EVALUATION METHODS FOR HEAT PIPES IN SOLAR THERMAL COLLECTORS – TEST EQUIPMENT AND FIRST RESULTS

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1. Introduction

Heat pipes in non- or low-concentrating vacuum tube collectors offer by the decoupling of absorber and solar circuit as compared to direct-flow collectors the advantage of a simpler hydraulic interconnection, while reducing heat loads in case of stagnation. However, it should be noted that heat pipes can only transfer a certain amount of thermal power and that the system of heat pipe and heat exchanger (manifold) represents an additional thermal resistance in the useful heat path of a solar collector. In a current research project[1] at the Institut für Solarenergieforschung Hameln (ISFH) the heat transfer characteristics of heat pipes applied in solar collectors are investigated. By experimental examination of heat pipes and manifolds with the aid of model calculations potentials for optimization in existing solutions are worked out. The focus of this paper comprises the presentation of newly developed test rigs and experimental results of the thermal behavior of commercially available heat pipes and manifolds in solar thermal collectors.

2. Heat pipes in solar thermal collectors

Heat pipes are characterized by a high heat transfer performance. In collectors they are used for heat transfer from absorber to manifold. Inside of heat pipes a heat driven two-phase thermodynamic cycle takes place. Therefore, in the evaporator section of the heat pipe, which is located at the absorber, the working fluid is evaporated and transported to the condenser section, which is located at the manifold. Here the condensation takes place. Driven by gravity the condensate flows back into the evaporator section where it evaporates again. Typically in solar thermal collectors cost-effective gravitational heat pipes without capillary structures (two-phase closed thermosyphons) are used. Within collectors heat pipes function as highly concentrating heat exchangers based on the area ratio of the evaporator to the condenser. This specific characteristic has influence on the thermal conductance as well as the heat transfer limitations of heat pipes. On the left side figure 1 represents an equivalent network of the mainly influencing thermal resistances on the overall thermal conductance of heat pipes. Furthermore, qualitative heat transfer limitations of gravitational heat pipes are shown on the right. There are several physical effects, which limit the maximum heat transfer rate of heat pipes. The most relevant for heat pipes in solar collectors are the entrainment limitation and the dry-out limitation (e.g. Faghri (1995)).

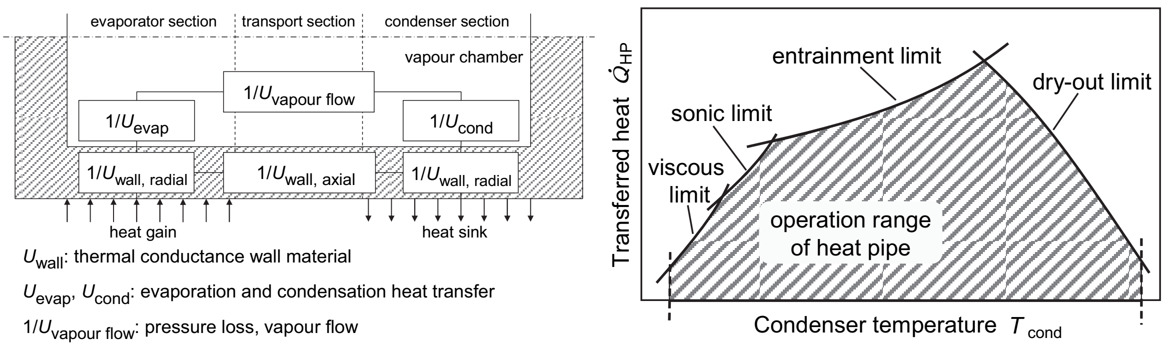


Figure 1: Equivalent resistance network of the main influences on the overall thermal conductance of heat pipes (left) and qualitative heat transfer limitations of gravitational heat pipes (right)

[1] The project "Wärmerohre in Sonnenkollektoren – Wärmetechnische Grundlagen und Bewertung sowie neue Ansätze für die Integration (HP-Opt)", FKZ 0325962A, in cooperation with the companies NARVA Lichtquellen GmbH and KBB Kollektorbau GmbH, is funded by the German Federal Ministry for the Environment, Nature Conservation and Nuclear Safety (BMU) based on a decision of the German Federal Parliament. The content of this publication is in the responsibility of the authors.

Within the useful heat path of collectors the heat pipe presents an additional thermal resistance. The thermal connection of the heat pipe condenser to the solar fluid (manifold) has to be taken into account, too. These additional thermal resistances decrease the overall efficiency of evacuated tubular collectors by typically one to four percentage points. Appropriate tests are therefore to be carried out to determine the heat transport characteristics of the heat pipe-manifold system and identify weak spots and potential improvements. In addition, the temperature-dependent heat transfer limitations of heat pipes are of interest. Due to heat transfer limitation at higher temperatures stagnation temperature reduction of the collector may result.

In the one-node collector model in figure 2 the useful heat path of heat pipe collectors is shown as a condensed equivalent network. Furthermore, the reduced efficiency of solar collectors with heat pipes compared to direct-flow collectors and the possible advantage of heat pipe use by reducing the stagnation temperature are shown qualitatively.

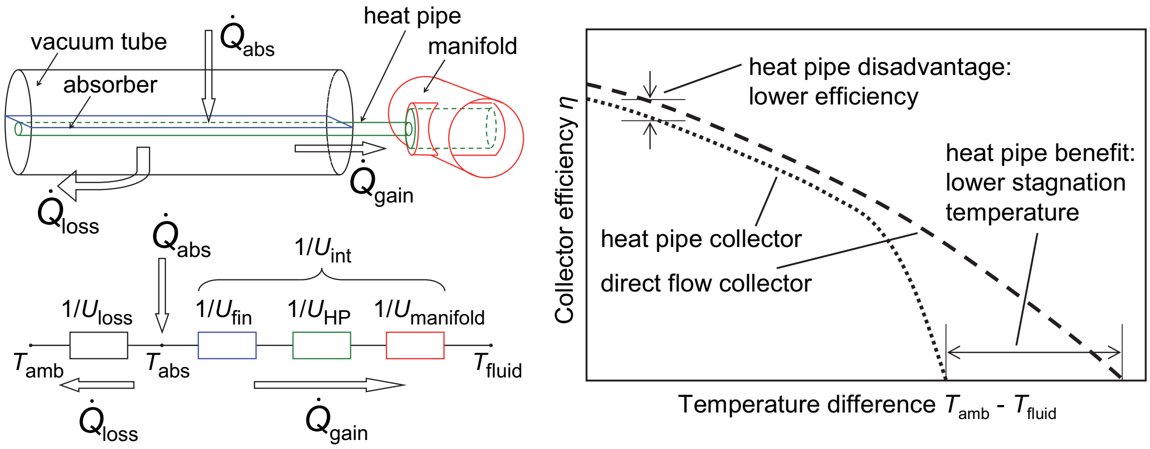


Figure 2: One-node collector model (left) and qualitative efficiency curves of collectors with heat pipes in comparison to direct-flow collectors (right)

3. Test rigs

For a detailed experimental investigation of the useful heat path of collectors with heat pipes, two test rigs were developed. One test rig has been built up for measurements on heat pipes and the second one is a test rig to study the thermal transport properties of manifolds. Both are presented in the following. In addition, test procedures for determining deterioration of heat pipes and manifolds have been designed. Regarding aging of manifolds especially the deterioration of thermal conductance pastes is relevant.

3.1 Heat pipe test rig

The main influencing operation variables in the heat transport capability of heat pipes are the amount of transferred heat $\dot{Q}_{HP}$, the condenser temperature T_{cond} and the inclination angle. In order to determine the effects of these parameters, the test rig is equipped with an electrical heat source, which is placed directly at the evaporator section of the heat pipe. A fluid circuit connected to the condenser section is used as a heat sink.

The test rig developed at the ISFH consists of the two main components fluid circuit and insulated test case with installed specimen (figure 3). The fluid circuit is a high-pressure water circuit, which can be operated at temperature levels up to 180 °C. To determine the useful heat output transported via the fluid, a Coriolis flow meter with a measuring range of 5 to 300 kg/h is used. Thus, even very small outputs down to 10 W are measurable. The test case is designed for heat pipe operating temperatures up to 400 °C.

Due to for this type of heat pipes specifically small condenser surfaces and resulting high heat flux densities, special attention was paid to the construction of the heat exchanger of the test rig (condenser to fluid). On the one hand, thermal conductance was optimized to a great extent through choice of material and flow management, while the required modularity for receiving various forms of condensers was preserved. A uniform heat transfer at the condenser surface was assessed by means of FEM simulation. To investigate heat pipes even at higher condenser temperatures than 180 °C, which is the maximum temperature of the fluid circuit, additional thermal resistances may be introduced as shown in figure 3. This way it is possible to increase the temperature of the heat pipe up to 400 °C (Schubert (2011)).

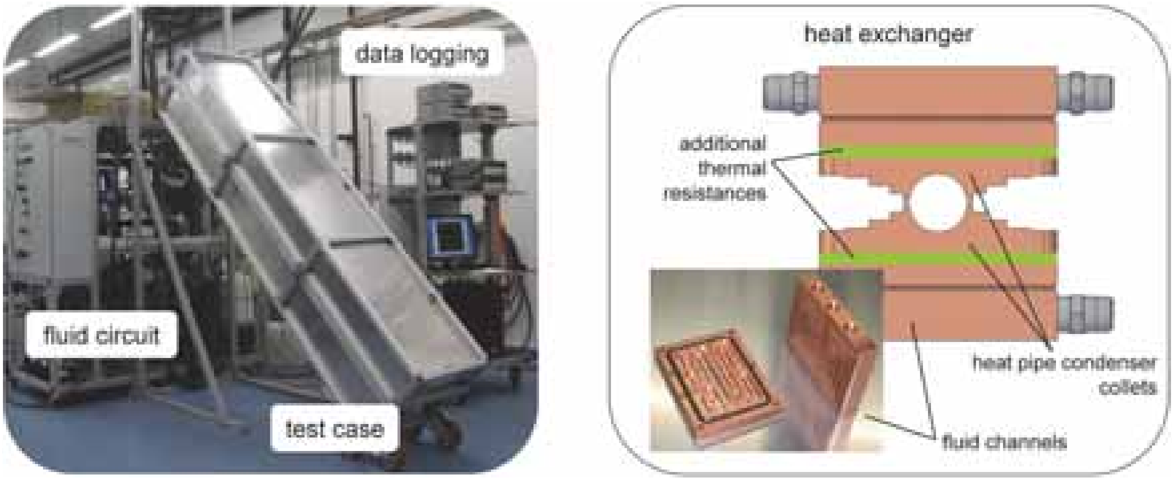


Figure 3: Complete heat pipe test rig (left) and heat exchanger between heat pipe condenser and fluid circuit with additional thermal resistances (right)

Using the test rig, the heat pipe's thermal conductance $[U_{HP}]$ = W/K as a function of the above mentioned factors is determined. The inclination angle, condenser temperature T_{cond} and evaporator temperature T_{evap} are set as boundary conditions. Within the standard measurement procedure temperatures are set (in particular on the evaporator) instead of heat loads, since by this type of boundary conditions steady states can be achieved much faster, which significantly shortens the testing time. The heat transfer is measured calorimetrically within the fluid circuit. It must be considered that the condenser side test case exhibits heat losses and thus the fluid heat output $\dot{Q}_{fluid}$ differs from the heat pipes heat transfer $\dot{Q}_{HP}$. To take account of this effect, the heat losses $\dot{Q}_{loss}$ and parasitic heat flows $\dot{Q}_{trans}$ of the test case were calibrated as a function of various operating parameters. Thus, the thermal conductance of heat pipes U_{HP} is determined using (see also figure 4)

$$U_{HP} = \frac{\dot{Q}_{HP}}{\Delta T_{HP}} = \frac{\dot{Q}_{fluid} + \dot{Q}_{loss} - \dot{Q}_{trans}}{\overline{T}_{evap} - \overline{T}_{cond}} \quad \text{with} \quad \begin{matrix} \dot{Q}_{loss} = f(U_{loss}) \\ \dot{Q}_{trans} = f(U_{trans}) \end{matrix} \quad \text{and} \quad \begin{matrix} U_{loss} = f\left(T_{amb}, \overline{T}_{fluid}, \overline{T}_{cond}\right) \\ U_{trans} = f(T_{amb}, \overline{T}_{fluid}, \overline{T}_{cond}) \end{matrix} \qquad \text{(eq. 1)}$$

By increasing the temperature difference between evaporator and condenser ΔT_{HP} the amount of transferred heat $\dot{Q}_{HP}$ increases. Thus, the heat transfer can be enlarged up to the heat transfer limitation of the heat pipe. Reaching a performance limit is typically characterized by the considerable increase in evaporator temperatures at the bottom of the heat pipe. For this reason, over the length of the evaporator several temperature sensors are distributed uniformly as shown in figure 4.

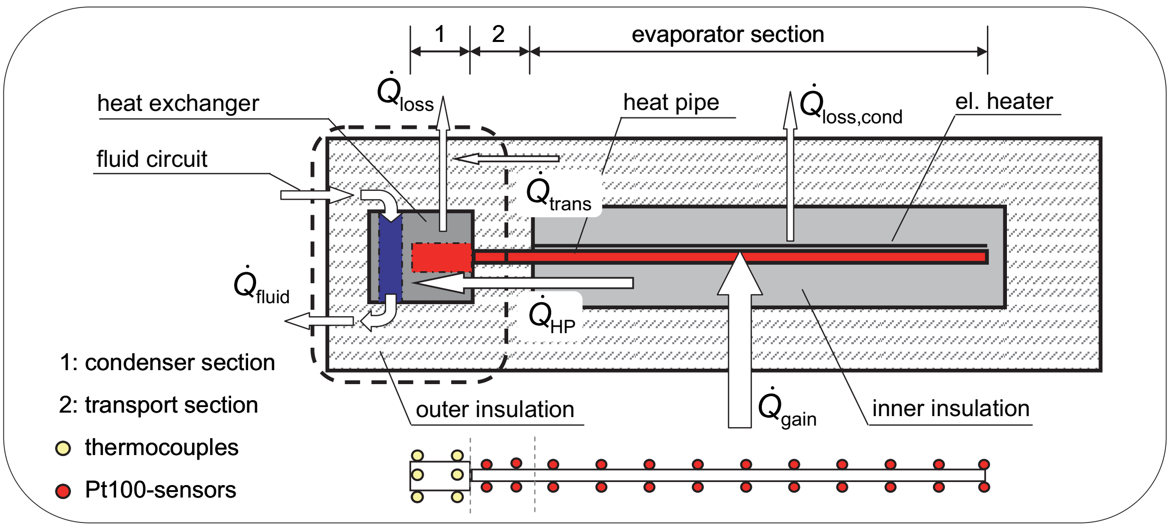
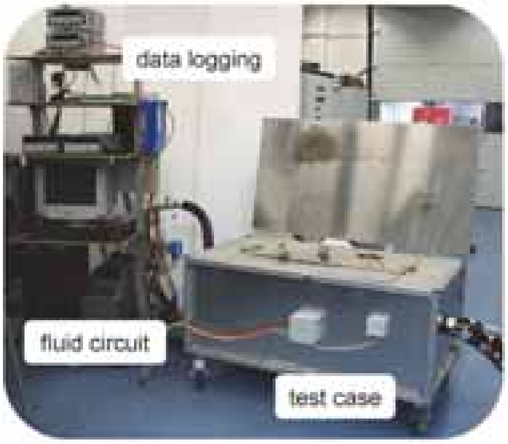


Figure 4: Schematically displayed heat flows inside the test case and positions of temperature sensors

3.2 Manifold test rig

Commercially distributed manifolds of heat pipe collectors are available in different designs. In principle they can be characterized by the quality of the thermal connection between heat pipes condenser surface to solar circuit working fluid. The manifold is defined as the heat exchanger, which is responsible for the temperature drop from the condenser surface to the mean fluid temperature of the solar fluid. Thus, the often used thermal conductance paste on the condenser surface is a partial aspect of manifold specimens. The test rig developed at the ISFH again consists of two main components, the fluid circuit and the test case with integrated specimen. As a heat source, an electric heater is used to replace the condenser of the heat pipe. The heat is discharged by the fluid cycle, which flows through the manifold. Figure 5 shows the test rig and presents the heat flows of concern as well as the temperature measuring points.



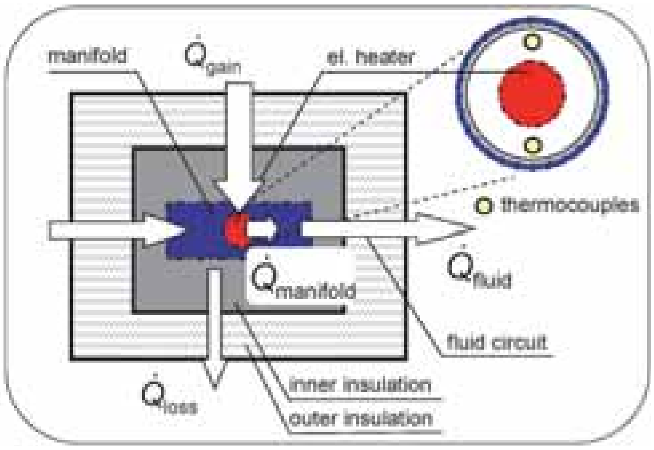


Figure 5: Complete manifold test rig (left) and schematically displayed heat flows inside the test case and positions of temperature sensors (right)

As boundary conditions mass flow rate $\dot{m}$, fluid inlet temperature T_{in} and electric power $\dot{Q}_{gain}$ or condenser temperature T_{cond} can be set. Taking into account the thermal losses of the test case the thermal conductance of the manifold $[U_{manifold}]$ = W/K leads to

$$U_{manifold} = \frac{\dot{Q}_{manifold}}{\Delta T_{manifold}} = \frac{\dot{Q}_{fluid} + \dot{Q}_{loss}}{\overline{T}_{cond} - \overline{T}_{fluid}} \text{ with } \dot{Q}_{loss} = f(U_{loss}) \text{ and } U_{loss} = f\left(T_{amb}, \overline{T}_{fluid}\right). \quad \text{(eq. 2)}$$

Thus, evaluations of the thermal conductance of manifolds regarding variation of mass flow, temperature, type of thermal conductance paste, etc. can be carried out. With this test rig it is also possible to examine the effect of deterioration of thermal conductance paste, as internal operating temperatures up to 400 °C are possible (without operating the fluid circuit pump).

4. Experimental results

Using the above mentioned test rigs commercially available heat pipes and manifolds from solar thermal collectors were examined regarding their heat transport properties. In the following sections exemplary results are presented and discussed.

4.1 Heat transfer limitations and thermal conductance of heat pipes

We examined commercially available heat pipes for solar collectors with different degrees of filling ratios, different working fluids and different geometries. For each specimen the heat transfer limitation and thermal conductance was determined at different condenser temperatures. Above condenser temperatures of 180 °C the thermal resistances in the heat exchanger of the test rig were applied.

Within the measurements it is possible to distinguish between the two relevant heat transfer limitations. The entrainment limit occurs, when the relative speed between the flow of steam and condensate, and thus the surface shear stress is so large, that the up flowing steam dams or even carries along the down running condensate. As a result, not enough condensate flows back into the evaporator and the end of the evaporator runs dry. The dry-out limit is just reached when the heat transfer is so high (or the filling ratio so low), that all the working fluid is involved in the heat pipe cycle and therefore the fluid pool at the bottom of the evaporator is depleted. This heat transfer limit is also characterized by a dry end of the evaporator, resulting in a high temperature.

By means of experiments, these two heat transfer limitations can be distinguished, because in contrast to hitting the dry-out limit the entrainment limit leads to stochastically pulsation of evaporator temperatures. This effect is clearly measurable and is based on the fact that the interaction between steam and condensate near the entrainment limit behaves unsteady. Time-varying flow conditions occur since the damming of the condensate can not be maintained quasi-stationary, thus resulting in significant temperature fluctuations as shown on the left side of figure 6. Further rising of the evaporator temperature leads to extended drying of the evaporator and therefore to rising temperatures at the end of the evaporator (see figure 6). Thus, the thermal conductance of the heat pipe is lowered, since the mean temperature difference between evaporator and condenser increases.

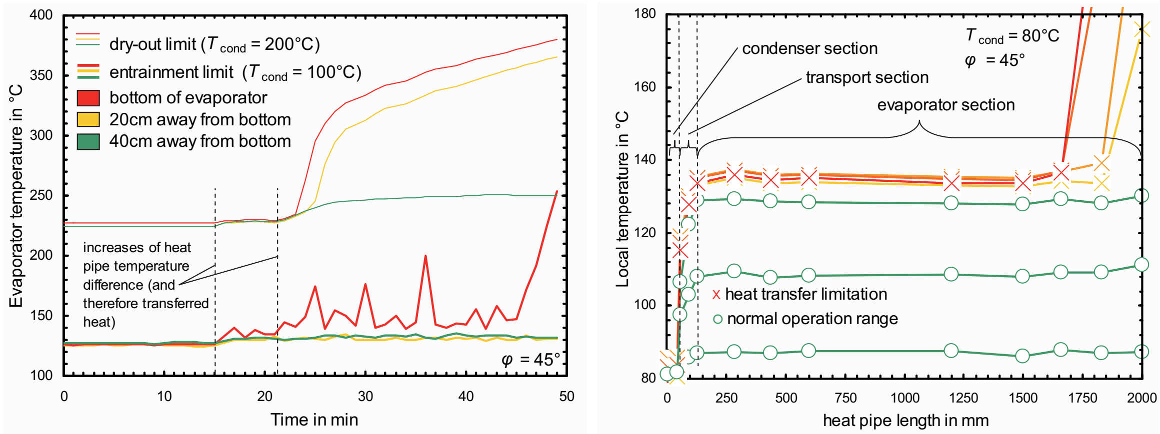


Figure 6: Time-dependent evaporator temperatures when hitting heat transfer limitations (left) and increasing evaporator bottom temperatures due to dry out of evaporator (right)

By varying the condenser temperature it is possible to determine the complete heat transfer limitations within the desired temperature range and thus the limit of the operating range of the heat pipe. Figure 7 shows the heat transfer limitations for three different commercially available heat pipes for solar thermal collectors. It is apparent that by usage of sample hp#1, the stagnation temperature of a solar thermal collector will be decreased since the heat pipe may only transfer heat up to a temperature level of 150 °C. The samples hp#2 and hp#3 possess a working range up to 285 °C and 260 °C, respectively. Main influencing factor on the entrainment limit is the inner diameter of the heat pipe between evaporator and condenser (position of maximum vapor velocity, e.g. Nguyen-Chi and Groll (1981), Bage (1989)) and the main factors influencing the dry-out limit are the filling ratio and the type of working fluid (e.g. Unk (1988)).

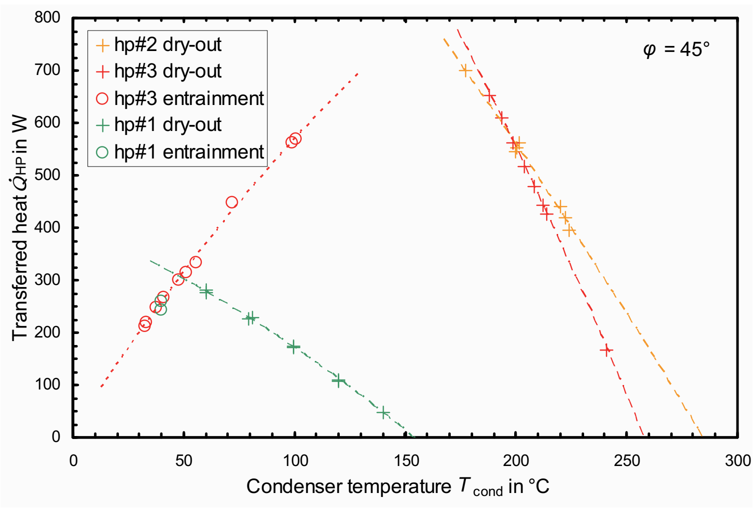


Figure 7: Experimentally evaluated entrainment and dry-out limitations of three different heat pipes

Within the normal operating range, the heat transfer is not affected by entrainment or dry out effects. Various heat pipes were investigated for their thermal conductance in that regard. The results presented were performed at a tilt angle of 45 °, where both the transferred heat (temperature difference between evaporator and condenser as test condition) and the temperature level of the heat pipe (condenser temperature as test condition) were varied. It turns out that the thermal conductance within the operating range of heat pipes is not constant. Figure 8 exemplifies the thermal conductance of two heat pipes as a function of condenser temperature and transferred heat.

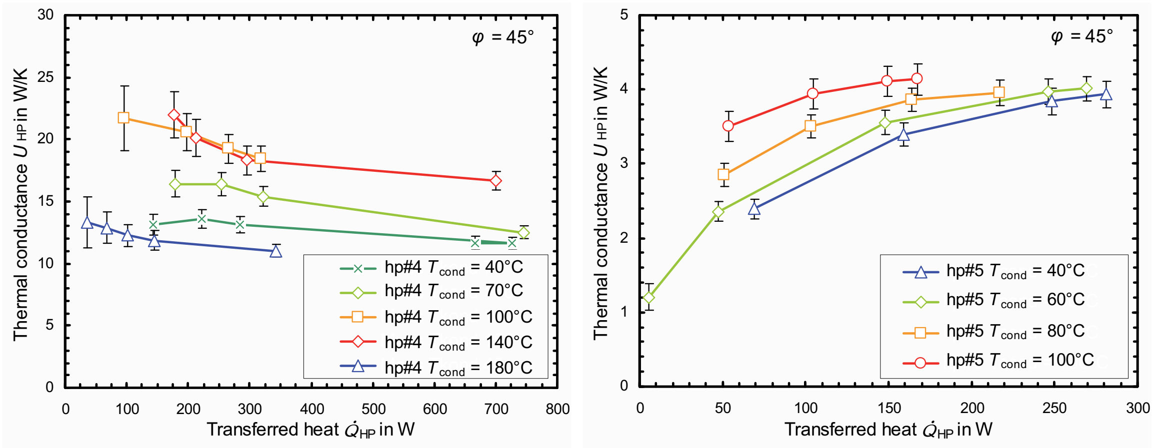


Figure 8: Thermal conductance of two heat pipe samples at different condenser temperature levels vs. transferred heat

It may be seen, that for sample hp#4 the thermal conductance slightly decreases with a larger heat transfer, whereas sample hp#5 behaves reverse. According to Nusselt's film theory the heat transfer coefficient of condensation decreases for higher heat transfers, as the film thickness increases and thus worsens the heat transfer from condensing surface to heat sink. This theoretical behavior does not apply to sample hp#5. The effect of sample hp#5 also occurred for another measured heat pipe specimen. After Hijikata et al. (1984) and Faghri (1995) this effect can be explained by the presence of non-condensable gases (inert gases).

Furthermore, it can be seen that the magnitudes of the thermal conductance of heat pipes differ between specimens. The small condenser surface, resulting in a high heat flux density, leads to the condensation heat transfer as the main factor influencing the overall thermal conductance. The condensation heat transfer is significantly influenced by the choice of working fluid. In particular, the thermal conductivity of the working fluid is a relevant variable, as parameter studies conducted at ISFH show. It should be noted that many publications are available regarding condensation heat transfer coefficients in heat pipes among which the work of Gross (1991) may be emphasized.

4.2 Thermal conductance of solar collectors manifold

As described in chapter 2 next to the thermal conductance of the heat pipe, the thermal conductance of the manifold is of high relevance, especially as the manifold is connected in series with the heat pipe in the collector's useful heat path. We examined several commercially available manifolds all in form of a dry connection. This means that the condenser of the heat pipe can be removed while the solar circuit remains closed and does not have to be emptied. Thus, the heat pipe's condenser surface is not in direct contact to the fluid, and thermal conductance paste is used inside the gap between condenser and manifold to improve the contact. For each manifold specimen a variation of the mass flow rate has been performed. The results are shown in figure 9. In addition, the thermal conductance of the heat exchanger, which has been developed for the heat pipe test rig (fig. 3), has been investigated, too.

The thermal conductance values of the manifolds at an exemplary mass flow rate of 100 kg/h has been found to be in the range from 2 to 10 W/K. Main influencing factor is the thermal connection of the condenser surface to the manifold. Manifolds, which use mechanical force to press themselves onto the condenser surface, e.g. by a clamping connection, have a far better thermal conductance than simple plug-in versions. Use of thermal conductance paste states a significant factor even for clamped manifolds. The right side of figure 9 shows the magnitude between the use of no thermal conductance paste, standard thermal conductance paste and liquid metal thermal conductance paste at a clamped manifold. Studies on the effect of thermal conductance paste for plug-in manifolds, and degradation studies are currently underway.

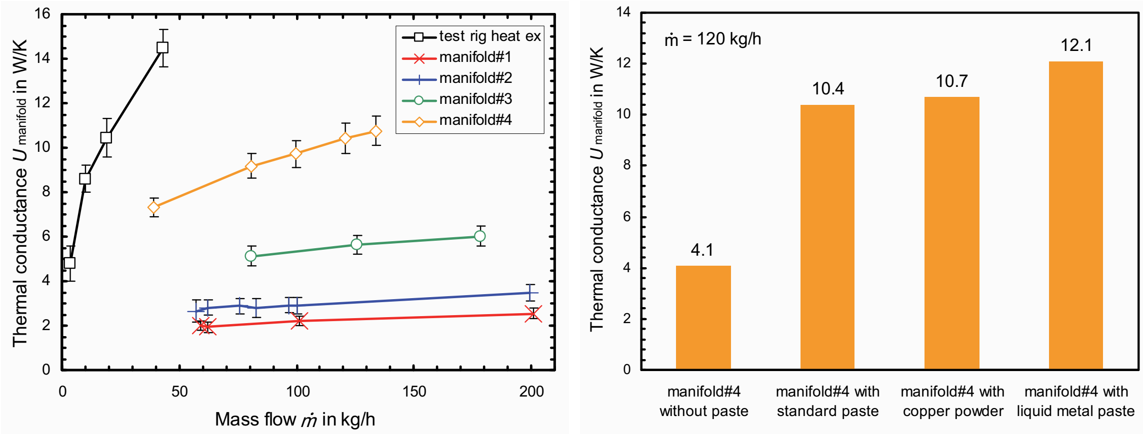


Figure 9: Thermal conductance characteristics of different manifold samples vs. mass flow rate (left) and influence of conductance paste on the thermal conductance value (right)

5. Internal conductance of useful heat path

In principle it has been shown, that with regard to the overall heat transfer capability both the heat pipe itself and the manifold represent the essential resistances. Since the thermal conductance values of both components are quiet similar in magnitude, both need to be regarded with attention. Taking into account the thermal conductance of the absorber fin, which can easily be calculated by means of the fin factor (Duffie and Beckman (2006)), the internal conductance of the collectors useful heat path may be estimated. To achieve comparability only single glass vacuum tube collector's with glass-metal-bond and flat fins were considered. Figure 10 shows three typical internal conductance values of heat pipe collectors. Additionally, the shares of the single conductance portions are presented.

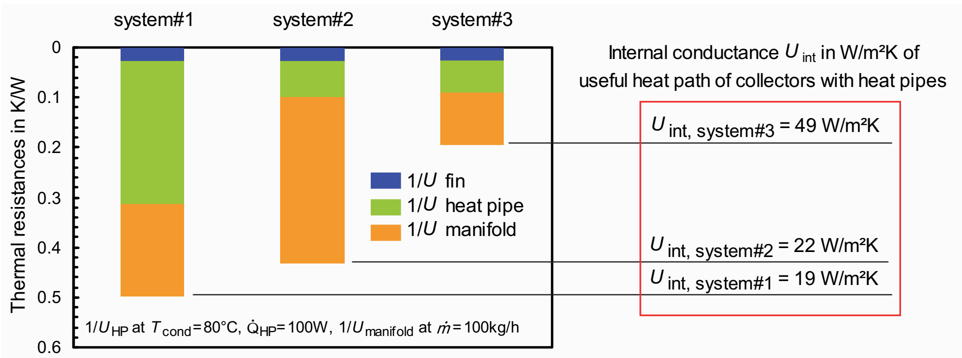


Figure 10: Internal conductance values of vacuum tube collectors with heat pipes

It turns out that the internal conductance of vacuum tube collectors with heat pipes lies between 20 and 50 W/m²K. The fin has only a minor influence, and depending on the specimen the heat pipe or the manifold present the largest thermal resistance in the useful heat path. Optimization of heat pipe collector's useful heat paths should therefore consider the manifold as well as the heat pipe. The impact of optimization on the overall efficiency of the collector can be estimated as follows. Taking into account an average transmittance-absorptance product of 0.8 and a typical loss coefficient of 1.3 W/m²K, the collector's conversion factor η_0 can be estimated using equation 3. Assuming an internal conductance U_{int} of direct-flow vacuum tube collectors of 80 W/m²K, the heat pipe collectors can roughly be compared to the direct-flow design as shown in table 1.

$$\eta_0 = (\tau\alpha)_{eff} \cdot F' = (\tau\alpha)_{eff} \cdot \frac{U_{int}}{U_{int} + U_{loss}} \quad \text{(eq. 3)}$$

Table 1: Comparison of conversion factors of vacuum tube collectors with heat pipes and direct-flow vacuum tube collectors

Collector	Internal conductance U_{int} in W/m²K	Transmittance-absorptance product $(\tau\alpha)_{eff}$	Loss coefficient U_{loss} in W/m²K	conversion factor η_0
Collector system #1	19.2	0.8	1.3	**0.749**
Collector system #2	22.1	0.8	1.3	**0.756**
Collector system #3	49.4	0.8	1.3	**0.779**
Direct-flow collector	80.0	0.8	1.3	**0.787**

Vacuum tube collectors with heat pipes exhibit due to the additional thermal resistances in the useful heat path by about one to four percentage points lower conversion factors than direct-flow collectors. Main variables and thus starting points for optimizing are the surface area of the condenser, the choice of working fluid in the heat pipe and the thermal connection of the condenser to the manifold.

Furthermore, it was shown in chapter 4.1, that the thermal conductance of heat pipes due to different effects is not constant for different operating conditions. Lower heat pipes thermal conductance at lower heat transfer rates e. g. because of non-condensable gases results in a non-constant internal conductance of the collector. This leads to a reduction in efficiency at lower irradiances (see eq. 3). In figure 11 the effect due to the non-constant thermal conductance of heat pipes on the conversion factor is shown.

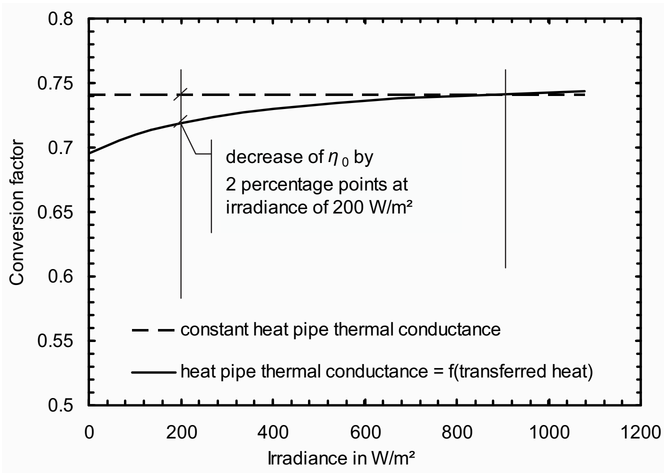


Figure 11: Influence of not constant thermal conductance of heat pipes on conversion factor vs. irradiance level

At an irradiance level of 200 W/m² the conversion factor is lowered by two percentage points for the considered setup in comparison to a high irradiance. This effect will not be detected in standard tests according to e. g. EN 12975, because the test uses high irradiance setups only. However, it should be noted that the conversion factor is not strongly lowered and that the collector yield at lower irradiance is less important for the annual collector yield. On the other hand it has to be mentioned, that collectors with fully functioning heat pipes have a benefit at low irradiance levels, which however is hardly to be measured in a system. Therefore, a check on non-condensable gases is recommended.

6. Conclusion and perspective

Vacuum tube collectors with heat pipes are common; their market share is growing and the benefits due to the thermal decoupling of the absorber surface from the solar circuit are obvious. However, the conversion factor of collectors with heat pipes is by one to four percentage points lower if compared to direct-flow collectors.

For the experimental investigation of the heat transfer characteristics of heat pipes and manifolds specific test rigs have been developed. An experimental benchmarking was carried out on commercially available heat pipes and manifolds, of which some of the results were presented in this paper. It has been shown that the addition of heat pipes and heat exchanger manifolds has a significant influence on the useful heat path of vacuum tube collectors. By means of experimental investigations and first theoretical parameter studies main influencing factors and thus potentials for improvement were identified. These are the thermal conductivity of heat pipe working fluid, the size of heat pipe condenser area and its connection to the manifold. This originates from the fact that in the area of condenser and manifold the highest heat flux density exists.

Further elaboration of a theoretical model of heat pipes developed at ISFH will allow more detailed sensitivity analyses and also provide more knowledge of operating conditions of the heat pipe. By means of a new filling device experimental parameter studies on own prototypes can be carried out and e. g. the influence of non-condensable gases on heat pipe performance may be investigated.

It could be shown that the different heat transfer limitations in gravitational heat pipes can be measured and distinguished with the test rig. With both common and newly developed theoretical models of the heat transfer limitations parametric studies will be carried out to elaborate optimal filling quantities and types or combinations of working fluids. The goal is to adapt the limitations and accordingly the operation range of the heat pipe for lowering stagnation temperatures in the secondary fluid circuit and thus provide a contribution to the stagnation security of solar thermal systems.

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