



Investigation of two-phase thermosyphon performance filled with modern HFC refrigerants

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Abstract

Two-phase closed thermosyphons (TPCTs) are widely utilized as heat exchanger elements in waste heat recovery systems and as passive heating/cooling devices. They are popular because of their high thermal conductivity, simple construction and reliability. Previous researches indicate that refrigerants are performing better than typical TPCT working fluids like deionized water or alcohols in the low temperature range. In the present study three HFC (Hydrofluorocarbons) refrigerants were tested: R134a, R404A and R407C. The total length of the investigated TPCT is 550 mm with equal length (245 mm) condenser and evaporator sections. Its outer diameter is 22 mm with 1 mm wall thickness. The evaporator section was heated by hot water with varying inlet temperature by 5 K step in the range of 288 K – 323 K. The condenser was cooled by cold water with inlet temperature kept at a constant value of 283 K. It was found that using R134a and R404A as working fluids heat transfer rates are the highest. For both refrigerants 10% is optimal filling ratio. They can be utilized interchangeably because the differences between their throughputs are within uncertainty bands. R407C performance was 50% lower. Other disadvantages of using this refrigerant are relatively high working pressures and higher optimal filling ratio (30%).

1 Introduction

TPCTs also known as wickless heat pipes are high thermal conductivity devices. Their construction is very simple – they are tightly sealed closed metal pipes partially filled with working fluid. TPCTs are utilized as elements of recuperative heat exchangers, cooling and temperature stabilization systems. Recuperative applications are characterized by a low temperature difference required for heat flow through TPCTs. In the design process of waste heat recovery systems, an essential aspect is a choice of working fluid and volumetric filling ratio. Pipatpaiboon et al. [1] obtained experimentally thermosyphon heat exchanger throughputs for three working fluids: distilled water, methanol and R134a. Best heat recovery efficiencies and highest heat transfer rates were reached with R134a (working temperature: 100 °C). Recently MacGregor et al. [2] tested the possibility of R134 a replacement with 5% ethyleneglycol - water solution. For low temperature difference range R134 a exhibited higher thermal effectiveness than

the glycol solution. Ong and Haider-E-Alalhi [3] compared the performance of a vertical TPCT for various filling ratios with different working fluids like R22, R134a and water. Water had the ability to transfer the highest throughput but was less efficient at low temperature differences. Jouhara and Robinson [4] investigated the application of dielectric Fluorinert™ liquids as wickless heat pipes' working fluids. They had an advantage over water, which was the reference substance, only for low heat transfer rates (<40 W). Danielewicz et al. [5] and Cieśliński and Fiuk [6] used methanol as working fluid in their prototype thermosyphon heat exchangers. Alcohols like methanol and ethanol, however are characterized by lower thermal effectiveness than water, which can be seen also in Negishi and Sawada [7], where ethanol and water overall heat transfer coefficients were directly compared for the same TPCT operating temperatures. Studies presented above indicate better performance of modern refrigerants (i. e. R134a), in a lower temperature difference range than water. Alcohols (methanol) and dielectric fluids also appear to be inferior to refrigerants, although there were no experimental studies that would compare their performance directly. Researches concerning modern refrigerants are usually narrowed down to popular R134a refrigerant [8, 9]. Definitely less works is dedicated to zeotropic blends refrigerants, i. e. R404A, R410A or R407C. Fadhl et al. [10]

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conducted CFD simulation of thermosyphon filled with R134a and R404A refrigerants. Experimental heat transfer rates obtained for the validation of numerical results indicate better performance of R404A compared to R134a. Yau and Foo [11] tested R134a, R22 and R410A refrigerants as revolving heat pipe working fluids. They have chosen R134a as most effective, mainly because the lower ODP (Ozone Depletion Potential) and GWP (Global Warming Potential) coefficients [12] than R22 and lower operation pressures than R410A. Considering the different working principle and flow pattern inside a revolving heat pipe, results of this study cannot be translated directly to TPCTs. Some of the researches deal with natural refrigerants, like Solomon et al. [13]. They investigated an anodized TPCT filled with a blend of propane (R290) and isobutane (R600a). More recent works are concentrated on using mixtures as TPCT working fluids, like in [14] where blend of fusel oil and deionized water was tested - it showed approximately 30% thermal resistance decrease relatively to pure water. Number of studies is also devoted to utilization of the nanoparticles as additives to working fluid. For example in [15] a performance of a TPCT filled with water with small additive of silver nano particles and surfactant was examined or in article [16] where carbon nanotubes/water nanofluid oxidized with acids was investigated. In both of the cited papers some significant improvements of thermal performance were shown. Despite of the recent trends in TPCT studies, there are few studies concerning modern refrigerants as working fluids of wickless heat pipe, and their performance is not compared directly. The present study provides such comparison in the temperature range characteristic for air conditioning recuperation systems [17, 18], heat recovery from wastewater in sewers [19] or ground heat exchangers [20].

1.1 Brief characterization of tested refrigerants

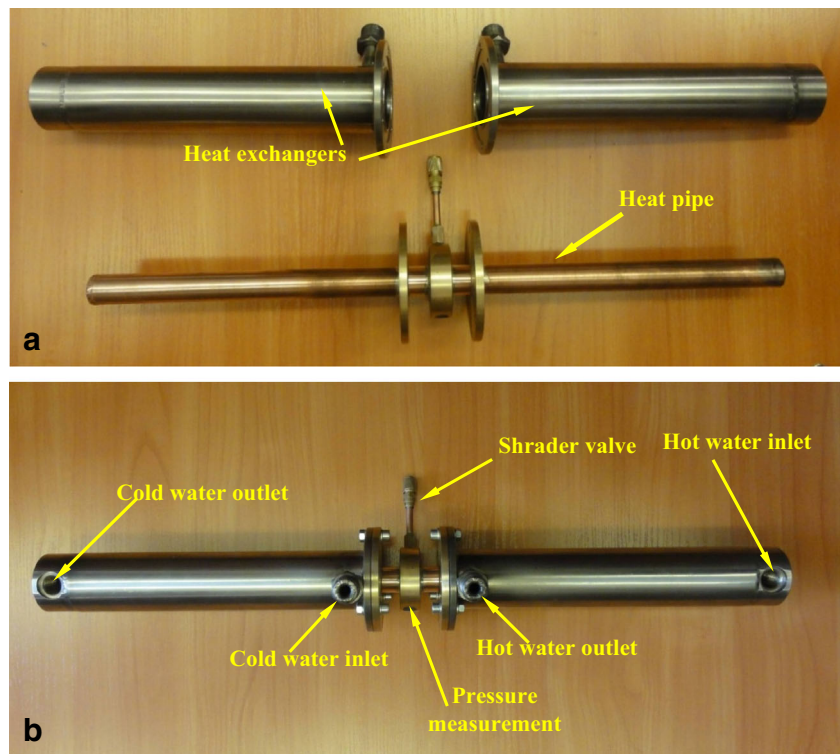
Refrigerants tested in present paper are from the HFC (hydrofluorocarbons) group, characterized by zero ozone depletion potential and medium to high global warming potential. They are widely used in refrigeration and air conditioning systems. R134a is one-component fluid replacing R-12. It is utilized for medium and high temperature refrigeration, automotive and residential air conditioning. R404A is a blend of HFC refrigerants used for medium and high temperature refrigeration. Its composition comprises: R125 (44%), R143a (52%) and R134a (4%) [21]. It is a zeotropic refrigerant, but its temperature glide is not high (0.4 K – 0.6 K). R407C is also a blend of refrigerants designed to replace R22 in air conditioning applications. Its composition is R32 (23%), R125 (25%) and R134a (52%) [21]. It is strongly zeotropic, with approximately 5–6 K average difference between bubble and dew temperature.

2 Experimental apparatus and procedure

The wickless heat pipe under investigation is 550 mm long (total length), 22 mm outer diameter and 20 mm inner diameter. To ensure excellent thermal conductivity and chemical compatibility with refrigerants the heat pipe container is made from copper. The heat pipe is enclosed by two stainless steel heat exchangers on the two opposite ends. Actual photo of heat pipe with heat exchangers before and after assembly can be seen in Fig. 1. Water flows through annular spaces between heat pipe and heat exchangers walls. Such assembly was done to heat the evaporator section and cool the condenser section. The evaporator section heat exchanger was fed by a hot water stream in range of temperatures from 288 to 333 K (15–50 °C) increased gradually during the experiment by steps of 5 K. The condenser section was fed by cold water at approximately constant temperature of 283 K (10 °C). Measured parameters were temperatures on the outer surface of heat pipe container, inlet/outlet water temperatures and the pressure inside TPCT. Temperatures are measured by T type thermocouples (Copper-Constantan). They were calibrated according to Bentley [22]. The thermocouple tips were brazed carefully to outer surface of the TPCT wall to ensure good connection without additional heat transfer resistances. The temperature measurement of inlet and outlet water streams was done by insertion of thermocouple tips halfway inside the water feeding tubes. Positions of the measurement points are shown in Fig. 2. Three points are on the evaporator section (T_1 – T_3), two on adiabatic (thermally insulated: T_4 – T_5) and three on condenser section (T_6 – T_8). The inside pressure was measured by an electronic, linear pressure transducer installed midway the length of the heat pipe (Fig. 1b). The arrangement of the experimental rig is shown in Fig. 3. The cold junctions of thermocouples are submerged in a distilled ice water pot (C). There are no extension leads between the hot and cold junction to ensure good repeatability and precision of measurement. The thermoelectric signal is transferred from the cold junction through copper wires to a strip chart temperature recorder (D) and bench-type digital multimeter (E). Raw voltages were logged from multimeter's output (emf – temperature conversion software was not used).

The procedure of measurement was carried out in following steps. First, the TPCT (A) was evacuated by a two-stage vacuum pump to the level of 15–25 μm (2–3,33 Pa) and charged with the refrigerant by a high precision filling system. Next, constant temperature water baths (J, K) were switched on to attain stable temperatures. After that cold and hot water was pumped by circulation pumps (L) to the heat pipe heat exchangers through thermally insulated pipes. Water flow rates were kept constant by control valves (F) at 15 l/h ($4,17 \times 10^{-6} \text{ m}^3/\text{s}$). There are two rotameters connected in parallel: “low flow” up to 25 l/h and “medium” which measurement range is 250 l/h (I). The rotameters were calibrated carefully before every measurement session. The volumetric flow was also measured by paddle-wheel sensors

Fig. 1 Pictures of heat pipe with heat exchangers: a) before assembly b) after assembly



with impulse counter meters (G). Steady state was reached after a relatively long time (from 0.5 to 2 h) because of the low volumetric flow. It was indicated by a constant value of temperatures over time, which can be observed from the strip chart temperature recorder (D). Another indication is the lack of pressure variation inside the heat pipe, read from the pressure meter (B). Both constant temperature water baths could be cooled by ice water from small capacity chiller (H) or tap water. The schematic depiction of the test rig is simplified for clarity, as the complicated system of cut-off valves and tubes would obscure the principle of the test rig operation.

3 Data reduction

Both heat exchangers mounted on the TPCT are balanced with the use of an equation originating from the first law of thermodynamics for open systems:

$$\dot{Q} = \dot{m}c_p(T_o - T_i), \tag{1}$$

where c_p – specific heat capacity and mass flow rate:

$$\dot{m} = \dot{V}\rho(T_i) \tag{2}$$

In above equations \dot{V} is volumetric flow rate, T_o – mean fluid temperature at outlet, T_i – mean fluid temperature at inlet. Density: $\rho(T_i)$ is a function of the inlet temperature.

Efficiency of the TPCT is calculated by:

$$\eta = \frac{\dot{Q}_c}{\dot{Q}_h}, \tag{3}$$

where \dot{Q}_h is the heat transfer rate, which the evaporator section absorbs from the hot water and \dot{Q}_c is the heat transfer rate transferred from the condenser section to cold water. The volumetric filling ratio (FR) of the wickless heat pipe is calculated with reference to the total volume:

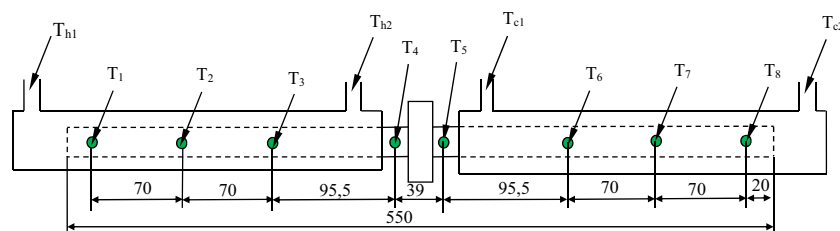


Fig. 2 Schematic view of heat pipe with temperature measurement points with marked positions: T_{h1} – temperature of inlet hot water, T_{h2} – temperature of outlet hot water, T_{c1} – temperature of inlet cold water, T_{c2} – temperature of outlet cold water, $T_1 \div T_8$ – temperatures outer surface of the heat pipe wall

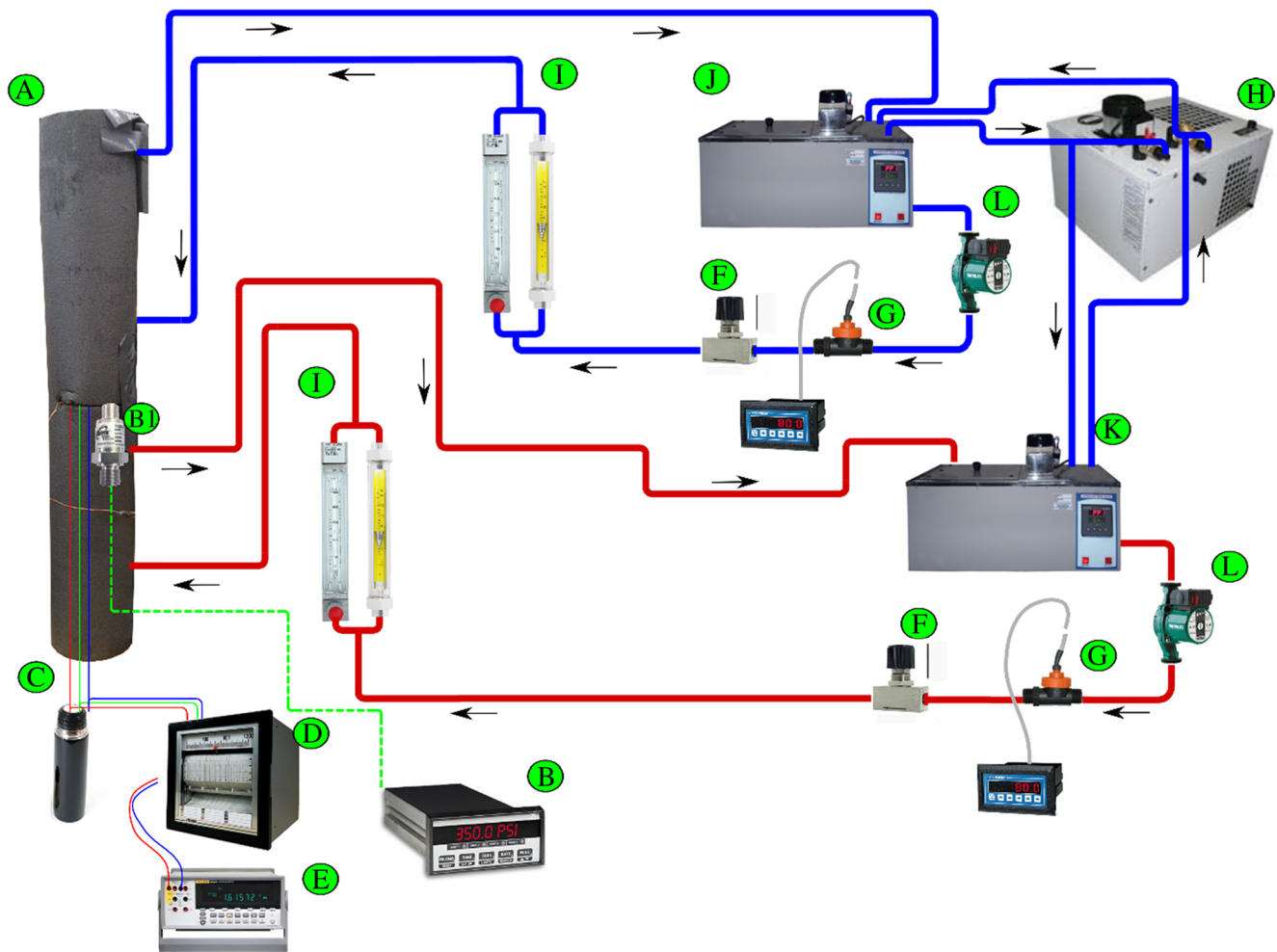


Fig. 3 Schematic of test rig: A) insulated TPCT, B) pressure meter, B1) pressure sensor, C) ice pot (Dewar flask with ice and water), D) strip chart temperature recorder, E) digital multimeter, F) control valve, G) low flow

paddle-wheel sensor with meter, H) low capacity chiller, I) two rotameters in parallel, J) constant temperature cold water bath, K) constant temperature hot water bath, L) circulation pump

$$FR = \frac{V_p}{V}, \quad (4)$$

where V_p is volume of liquid working fluid (liquid pool) and V is total volume of TPCT.

Thermal resistance of the evaporator section is given by:

$$R_e = \frac{T_e - T_{sat}}{\dot{Q}}, \quad (5)$$

where T_{sat} – saturation temperature [K], $T_e = (T_1 + T_2 + T_3)/3$ – mean temperature of the evaporator section, where the temperatures are measured at positions shown in Fig. 2.

3.1 Uncertainty analysis

The thermocouple calibration process showed that uncertainty of the temperature measurement is 0.25 °C for the range utilized during the experiment. Heat transfer rates are obtained from eq. (1) which involves the measurement of the

volumetric flow rate with 1 l/h uncertainty. During calibration of the flow rate and temperature, a greater deviation from fitting curve than 0.25 °C and 1 l/h was not recorded. This leads to the conclusion that, in spite of lack of knowledge of the probability distribution, if no data is found outside the stated ranges, the uncertainty interval has a level of confidence as high as 95%. Every quantity is measured independently (no correlation). Assuming that the uncertainty of c_p value is negligible, the combined uncertainty (u_c) of \dot{Q} can be obtained from a first-order Taylor series approximation (law of propagation of uncertainty):

$$u_c^2(\dot{Q}) = \left(\frac{\partial \dot{Q}}{\partial \dot{m}}\right)^2 u^2(\dot{m}) + \left(\frac{\partial \dot{Q}}{\partial T_i}\right)^2 u^2(T_i) + \left(\frac{\partial \dot{Q}}{\partial T_o}\right)^2 u^2(T_o) \quad (6)$$

Where u – uncertainty of a measurement of quantity. After applying eq. (1) and (2) to eq. (6):

$$u_c^2(\dot{Q}) = (c_p(T_o - T_i))^2 u^2(\dot{m}) + (\dot{m}c_p)^2 u^2(T_i) + (\dot{m}c_p)^2 u^2(T_o) \quad (7)$$

Uncertainty of thermal efficiency:

$$u_c^2(\eta) = \left(\frac{1}{\dot{Q}_h}\right)^2 u^2(\dot{Q}_c) + \left(\frac{-\dot{Q}_c}{\dot{Q}_h^2}\right)^2 u^2(\dot{Q}_h) \quad (8)$$

Typical uncertainty values are given in Table 1:

It can be seen that uncertainties vary relatively little through the range of heat transfer rates (4.5–109.4 W) utilized in the experiment (6.3–9.4 W). Because the initial heat transfer rates are low, uncertainties can exceed the lowest measured heat transfer rates. Relative uncertainties decrease gradually, as heat throughput increases, to about 10% for an approximate level of 100 W. Combined uncertainties are showed graphically in plots in the form of error bars. The expanded uncertainty of FR is estimated at $\pm 5\%$. Relative uncertainty of the absolute pressure measurement is 0.5%.

4 Experimental results and discussion

The purpose of this work is to test the potential of the utilization of modern refrigerants as vertical TPCT working fluids. Parameters obtained by experimentation are heat transfer rates at evaporator and condenser sections, thermal efficiency and absolute internal pressure. The power throughput of the TPCT for identical heating and cooling conditions is the most straightforward parameter for working fluid evaluation. Efficiency is a measure of heat losses for the considered wickless heat pipe construction. Internal pressures are important for calculation of the wall thickness of the container. This data gives enough knowledge for the selection of the most effective working fluid. Experimental values are represented by markers. Curves connecting them are pure interpolation,

showing the approximate trend of experimental results (increasing, decreasing). Heat transfer rates absorbed by the evaporator section against hot water inlet temperatures (range from 15 °C -50 °C) for working fluid R134a are showed in Fig. 4. Refrigerant R134a exhibits the greatest differences in throughput for varying volumetric FR. There is an obvious advantage of 10% filling ratio. 40% FR seems to be the worst as heat transfer rates are very small for low temperature differences between the evaporator and condenser. Generally 20%, 30%, and 40% filling ratios are characterized by virtually identical heat transfer rates for most of the temperature range. Differences between values of throughputs for these filling ratios are relatively low - they often stay within uncertainties bands. Some heat transfer rate values are not plotted, for example for 40% FR in the range of 288–293 K hot water inlet temperature. They were excluded because of heat pipe was not working (approximately zero heat transfer rates). Figure 5 shows heat transfer rates for varying FRs and inlet hot water temperatures at the condenser section.

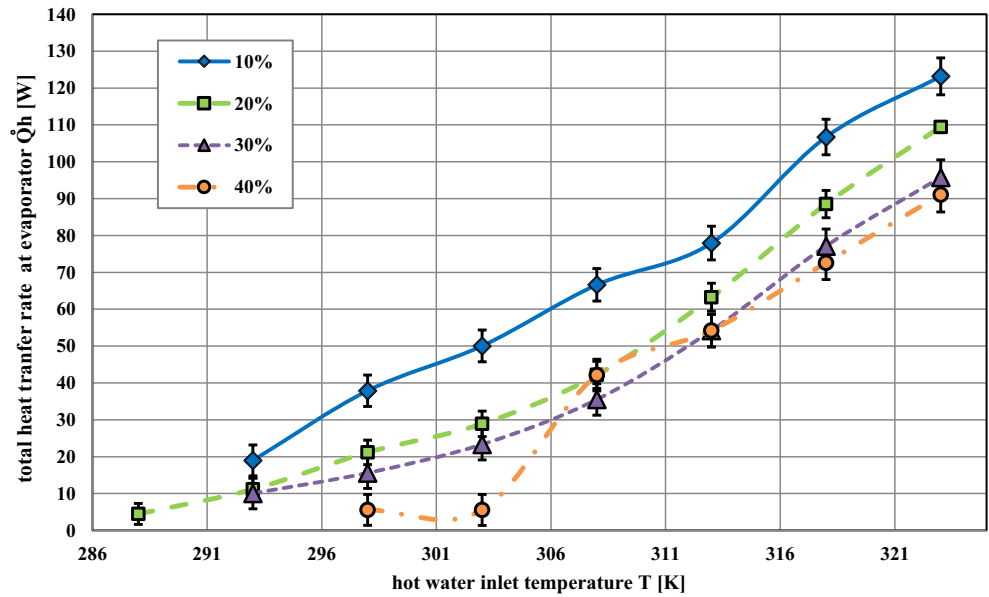
Throughputs are somewhat lower than at the evaporator because of heat losses to the surroundings. Presenting the condenser heat transfer rates is a way to check the correctness of empirical results (presence of anomalies). This is most often omitted in other studies and mean heat transfer rate at evaporator and condenser is stated.

The thermal efficiency of R134a is illustrated in Fig. 6. As expected the combined uncertainties of efficiency become very large for low hot water inlet temperatures. This is because small heat losses are of the same magnitude as throughputs. For the lowest values the uncertainty band can span the whole 100% range. For high inlet temperatures most probable efficiency is about 80%. It seems to be set at approximately constant level in the full experimental temperature range, yet high uncertainties at lower temperatures sub range make efficiency virtually unknown there. The remaining 20% cannot be interpreted only as a loss to surroundings, because the heat pipe is well insulated – maximum heat leaks are estimated as $4 \div 5\%$ (for the maximum water inlet temperatures). The discrepancy between measured efficiency and the ideal one, originates majorly from the measurement uncertainties as showed

Table 1 Combined uncertainty values of heat transfer rate at evaporator \dot{Q}_h and at condenser \dot{Q}_c for 20% of TPCT filling ratio with R134a refrigerant

\dot{Q}_h [W]	$u_c(\dot{Q}_h)$ [W]	$u_c(\dot{Q}_h)/\dot{Q}_h$	\dot{Q}_c [W]	$u_c(\dot{Q}_c)$ [W]	$u_c(\dot{Q}_c)/\dot{Q}_c$
4.5	6.3	142%	3.8	5.4	142%
11.2	6.4	57%	7.6	5.4	71%
21.2	6.5	30%	15.1	5.5	36%
28.9	6.6	23%	22.7	5.6	25%
42.2	6.9	16%	34.0	6.0	18%
63.3	7.6	12%	52.0	6.7	13%
88.5	8.5	10%	71.8	7.7	11%
109.4	9.4	9%	83.1	8.3	10%

Fig. 4 Heat transfer rates at evaporator section against inlet hot water temperature for different filling ratios with R134a refrigerant (plots present only temperature ranges where heat pipe was working for given FR)



in Fig. 6. Relative uncertainty of efficiency is the greatest of all of the measured quantities, because it is cumulated from uncertainties of the heat transfer rates to cold and hot water (see Table 1). Constant value of efficiency over temperature range comes from averaging of various measurement series.

In the Fig. 7 temperature distributions along the heat pipe for working fluids R404A and R134a are showed - the comparison with saturation temperatures inside TPCT is made for 10% FR. Working fluid vapor temperature in the evaporator section is close to the temperature of the wall (slightly lower), thus film in the evaporator section evaporates. Generally, for all of working fluids, for 10% FR evaporation is the most

intensive (due to the thinnest falling film), comparing to 20–40%. For refrigerant R404A the temperature of the vapor is higher than wall temperature at the upper part of the evaporator section which is shown precisely in Fig. 7. As a result vapor condenses on the falling film in the part of the evaporator section. For R134a (10% FR) saturation (vapor) temperature is lower than wall in the evaporator section. As can be seen in Fig. 7 the condenser section temperature is well below saturation temperatures, for both of the working fluids, so the condensation process occurs. The possible explanation for condensation at top part of the evaporator section is a subcooling of film falling from the condenser due to local

Fig. 5 Heat transfer rates at condenser section against inlet hot water temperature for different filling ratios with R134a refrigerant

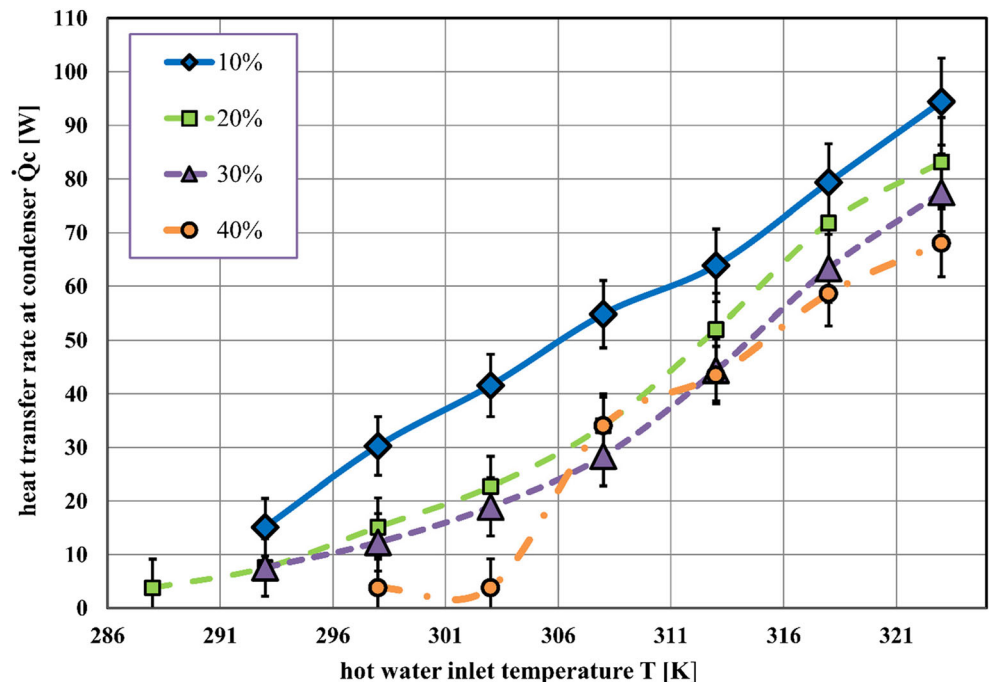
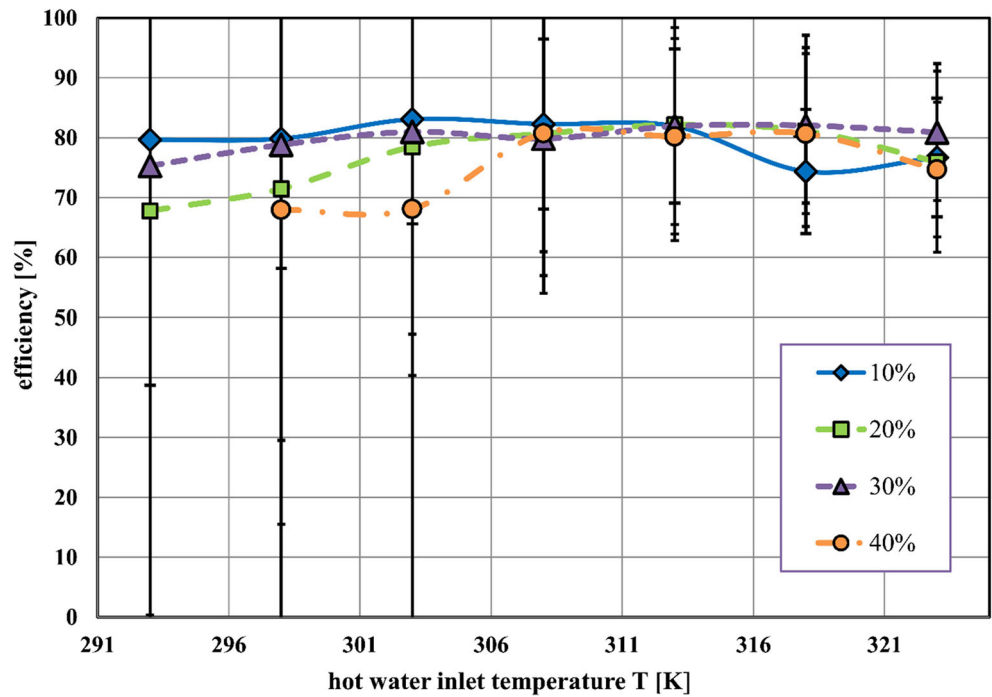


Fig. 6 TPCT thermal efficiency for various R134a filling ratios and hot water inlet temperatures



change of the concentration of the refrigerant blend. As one can observe in Fig. 7 temperature at the top of condenser for R134a is decreasing sharply, even the two first temperatures are nearly even. It can be a result of presence of small amount of non-condensable gases (NCGs). Even, the special care was taken to check air-tightness of whole system (TPCT plus filling equipment) small residuals of other gases could be present

and might be dragged by the vapor flow to the top part of the condenser section. Interestingly, small NCGs effects were observed only for R134a (contamination of the refrigerant is improbable). However, because only the small part of the condenser is blocked heat transfer is not affected considerably. Figure 8 shows the heat transfer rate at the evaporator section for refrigerant R404A. An increase in inlet hot water

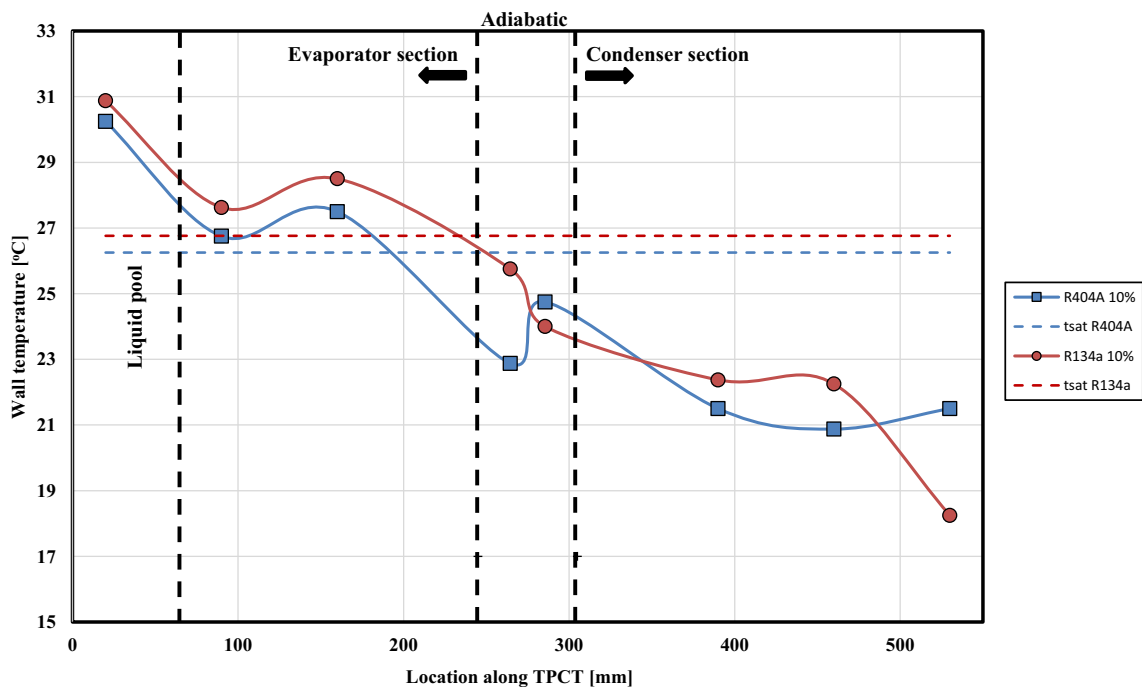
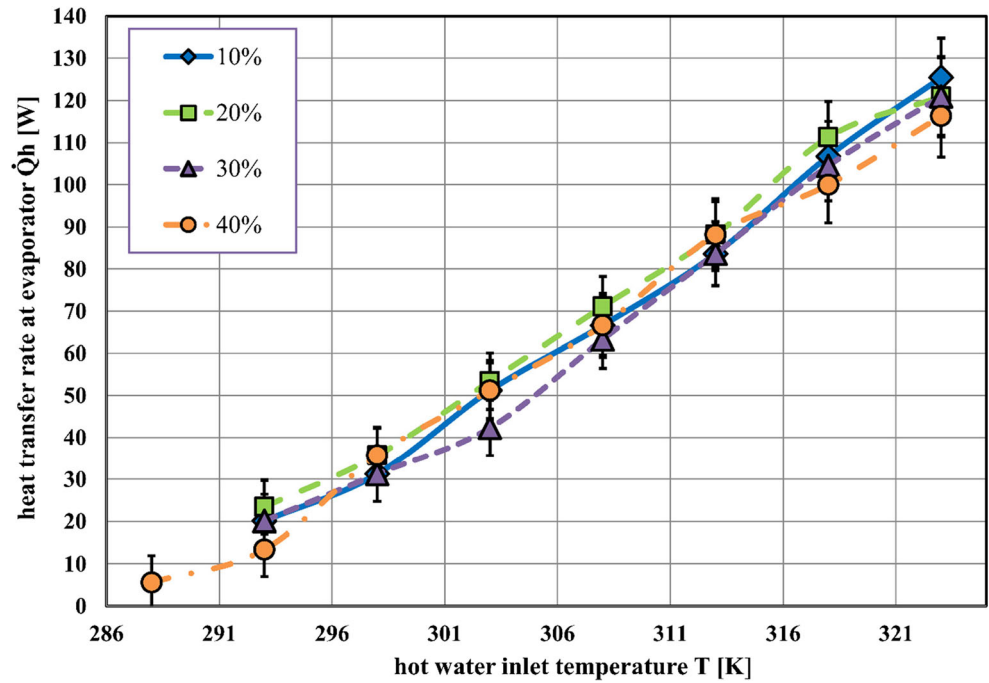


Fig. 7 Temperature distribution along the heat pipe for working fluids: R404A and R134a 10% FRs with marked saturation temperatures for each refrigerant inside TPCT

Fig. 8 Heat transfer rates at evaporator section against inlet hot water temperature for different filling ratios with R404A refrigerant



temperature causes approximately a linear increase in TPCT throughput, as it could be observed in the case of R134a. Although, there are relatively small differences for all considered filling ratios, comparing to R134a.

For all FRs the TPCT the performance is satisfactory, differences in the heat transfer rates are within uncertainty band. Heat transfer rates at the condenser have the same trends at the evaporator (Fig. 9) but lower values because of heat losses. Efficiencies stay at constant level of 80% (Fig. 10) similar to R134a. Although

at lowest temperatures uncertainties are relatively high which makes the obtained efficiency values unreliable.

For last tested refrigerant R407C, evaporator heat transfer rates are the lowest (Fig. 11). There is also a wide range of omitted values, for which the measured throughputs are very low (a couple of watts). This can be considered as a range where TPCT is virtually not functioning. At 10% FR throughputs are on average 10–15 W lower than the rest. Heat transfer rates for 10% FR are generally the lowest, where for other FRs they are nearly

Fig. 9 Heat transfer rates at condenser section against inlet hot water temperature for different filling ratios with R404A refrigerant

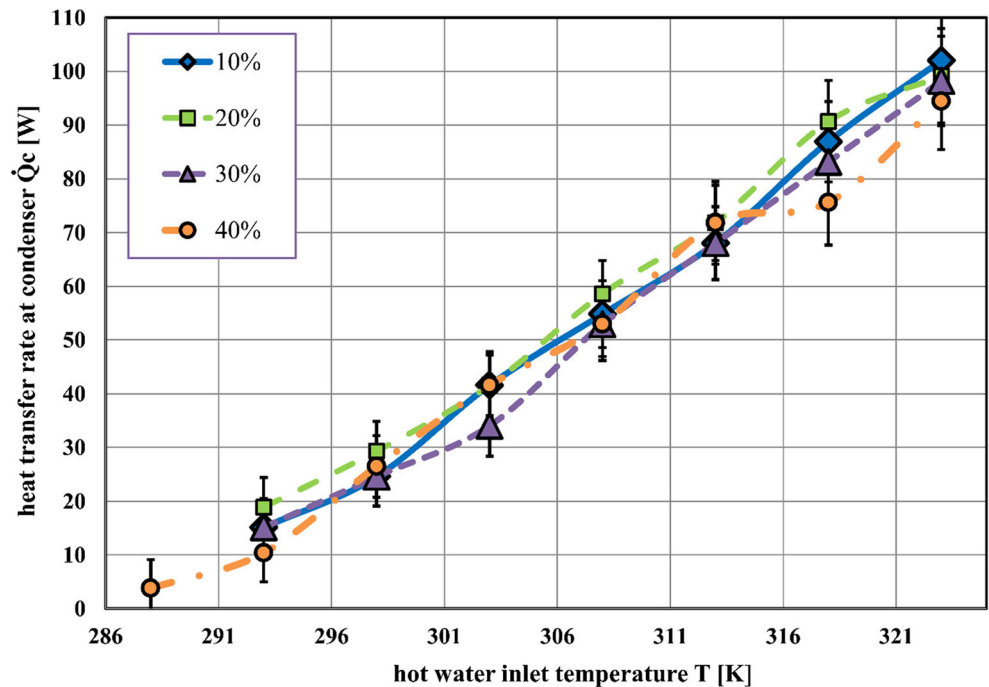
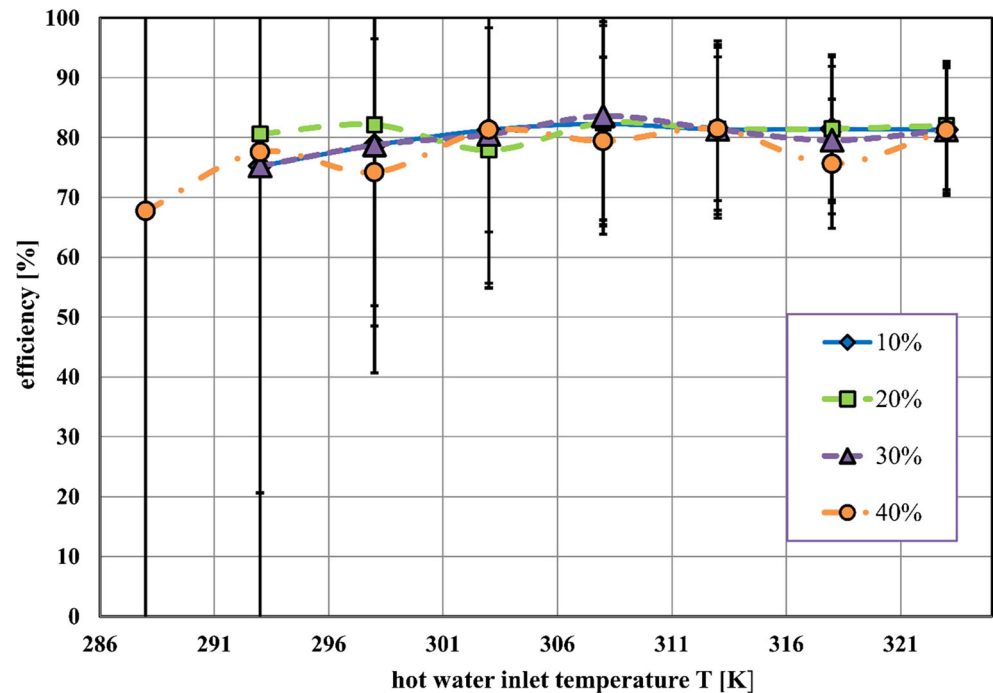


Fig. 10 TPCT thermal efficiency for various R404A filling ratios and hot water inlet temperatures



identical. Maximal heat transfer rate at the evaporator is 82 W for 30% FR. Generally R407C throughputs are 50–60% of R134a or R404A. The same situation holds for the condenser (Fig. 12) where throughputs are lower due to heat losses.

R407C is a strongly zeotropic refrigerant, which is a characteristic that has apparently adverse effect on heat transfer properties. The process of condensation can occur at the top of the evaporator section, because of the difference between dew and bubble point temperatures. As a result, part of the

evaporator is blocked and does not absorb heat from outside. Thus R407C it is not recommended for utilization in TPCTs. In Fig. 13 thermal efficiencies are shown which also stay at 80% as in the case of other tested refrigerants. R407C exhibits high uncertainties for widest range of inlet hot water temperatures for which efficiencies are virtually unknown.

In Fig. 14 typical temperature distributions along the TPCT wall are plotted for different heating water inlet temperatures. The temperatures are measured at the evaporator, adiabatic and

Fig. 11 Heat transfer rates at evaporator section against inlet hot water temperature for different filling ratios with R407C refrigerant

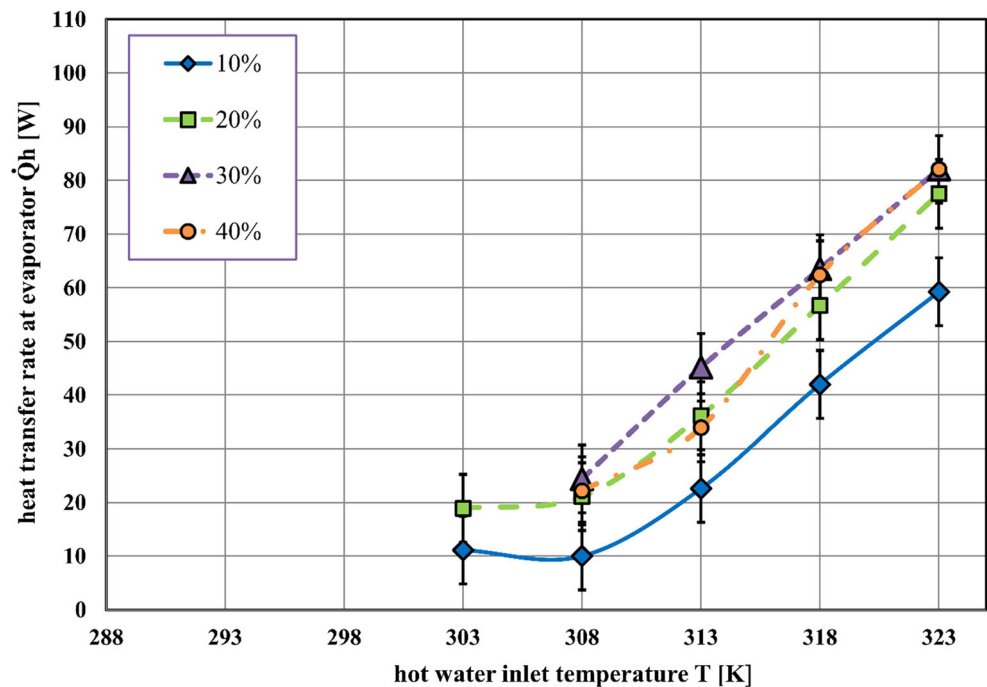
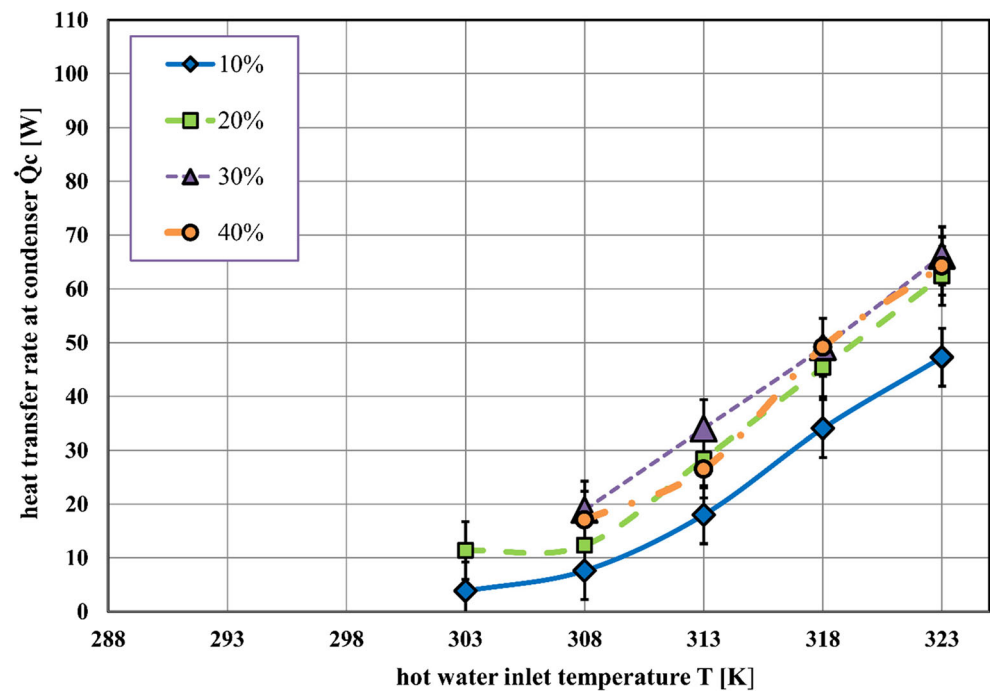


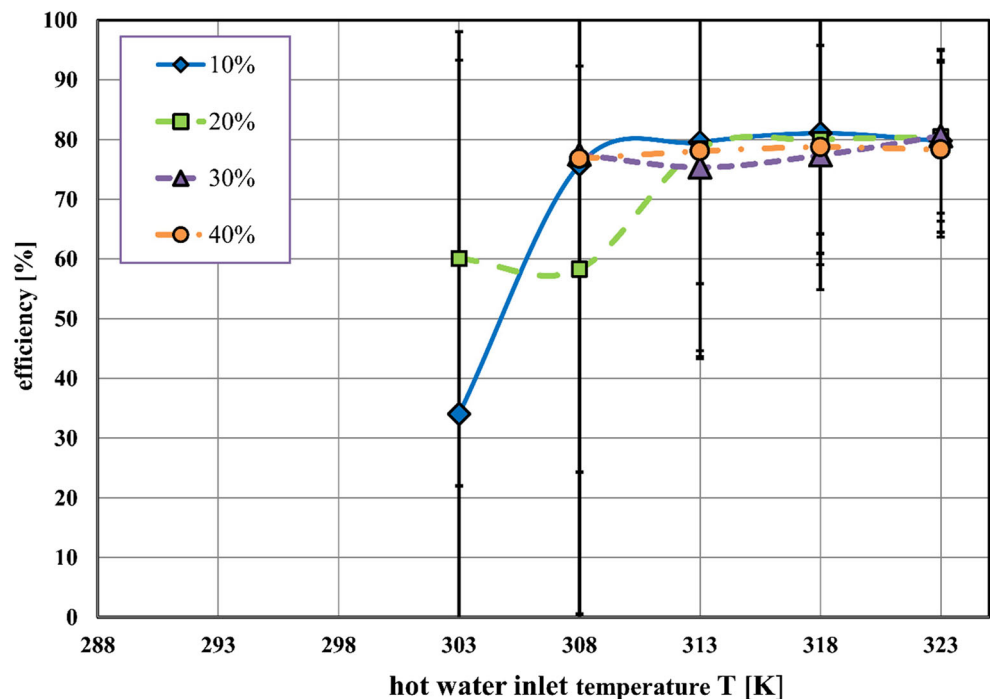
Fig. 12 Heat transfer rates at condenser section against inlet hot water temperature for different filling ratios with R407C refrigerant



condenser sections at the positions according to Fig. 2 (thermocouples 1–8). At the lower part of TPCT liquid pool is present and in the rest of the evaporator section vapor flows at constant temperature in countercurrent direction to liquid film. The condenser section position is also marked in the Fig. 14. Comparison of pressures for all working fluids is presented in Fig. 15. Average values for refrigerants R407C and R404A are similar. For R134a pressure values are lower (for 20–40% are comparable and for 10% are the lowest). The lower pressure for 10% FR can be

explained by bigger volume over the liquid level than for 20–40%. The heat transfer intensity can be also different at various FRs. As the result, temperature distributions are varying corresponding to variant average TPCT temperatures and saturation pressures. At small filling ratios evaporation of the falling liquid film is the dominating heat transfer mechanism, which is generally more efficient than pool boiling. As a result the heat transfer coefficient at evaporator section is high. On the other hand the condenser section is fully wetted by a thin condensate film, so one

Fig. 13 TPCT thermal efficiency for various R407C filling ratios and hot water inlet temperatures



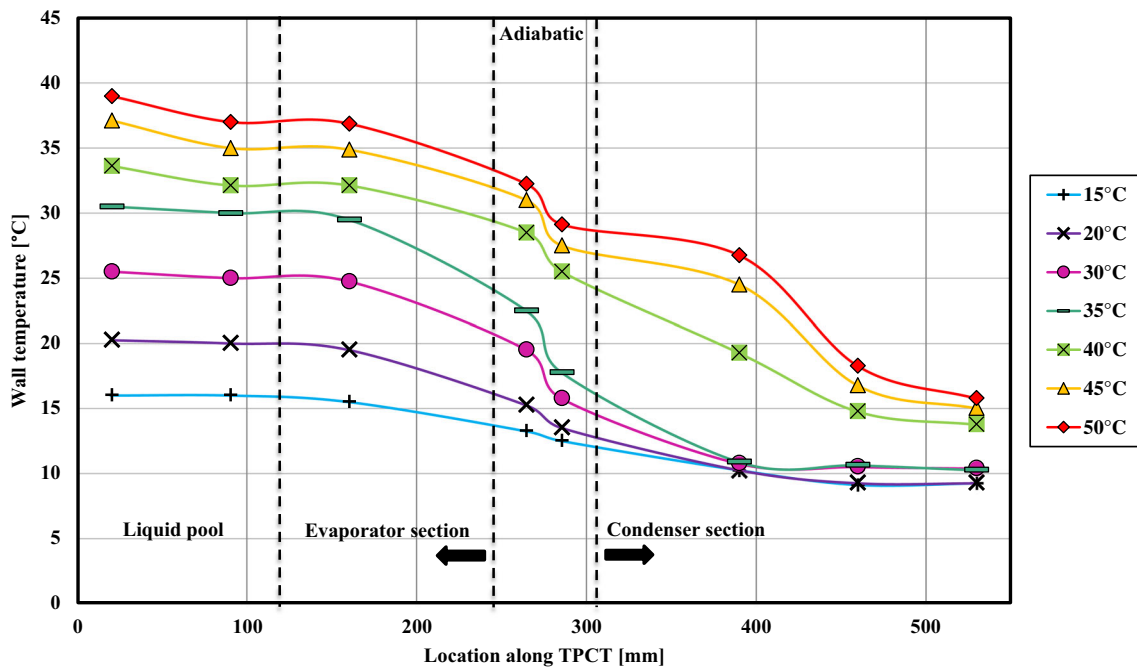


Fig. 14 Temperature distribution along the heat pipe, for 20% filling ratio – R134a working fluid

can expect that the heat transfer coefficient at the condenser section would be greater than at the evaporator (which is partially wetted with thin film and partially flooded with liquid). Thus the saturation temperature is shifted towards the condenser (lowered), and mutually the inside pressure is lower than in different examined cases. For 20–40% amount of working fluid is enough for circulation inside the tube. If tube is filled with 10%, working fluid probably evaporates completely (wall is not wetted by the falling film), or it is in immediate state before the dry-out.

Typical thermal resistances of the evaporator section for various filling ratios are given in Fig. 16 (for refrigerant R404A). As

one can see the thermal resistances are decreasing exponentially with increase of the hot water inlet temperature (mutual increase of heat transfer rate). The highest thermal resistances characterize 40% FR and they are consequently higher than for any other FR for nearly all of investigated temperatures of heating water. Resistance for 40% FR is at least 30% greater than resistance for 10% FR and for the lowest temperatures this difference can exceed 100%. These experimental results confirm that for lower FRs heat transfer in the evaporator section of TPCT is more efficient – falling film thermal resistance is lower than liquid pool. It can be also stated that the thermal resistances are the

Fig. 15 Comparison of pressures inside TPCT for various filling ratios and hot water inlet temperatures (R134a, R404A, R407C)

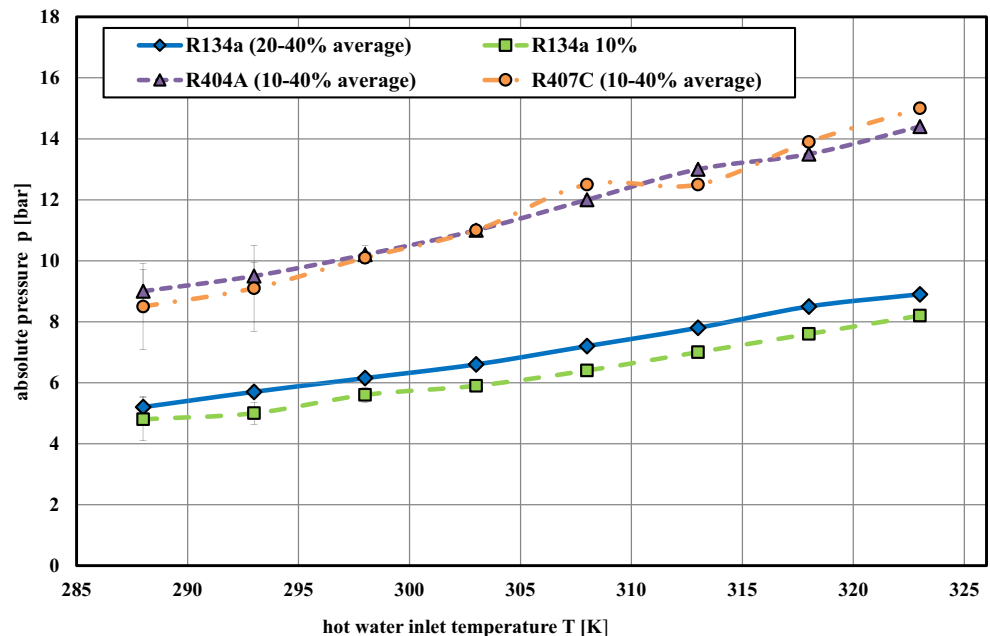
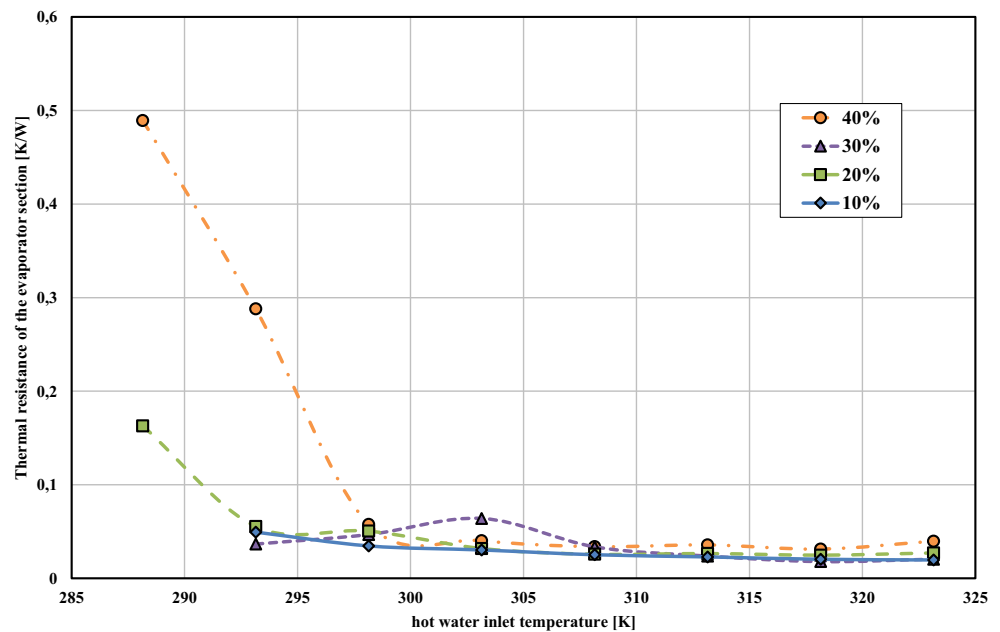


Fig. 16 Thermal resistance of the evaporator section for different filling ratios for the R404A refrigerant in the function of hot water inlet temperature



lowest, on the average, for 10% FR, even the measured values for 20% and 30% FRs are virtually equal or even lower than for 10% for the certain temperatures. It is not evident which FR: 20% or 30% performs better – discrepancy between this two is within uncertainty band (relative uncertainty of the thermal resistance is slightly greater than uncertainty of heat transfer rate).

The best FR and the highest TPCT throughputs for all three refrigerants are given in Table 2. Refrigerant R134a has lower price comparing to the R404A and R407C. The maximum operating pressure is also stated. The best working fluid is R404A, with a slight advantage over R134a. For both working fluids 10% FR is optimal. FR is an important parameter. Too low FR can lead to dry-out limit [23], but if it is too high it can degrade TPCT performance. For 10% FR of R134a, however, falling liquid film is the thinnest, thus the heat transfer characteristics of TPCT are the best. It appears it is a state immediately before occurrence of the dry-out. Higher FR increases wickless heat pipe mass and its manufacturing cost. These lead to conclusion that FR should be ideally set as low as possible. R134a has one advantage over R404A that is lower operation pressures. However, operating pressure difference between these two, is not crucial for TPCT container construction. Copper pipe with 1 mm thick wall, can withstand both

pressures. As a heat pipe container with a thinner wall would be too susceptible to external damages, so it is not preferable. Refrigerant R407C exhibits the worst performance, with about 50% lower heat transfer rates, high operation pressures and higher optimal FR than R134a and R404A. Degraded performance of R407C can be a consequence of its strong zeotropic properties. Even though R404A has a little advantage in throughputs than R134a, it lays practically within uncertainty band. In effect, R404A and R134a are characterized by practically identical performance, and can be utilized as TPCT working fluid interchangeably.

5 Conclusions

Three modern refrigerants were tested as TPCT working fluids: R134a, R404A and R407C. The reason for choosing refrigerants is their better performance in the low temperature range compared to water or alcohols, which was discussed in the introduction section. Four volumetric FR were considered: 10%, 20%, 30% and 40% of the total wickless heat pipe volume. A vertical two-phase thermosyphon was heated and cooled by water flowing through heat exchangers mounted

Table 2 Comparison of TPCT working fluids performance

Working Fluid	The best Filling Ratio	Highest heat transfer rate at evaporator [W]	Maximal absolute operation pressure [bar]	Price of 1 kg of refrigerant [\$/kg]
R134a	10%	123	8.9	8.72
R404A	10%	125	14.9	10.35
R407C	30%	82	16.0	10.20

on the evaporator and condenser sections. The hot water inlet temperature was adjusted from 288 K by steps of 5 K to 323 K. Cold water was kept at a constant level of 283 K. This range of temperatures is typical for waste energy recuperation (HVAC systems). Measured parameters were heat transfer rates at the evaporator and condenser, thermal efficiency and the absolute pressure inside the heat pipe. Experimental research showed that R134a and R404A exhibit the highest heat transfer rates for 10% FR. Differences between them are within uncertainty of measurement. R134a has a slight advantage because of its lower operating pressure than R404A (approximately 4 bars lower). Although, in the author's view, this does not have an impact on TPCT construction (wall thickness). This leads to the conclusion that these two refrigerants can be utilized interchangeably. R407C is characterized by about 50% lower throughputs than the other two. Disadvantages of this refrigerant are also the high working pressure (at the level of R404A) and a high optimal FR, which is 30%. This makes R407C the worst of the three. Probably its strong zeotropic behavior is the reason for its degraded performance. The heat pipe filled with R134a refrigerant performs best because it is a one-component fluid, which always evaporates at one (constant) temperature. The other two working fluids considered are zeotropic mixtures, which can degrade thermal effectiveness, e.g. imposing additional mass transfer resistance in the condensation process [24].

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