_{Sol}}{Q_{DHW}} \qquad \text{(eq. 1)}$$

The deviations of the fractional system gain of the heat pipe system compared to the standard system are almost constant, between -0.5 and -0.8 %-absolute over the whole range of the considered tapping rates. That can be explained by the slightly lower annual yield of the heat pipe system. The losses in annual yield amount to about 1 % compared to the standard system (see Figure 4 right). That leads to higher auxiliary energy demands, which range between 1.5 and 3.5 % (about 2 % for typical tapping rates).

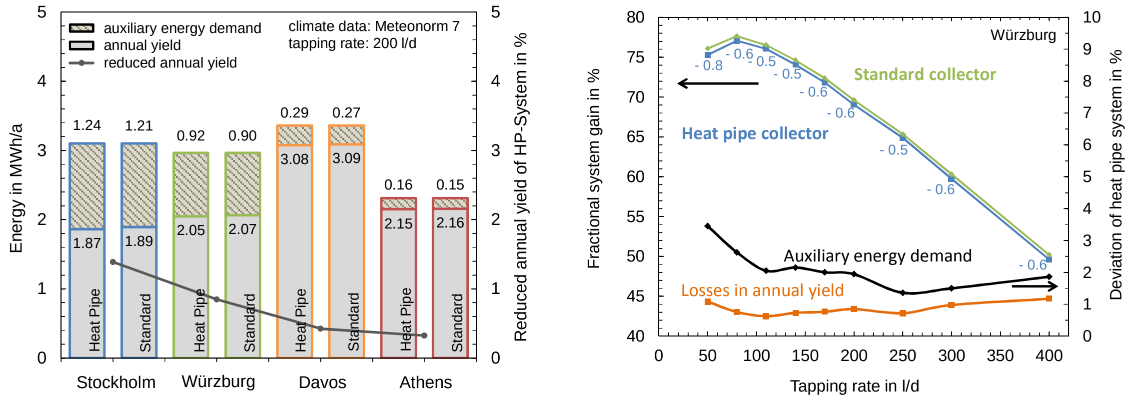


Fig. 4: Annual energy balances for all considered locations at a tapping rate of 200 l/d (left) and fractional system gain, deviations in annual yield and auxiliary energy demand for Würzburg against the tapping rate (right)

3.3. Validation with TRNSYS simulations

To validate the results of the DST procedure, the performances of both systems were simulated in TRNSYS. The characteristics of heat pipe collectors with inherently temperature limitation were implemented in a newly developed collector type model. Conditions such as the tapping and meteorological data were set equal to the DST procedure. The fractional system gain is determined as f_{save} according to IEA TASK 26, as the quotient of the auxiliary energy demand for domestic hot water with solar thermal system and the auxiliary energy demand of the same system without a solar thermal system (Streicher et al. 2003). Thus, storage losses generated by the solar system can be estimated and separated from conventionally generated losses. In the DST model the detailed procedure for calculating the heat storage losses is unfortunately unknown. For validation purpose, the electrical energy demand for the controller and pumping is not considered in both models, since their influence is negligibly small.

In the case of the heat pipe system the fractional system gain calculated according to both approaches have a maximum deviation of 5 %-absolute for all four locations and over a wide range of tapping rates (see Figure 5, left). Small tapping rates show the greatest differences. The reason could depend on the different consideration of the heat losses of the storage tank. In Würzburg and at a typical tapping rate of 200 l/d the fractional system gain differs by 1.0 %-absolute and the auxiliary energy demand by about 6.0 % (55 kWh), as shown in Figure 5, right.

According to the TRNSYS simulations the annual performance of the heat pipe system is, unlike the results of the DST model, slightly better ($\Delta f_{sol} \approx 0.5$ %-absolute) than that of the standard system. This difference can be explained by the extremely simplified system modelling of the DST procedure. For example, the heat capacity of the collector, which is significantly lower for the heat pipe prototype, is not considered in the DST performance prediction. Only the simulation in TRNSYS can realistically represent the comparison in that point. Apart from these explainable small differences, both approaches confirm with sufficient accuracy, that the considered heat pipe collector prototype exhibits the same annual system performance as the standard collector.

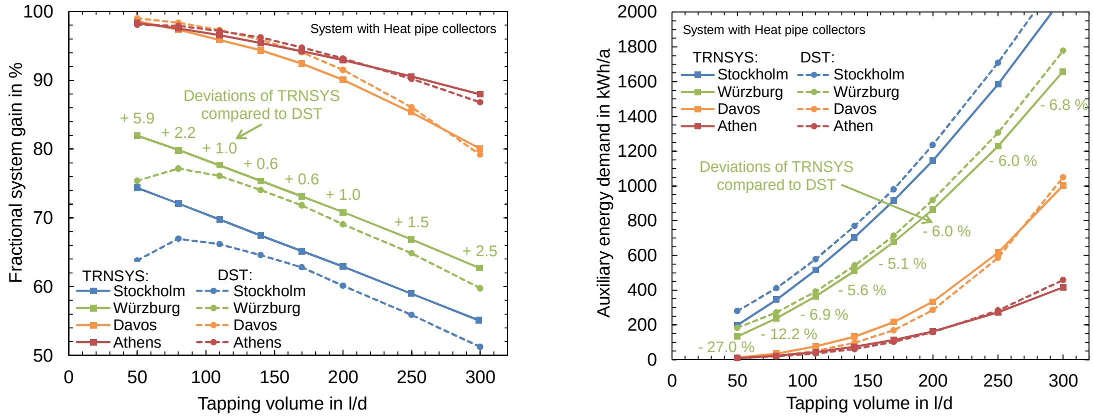


Fig. 5: Fractional system gain (left) and auxiliary energy demand (right) calculated with TRNSYS compared to the DST results against the daily tapping rate

3.4. Stagnation mode

In addition to the annual performance prediction, we investigated the stagnation behavior of both systems over several experimental test periods. During the tests the standstill of the solar circuit was realized by simultaneous switching-off the solar pump at solar noon and at an irradiance of about 1000 W/m². To evaluate the stagnation behavior we considered 30 stagnation days. The vapor formation has been investigated at different system pressure levels in the range between 1.0 / 1.8 and 2.0 / 2.8 bar system overpressure / absolute pressure at the collector.

In the case of the heat pipe collector we measured a maximum collector outlet temperature of 125 °C, which is up to 95 K lower than the maximum temperature recorded with the standard system. Figure 6 shows a typical stagnation sequence for both systems and the respective response of representative temperatures after deactivating the solar pumps at an overpressure level of 2 bars. For the standard system (red), a rapid increase takes place, so that the collector temperature significantly steps over the boiling temperature. The temperature at the inlet of the expansion vessel rises quickly to 115 °C. As a result of the vaporization, the system pressure increases in this case until the safety valve is triggered (> 6 bar). On the contrary, the temperature of the system with heat pipe collector (blue) achieves a maximum value of approx. 116 °C and the boiling temperature is not reached. The temperature at the expansion vessel is not affected by the stagnation sequence at all.

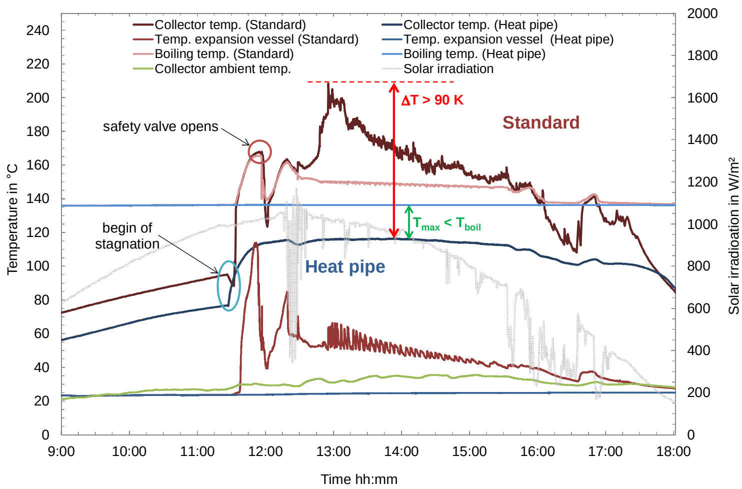


Fig. 6: Representative system temperatures over a stagnation sequence at an overpressure level of 2 bars

The boiling temperature of the solar fluid depends on the occurring system overpressure. For low system overpressures of only 1 bar, vapor formation takes place in the heat pipe collector, too. Figure 7 shows the maximum temperatures and pressures in both systems, as well as the resulting vapor formation in the solar circuit on the selected stagnation day. In the case of the standard system, the vapor at a temperature of about 155 °C is transported up to the solar station and the expansion vessel. In contrast to that, the maximum temperature at the heat pipe collector reaches 123 °C instead of 210 °C. The vapor transport is limited to few meters. The major part of the solar circuit shows a maximum temperature below 100 °C, which is significantly lower compared to the reference system.

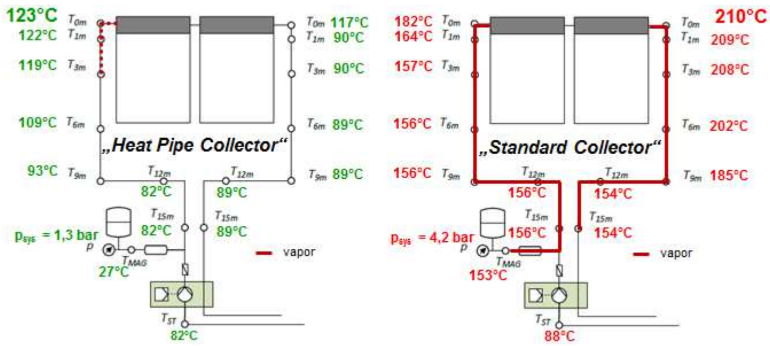


Fig. 7: Maximum temperature and pressure load in the solar circuit of heat pipe and reference system at a low system overpressure level of 1 bar

4. Parameter study with TRNSYS

4.1 System dimensioning

In addition to the annual performance both systems, as described in chapter 3, we investigate by means of TRNSYS simulations the influences of other boundary conditions, such as smaller designed collector fields. It was expected, that the differences of the annual performance between the systems could increase significantly with smaller collectors. Therefore the aperture area in the simulation was reduced whereby the storage tank volume was kept constant at 400 l. In general, the annual yield as well as the fractional system gain decreases and the auxiliary energy demand increases for both systems (see Figure 8 for Würzburg and for a tapping rate of 200 l/d). By reducing the aperture area to 4 m² (-33 %) the annual yields are correspondently reduced by about 20 % and the auxiliary energy demands rise by 40 %. If we focus on the differences between the collector types, the influence of the system performance is almost negligible. With refer to the annual yield, the relative deviation of the heat pipe system ranges from +0.6 % to -0.9 % by reducing the aperture area from 6 m² to 4 m². In consequence of this, the deviation in auxiliary energy demand varies from -1.2 % to +1.1 %. Contrary to our expectations the simulation results show that the annual performance of the heat pipe system is not significantly affected by smaller designed collectors compared to the standard system.

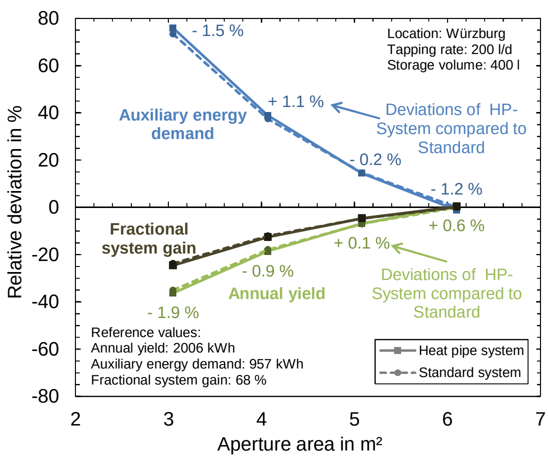


Fig. 8: Deviation of annual yield, fractional system gain and auxiliary energy demand for the heat pipe and standard system in dependency of the aperture area of the collector field

4.2 Shut-off behaviour and maximum temperature

The maximum fluid temperature of the heat pipe collector depends on several conditions. In particular, the type and amount of working fluid, which is inside the heat pipe, are the most important parameters. As a general rule, less fluid leads to lower maximum temperatures. A temperature shift by about ± 20 K can be simply managed by properly dosing the mass of a selected fluid in the heat pipe. Another important parameter is the slope of the collector power shut-off, which affects the kink point[1] (see Figure 9, left) and therefore the collector efficiency in the operating range. The slope of shut-off can be influenced i.e. by the type of working fluid, the heat losses and the internal thermal conductivity of the collector. Low heat transfer coefficients lead to flatter curves. By designing solar heat pipes for temperature limitation, both aspects have to be taken into consideration in order to achieve a significant reduction of the maximum temperature without compromising the annual system performance.

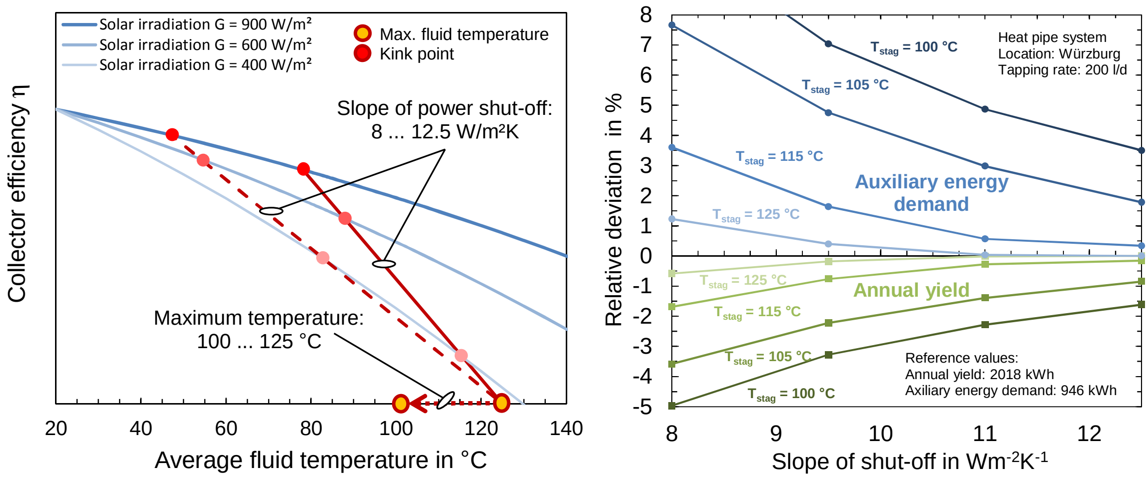


Fig. 9: Illustration of the varied parameters maximum temperature and slope of heat pipe power shut-off (left) and the annual yield and auxiliary energy demand as result of the parameter study (right)

The presented heat pipe prototype collector featured a maximum stagnation temperature of 125 °C and power limitation slope of 12.5 W/m²K. For a comprehensive evaluation, we further investigated the system performance for lower maximum temperatures and different slopes of power shut-off (see Figure 9, right). On the basis of our previous experience with different heat pipe prototypes, the maximum temperature was varied between 100 and 125 °C and the slope between 8 and 12.5 W/m²K. The parameter study was carried out with the system dimensions described in chapter 3 (collector area: 6 m², storage tank: 400 l). At a maximum temperature of 100 °C and a slope of 12.5 W/m²K the solar yield is reduced by 1.6 %, which leads to an increase in auxiliary energy demand by 3.5 %. If the stagnation temperature is limited to 105 °C, the solar yield is reduced by only 0.8 % and the auxiliary energy demand is increased by 1.8%. If lower values of the power limitation slope are considered, the effect is approximately negligible at T_{max} = 125 °C. In the case of T_{max} = 100 °C, significant losses are expected. A slope of 9.5 W/m²K, for example, leads to a lower annual yield by 3.5 % and a higher auxiliary energy demand by 7 %.

The parameter study shows, that a further reduction of the maximum temperature has a negative impact on the system performance. If a higher auxiliary energy demand by about 2 % is tolerated, the maximum temperature can be reduced to 115 °C with a minimum slope of 9.5 W/m²K or to 105 °C with a minimum slope of 12.5 W/m²K. If a temperature limitation to 115 °C is reached with a slope of 12.5 W/m²K, the demand of auxiliary energy increases by only 0.3 %. In consequence of this, vapor formation and the transportation of high temperatures in the solar circuit can be further reduced, in particular at a low system overpressure level, as shown in Figure 7.

[1] The kink point is not fixed at a certain temperature, but is affected by the slope of the shut-off and the occurring solar irradiation

5. Conclusion

We investigated a newly developed evacuated tube collector based on optimized heat pipes able to limit the temperature loads at the manifold to a maximum temperature of 125 °C. Additionally to collector efficiency measurements, we compared the annual yield of a heat pipe collector field in a domestic hot water system with an identical system with direct flow collector according to ISO 9459-5. The results show, that the heat pipe collector reaches almost the annual system performance of the reference collector. The system simulations in TRNSYS have confirmed the results of the DST procedure with a sufficient accuracy.

By means of extensive stagnation tests, we report a maximum temperature at the solar circuit of 125°C, which is 95 K lower compared to the reference system. As a main result, the vaporization can be completely avoided for a typical pressurized system and significantly reduced for low system overpressures. The use of these heat pipes can thus prevent the propagation of high temperatures in the system. The major part of the solar circuit shows a maximum temperature level below100 °C. Due to the significant reduction of thermal and pressure loads in the whole solar system the system design can be adapted and simplified. According to our parameter study with TRNSYS, a further reduction of the maximum temperature to 115 °C is possible without significantly affecting the annual performance.

As a result of the already achieved limitation of the maximum temperature to 125 °C and of the successful suppression of vapor formation, solar circuit components, such as the expansion vessel, the ballast vessel and the solar piping, can be resized or made by less expensive materials, i.e. by polymerics. According to our current estimation, the investment costs for the considered domestic hot water system can be reduced by about 8 – 10 %. Further simplifications in the installation procedure, i.e. by simpler filling and degasification of the solar circuit are possible. Finally, the lower system load can significant reduce the maintenance effort, in particular thanks to the prolonged life-time of the solar fluid. Taking into account the overall costs during the life-time of the system, the use of heat pipe collectors with temperature limitation can achieve a cost saving of 20 – 30 % compared to a state-of-the-art solution.

6. Acknowledgement

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7. References

Eisenmann W., Pujiula F., Köln H., Baumann A., 2004. On the Determination of the Effective Thermal Capacity of Solar Collectors, Proceedings 14. EuroSun Conference, Freiburg im Breisgau

Föste S., Schiebler B., Giovannetti F., Rockendorf G., Jack S., 2015. Butane heat pipes for stagnation temperature reduction of solar thermal collectors, Energy Procedia 91 p. 35-41

ISO 9459-5, 2007: System performance characterization by means of whole-system tests and computer simulation. Beuth Verlag, Berlin

Jack S., Rockendorf G., 2013. Wärmerohre in Sonnenkollektoren – Wärmetechnische Grundlagen und Bewertung sowie neue Ansätze für die Integration, final project report (published in german), FKZ 0325962A

Jack S., Parzefall J., Luttmann T., Giovannetti F., 2014. Flat plate aluminium heat pipe collector with inherently limited stagnation temperature. Energy Procedia 48 p. 105-113

Mientkewitz, G.; Zabel, J., 2010. Möglichkeiten eines Heatpipe-Kollektors ohne Stagnationsprobleme. Conference proceedings 20. OTTI Symposium Thermische Solarenergie, Bad Staffelstein

Reay D.A., Kew P.A., 2006. Heat Pipes: Theory, Design and Applications. 5th edition, Butterworth Heinemann, Oxford

Spirkl W., 1990. Dynamische Vermessung von Solaranlagen zur Warmwasserbereitung, VDI-Fortschrittberichte, Reihe 6: Energieerzeugung Nr. 241

Spirkl W., 1997. Dynamic System Testing Program Manual, InSitu Scientific Software

Streicher, W. et al., 2003. Report on Solar Combisystems Modelled in Task 26 (System Description, Modelling, Sensitivity, Optimisation), Report of IEA SHC – Task 26 Solar Combisystems

TÜV-Rheinland, 2014. Solar KEYMARK Certificate Licence Number 011-7S471 R. DIN CERTCO, Berlin

Visser H., 1997. Dynamic Testing of Active Solar Heating Systems, Final Report of the IEA SHC – Task 14 Dynamic Component and System Testing Subtask Software

8. Appendix

8.1 Specific system parameters

The specific system parameters of both systems are listed in Table 2. The collector coefficients as well as the stagnation temperature of the heat pipe collector were determined at the ISFH (see Figure 1). For the direct-flow collector, the parameters were taken from a referenced Solar Keymark certificate (TÜV-Rheinland 2014). The effective heat capacities of both collectors have been determined according to DIN EN 12975-2 by weighting the physical capacities of the single components. The significant difference is due to the substantially lower mass of solar fluid in the heat pipe collector.

Tab. 2: Specific system parameters used for performance simulation with both the heat pipe and the standard system

Parameters	Symbol	Unit	ETC with heat pipe	ETC Standard
Collector:				
Aperture area	A^*	m²	6.1	6.1
Conversion factor	η_0	-	0.754	0.774
Linear heat loss coefficient	a_1	$Wm^{-2}K^{-1}$	2.02/12.50[1]	1.94
Quadratic heat loss coefficient	a_2	$Wm^{-2}K^{-2}$	0.007	0.006
Maximum fluid temperature in stagnation	T_{stag}	°C	125	300
Effective collector capacity	C_{eff}	$kJm^{-2}K^{-1}$	4.75	10.19
Solar controller:				
On/off – criteria	$\Delta T_{on/off}$	K	7/3	7/3
Max. storage temperature	$T_{max,st.}$	°C	90	90
Max. collector temperature	$T_{max,coll.}$	°C	130	130
System:				
Storage volume	$V_{st.}$	l	400	400
Electrical auxiliary power	P_{aux}	kW	6	6
Heat loss rate storage	$U_{st.}$	WK^{-1}	2.69	2.69
Length of solar circuit piping (flow + return)	L_{sc}	m	30	30
Heat loss rate solar circuit	$U_{sc.}$	$Wm^{-2}K^{-2}$	3.19	3.19
Expansion vessel	V_{ev}	l	50	50
Ballast vessel	V_{bv}	l	12	12

[1] Slope of the power shut-off