A Concept of Evaporator Aided by Capillary Effect

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ABSTRACT

Development of devices with reduced pumping power is gaining significant attraction recently. One of the examples are Organic Rankine Cycles which prove to be an attractive perspective for production of electricity and heat in the distributed generation. Actually the pumping power in such systems is very significant. In some cases it can reach up to 30% of generated power by the system. Hence reduction of pumping power is very tempting. Such possibilities are offered by application of the porous structures to aid the fluid motion.

In the paper is presented an experimental facility designed for studies of the capillary effect in tubes made of porous media together with the results of commissioning tests. Examined are two fluids, namely ethanol and water, at three different fillings and three different evaporator temperatures. The results in the form of pressure distribution with time have been presented. The potential application of such effect is for example in the evaporator of the domestic micro CHP unit, where the reduction of pumping power could be obtained. Preliminary analysis of the results indicates water as having the best potential for developing the capillary effect.

The net effect obtained in the facility was approximately 22hPa in pressure difference has been obtained for studied porous material, namely the sintered tube from stainless steel powder with the average pore-size of 3 micrometers.

INTRODUCTION

Loop heat pipes (LHPs) as reliable two-phase passive thermal control devices can manage large amounts of heat with a good control of the heat source temperature. LHPs operate passively by means of capillary forces generated in the evaporator and its design needs special attention as it is coupled with the compensation chamber, which is responsible for establishing the loop's operation temperature. Additionally, the compensation chamber self-regulates the working fluid inventory during the LHP operation. Several applications of LHPs have already shown their thermal control capability in space environments, Launay [1].

On the basis of the concept of LHP the idea of evaporator shell and tube heat exchanger has been put forward. The promising area of application of the general idea of LHP is the modern dispersed energy sector development, and in particular applications utilizing cogeneration and recovery of waste heat.

NOMENCLATURE

d	[m]	diameter
$\dot{m}$	[W]	mass flow rate
p	[N/m^2]	pressure
$\dot{Q}$	[W]	rate of heat
T	[K]	temperature

Special characters

Δ	error of determination

Subscripts

cond	condenser
p	pore
e	effective
evap	evaporator
p	preheat
0	ambient or reference

One of such examples that could in future supplement the centralized energy sector is the micro heat and power unit (micro CHP) operating according to the Clausius-Rankine cycle with organic fluid (ORC) [2,3]. In such system the heat produced by the boiler can be used for central heating and the utility hot water for domestic use, but as a byproduct electricity can also be generated, which can be used on site or sold to the grid. The source of heat for such micro power plant, in relation to the local capabilities, can be the fossil fuel or renewable sources of energy. Such heat in the micro power plants is better used than in professional power plants producing electricity only.

As the capillary evaporator and the compensation chamber form only one component in the LHP, their design must be carefully carried out to promote the desirable device operation. In the paper presented is the experimental facility designed for studies of the capillary effect in the evaporator fitted with the tube made of porous material together with the results of commissioning tests. Examined are two fluids, namely ethanol and water, at three different fillings and three different evaporator temperatures. The results in the form of pressure distribution with time have been presented as well as integral characteristics of the evaporator.

EXPERIMENTAL FACILITY

The most complicated parts of LHP are the evaporator and compensation chamber, which are most of the times configured to be present in one containment, as the compensation chamber usually is directly prior to the evaporator. The wick structure is responsible for the transport of working fluid in the loop. The wick is usually made of high quality sintered porous powders to induce the capillary pressure difference indispensable for self excited working fluid circulation in the loop. In our analysis we are going to consider the arrangement presented in Fig. 1. The principle of operation of the loop heat pipe is relatively simple. The heat is supplied to the preheater and the evaporator where the working is intended to reach saturation state (preheater) and subsequently evaporate (evaporator). The working fluid forms a meniscus at the contact surface between liquid and vapor in the wick structure. Arising capillary forces in the form of capillary pressure push out vapour in the direction of condenser rendering the fluid transport around the loop. It ought to be stressed that no circulation pump is needed in such arrangement. The compensation chamber serves for storing and sustaining of the surplus of working fluid and control of LHP operation. As can be noticed the only form of energy required to be supplied to the device is the thermal energy. As such the waste energy can be considered and that opens significant potential for other applications. The primary wick in the evaporator must produce the capillary pressure necessary to overcome the total pressure drop in the loop sustaining in such way the continuous operation of the LHP and hence the discussed evaporator.

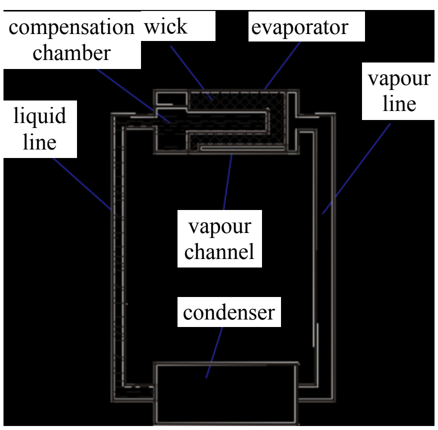

Figure 1 Principle of operation of the Loop Heat Pipe

To take the advantage of capillary forces for aiding of the working fluid pumping in the foreseen domestic micro CHP facility, it is assumed that in the perspective CHP arrangement evaporator will be a shell and tube recuperator, in which tubes will be made of sintered metal powder in the form of a porous tubular wick. Porous material will transport the working fluid from the inner to the outer surface of the wick, from where it will be evaporated and transported further to turbine. The presence of the wick will cause a reduction of pressure head demand for pumping of the working fluid. The evaporator collector that feeds the working fluid to evaporator tubes will additionally serve as a compensation chamber. The preheater present in Fig. 2 heats the working fluid up to saturation temperature. Structural scheme of the cycle is shown in Fig. 2.

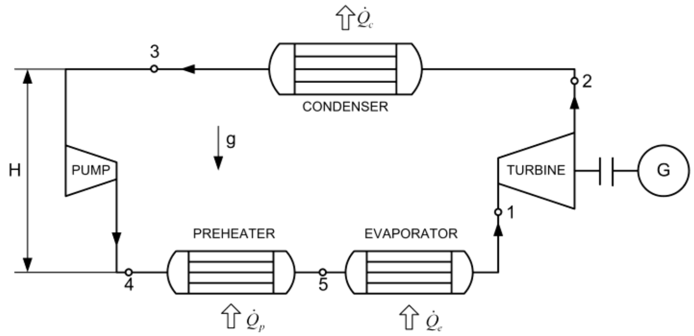

Figure 2 Structural scheme of the ORC cycle

Due to the fact that the fundamental element as well as the topic of present investigations is the tube with porous filling referred to later as heat exchanger which utilizes capillary pressure difference arising in the porous structure the activities started with selection of the appropriate structure of the tube with the porous filling. The required porous tube to be used in LHP evaporator is made of sintered stainless steel powder AISI 316L, which results in a fully permeable porous wick featuring the mean pore structure of 1, 3 and 7 micrometers, bearing the commercial name of SIPERM (sintered permporous). The cross-section of the material sample is made of stainless steel powder, characteristic for a very high adsorption coefficient due to irregular shape of powder grain. The implemented technology for obtaining such porous structure is named the technique of isostatic pressing. AISI 316L is a stainless steel with elevated corrosion resistance in aggressive environment. It enables operation in temperatures reaching 300/400°C.

Selection of stainless steel as the material of the test section was dictated primarily by the two most important assumptions made for the study. The first objective of the study was to be able to carry out research with different working fluids such as water, ethanol, hydro fluorocarbons without the problems of material damage/corrosion. Actually stainless steel renders it possible to work with most of working fluids, [6]. The second fundamental criterion was the possibility of mechanical and thermal processing of the wick structure. Bearing in mind these constraints the porous tube connected by welding with the stainless steel flange was developed. The porous tube is closed on one end with the plate made of SIPERM to separate the high and low pressure zones of the arrangement, whereas the other end was welded to the flange to be able to position the tube within the internally grooved tube, named evaporator.

Another challenge in the development of the test section of the experimental facility, requiring implementation of advanced processing technique, was the body of the evaporator. The technology of production of the wick on the basis of permeable porous tube disabled cutting of the grooves on its external surface. These grooves are necessary to transport away the produced vapor. Such grooves cut in the porous material are characteristic to the studies by Chernysheva and Maydanik [7], Bai et al. [8] or Hartenstine et al. [9]. In order to alleviate that problem the groves were cut inside the tube serving as the casing for evaporator. The tube was made of copper M1E by means of the wire electro-erosion machine with the accuracy of 0.01mm, Fig. 3.

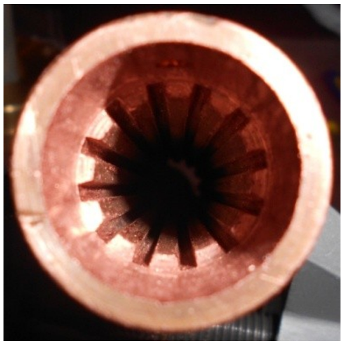

Figure 3 Grooves on internal side of the copper tube

The volume of the compensation chamber (CC) was experimentally adjusted. The CC was connected with the evaporator by means of special Teflon labyrinth sealings assuring the adequate tightness and possibility of non-damaging inspections of the inside chamber of the evaporator, wick and compensation chamber. The view of the connection between compensation chamber and the evaporator is shown in Fig. 4.

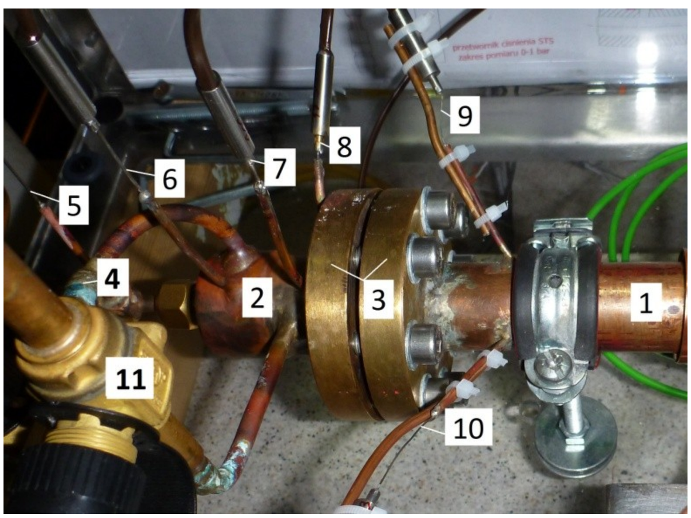

Figure 4 View of the connection between the compensation chamber and the evaporator; 1- evaporator casing, 2-compensation chamber, 3- flange connection between compensation chamber and evaporator, 4- filling/emptying socket, 5- thermocouple T12, 6-10 – thermocouples T1-T5, 11 – cut-off valve for pressure transducer P1

Transport lines connecting the evaporator to condenser were made of capillary tubes with internal diameters of 1.95mm in case of the vapour line and 1.8mm in case of the liquid line. Cooling of the condenser section was accomplished by means of water circulating in a closed-loop by the circulation pump providing the flow rate up to about 0.175 dm3/min. The test facility was also equipped with the visualisation section. To accomplish that applied were the two-way valves enabling to direct the flow of working fluid from the copper capillary tube to the transparent glass tube made of borium-silicone glass. The source of heat necessary to induce the process of heat transport in the wick in the form of capillary pressure difference was obtained by application of a heating wire of 400W duty allowance installed in the middle part of the evaporator casing. Uniform radial supply of heat to the wick was assumed. The supply of electric current was controlled by the PID controller in relation to the temperature on the contact surface between the heating element and the evaporator casing.

The measurement system of the facility consists of the following instrumentation:

- pressure; accomplished by the set of 4 pressure transducers with the measurement range 0-1 bar of absolute pressure and measurement class of 0.1. The transducers are integrated to digital displays AR500.
- temperature; set of 18 T-type thermocouples manufactured in the class 1 accuracy connected to temperature meters EMT200 with measurement error equal to +/- (0.002 x [T] + 0.3ºC + number) integrated with multiplexer PMP201.
- electric power; multimeter UT71E with measurement accuracy + /- (2%+50) recording the electrical power demand through the resistance heated wire generating the rate of heat supplied to the evaporator section.

The view of experimental facility without necessary insulation of evaporator, vapour and liquid lines is presented in Fig. 5.

Figure 5 View of the experimental facility without necessary insulation of evaporator, vapour and liquid lines.

In order to reduce the heat dissipation from the test facility the heating section, buffer tank with tappings and thermocouples were insulated using mineral wool with aluminium coating. The vapour line and liquid lines were insulated using the synthetic rubber of 13mm thickness.
As it was mentioned earlier the objective of investigations was to determine the capabilities of the facility to transport heat and mass through implementation in the pumpless arrangement of the porous structure with known parameters. The testing

procedure considered tests at three different levels of filling the installation with working fluid. Due to the lack of necessary information about the filling with working fluid the knowledge from refrigeration technology about the refrigeration installation filling was used. Another considered aspect of investigations which was attempted to be identified was the influence of the working fluid selection, installation filling and different levels of heat supply on the transport capabilities of the heat exchanger at the assumption that the liquid and vapour lines were of the same length and the distance between the condenser and evaporator was also the same. The latter assumption leads to the same level of hydrostatic pressure in the installation. Due to the above restrictions the following range of experimental tests was set for scrutiny:

- 2 different working fluids, namely water and ethanol,
- 3 different porous structures; pores size of 1, 3 and 7μm,
- 3 different fillings of the installation, i.e. 60, 65 and 70 ml,
- 3 different values of evaporator casing temperature settings, namely 90°C, 100°C and 110°C.

Table 1 presents the whole range of investigated parameters together with attainable pressure differences. Accomplished experiments indicate the crucial issue of the adequate filling with the working fluid. For both fluids considered it was found that the pressure difference can reach up to 22hPa for water as working fluid and the pore size of 1μm at the installation filling of 65ml. For other values of fillings significantly lower values of pressure difference have been obtained.

Table 1 Range of investigated parameters

No.	Filling volume	Heater setting	Working fluid	Pore size	P_2-P_1
	[ml]	[°C]		[μm]	[hPa]
1.	60	90, 100, 110	H_2O	R1	5
		90, 100, 110	Ethanol	R1	13
2.	65	90, 100, 110	H_2O	R1	21
		90, 100, 110	Ethanol	R1	15
3.	70	90, 100, 110	H_2O	R1	13
		90, 100, 110	Ethanol	R1	12
4.	60	90, 100, 110	H_2O	R3	14
		90, 100, 110	Ethanol	R3	16
5.	65	90, 100, 110	H_2O	R3	10
		90, 100, 110	Ethanol	R3	22
6.	70	90, 100, 110	H_2O	R3	6
		90, 100, 110	Ethanol	R3	14
7.	60	90, 100, 110	Ethanol	R7	5
8.	65	90, 100, 110	Ethanol	R7	10
9.	70	90, 100, 110	Ethanol	R7	8

RESULTS

During experiments four pressure readings were recorded, namely the ones in the compensation chamber, P1, vapour space beyond the wick, P2, before the condenser, P3, and after the condenser, P4. Here the discussion of the results will be done on the basis of the influence of different pore size and evaporator casing temperature on the attainable pressure difference. Presented will be the results for one value of the installation filling, namely 65ml and one applied wall temperature to the evaporator casing equal to 100°C. For the case of water as working fluid the results are presented in Figs. 6 and 7, whereas in the case of ethanol in Figs. 8 to 10.

Water as working fluid

Analysis of pressure distributions for particular measurements in case of water as working fluid indicates that the pressure drops in the liquid and vapour lines are small in comparison to the pressure difference before and after the evaporator (P2 (pressure after the evaporator and compensation chamber) is the highest pressure in installation and P1 (pressure before compensation chamber) is the lowest pressure in installation). In all distributions of pressure significant pressure fluctuations are observed. During investigations, in which the measurement series lasted for about 2.5 hours we can observe that the first half an hour is the time of reaching the steady state conditions. In case of experimental run presented in Fig. 6 that means that the rate of pressure increase is about 445 Pa/s. In authors opinion the subsequent pressure fluctuations can be explained by the fact that production of vapour in the grooves can contribute to choking of the porous structure. It ought to be stressed that in all runs the steady state conditions are reached after about 30 minutes of the experiment duration. In the case of research with water as working fluid the initial installation pressure was at the level of 40hPa. The supply of heat rendered the pressure increase. In case of the parameters corresponding to the run presented in Fig. 6 the pressure in the compensation chamber P1 stabilised at the level of 154hPa, whereas the pressure in the vapour line at 136hPa.

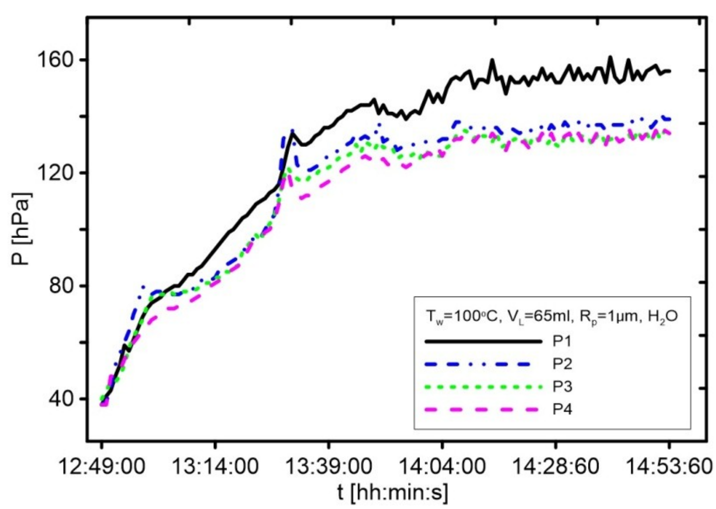

Figure 6 Pressure distribution of water in particular nodes of the cycle, T_w=100°C, filling 65ml, pore diameter d_p=1μm

From that pressure difference results the characteristics of the evaporator wick. In the case of the considered experiment that is equal to 22hPa for the temperature of evaporator casing of **10**0°C and the pore size of 1μm. In the case of the experiment presented in Fig. **7** where in relation to the data presented in Fig. **6** the size of the pore is increased from 1 to 3μm the pressure P1 stabilises at the level of 140hPa, whereas the pressure in the vapour channel is 130hPa. That confirms the fact that the reduction of the pore size leads to the increase of produced pressure. In case of the evaporator casing temperature of 100^{o}C that is 5hPa.

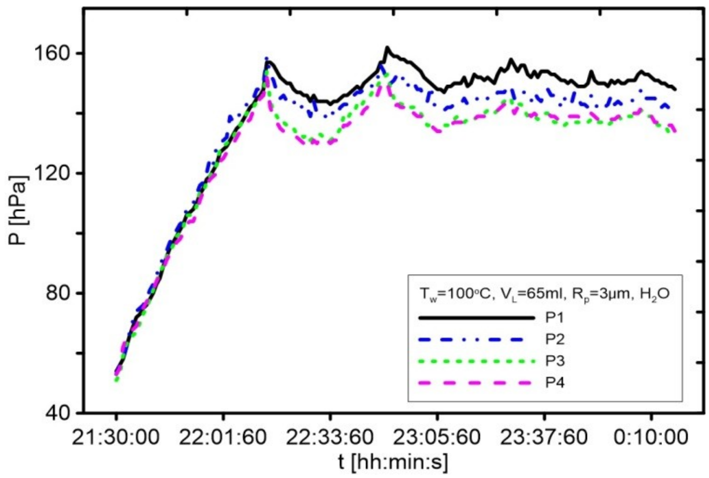

Figure 7 Pressure distribution of water in particular nodes of the cycle, T_w=100°C, filling 65ml, pore diameter d_p=3μm

Ethanol as working fluid

The second analysed fluid was chemically pure ethanol. That is one of the fluids which can be considered in the perspective installation of the domestic micro CHP, Mikielewicz and Mikielewicz (2010).

The analysis of pressure distributions for particular experimental runs is presented in Figs. 8 to 10. The general observation again is that the pressure drops in the liquid and vapour lines are small in comparison to the pressure difference between the points before and after the evaporator. Another important observation is that the pressure fluctuations are generally smaller than in case of water as working fluid despite the fact that the operational pressure of the loop is much higher than in case of water. The experimental run lasted also for about 2.5 hours and the first half an hour was related to the start-up, similarly as in the case of water as working fluid. The initial pressure in the installation is significantly different from the one present for the case of water. In case of filling of the installation with the volume of 65ml of ethanol the initial pressure stabilized at the level of about 135hPa, and due to the supply of heat the maximum pressure reached 302hPa, for the case of data from Fig. 8. That corresponds to the rate of pressure change in the system equal to about 93 Pa/s. In case of other pore sizes as well as other evaporator casing temperatures the pressure reached values of 400hPa. In case of the parameters corresponding to the Fig. 8 the pressure P1, pressure in the compensation chamber, settled at the level of 320hPa, whereas the pressure in the vapour line at the level of 299hPa. Therefore the wick produces the pressure increase of the order of 22hPa for the temperature of evaporator casing equal 100^{o}C and the pore size of 1μm. In the second case where the only difference to the setting from Fig. 8 is the pore size equal to

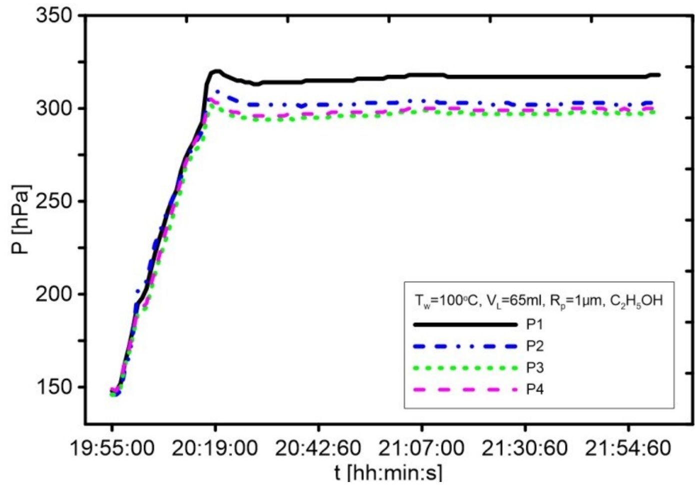

Figure 8 Pressure distribution of ethanol in particular nodes of the cycle, T_w=100°C, filling 65ml, pore diameter d_p=1μm

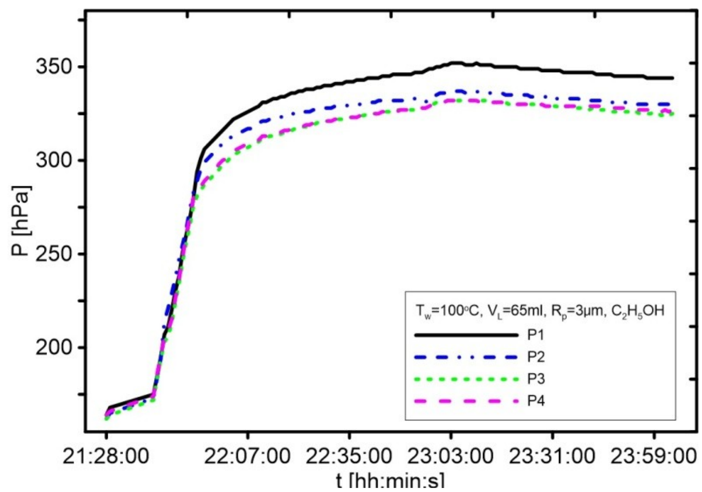

Figure 9 Pressure distribution of ethanol in particular nodes of the cycle, T_w=100°C, filling 65ml, pore diameter d_p=3μm

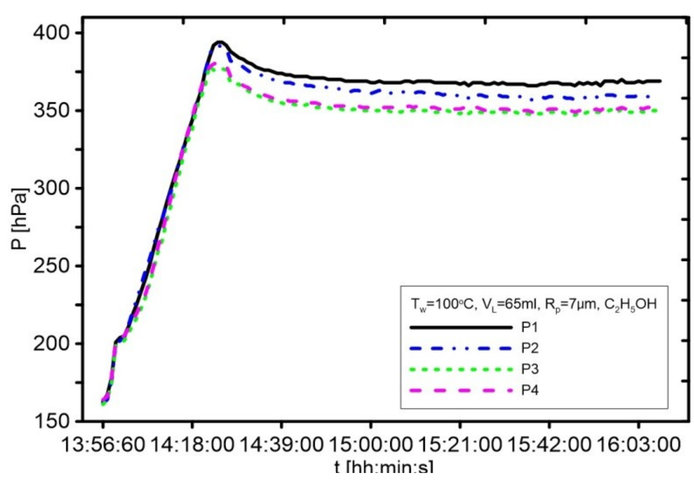

Figure 10 Pressure distribution of ethanol in particular nodes of the cycle, T_w=100°C, filling 65ml, pore diameter d_p=7μm

3μm the pressure settled at the pressure of 351hPa, whereas the pressure in the vapour channel at the level of 339hPa, respectively. That conforms the fact that the reduction of the pore diameter leads to the increase of pressure. In the case of the evaporator casing temperature of 100^{o}C that is 21hPa. In Fig. 10 presented have been the results for the case when the pore size is 7μm. In that case the pressure in the facility stabilized at the level of 370hPa, whereas the pressure in the vapour channel at the level of 350hPa. We can see now that the size of pores has practically no influence on the attained capillary pressure difference.

In table 2 presented are the results of mass flow rate attainable in the facility. Compared are the values for mass flow rate determined from the balance of evaporator and the balance of condenser. In the last column the difference between the two cases is presented. That difference does not exceed the value of 11%. That discrepancy should be regarded as small, especially in the light of the fact that the mass flow rates were determined by means of the energy balance method.

Table 2 Comparison of the values of mass flow rate determined by means of the energy balance in evaporator and condenser

T_w	Fluid	Pore size	Q_{evap}	$\dot{m}_{evap}$ $\times 10^{-5}$	$Q_{cond,w}$	$\dot{m}_{cond}$ $\times 10^{-5}$	Δ
[^{o}C]	[-]	[μm]	[W]	[kg/s]	[W]	[kg/s]	[%]
110	H_2O	R1	-	-	-	-	-
100	H_2O	R1	48.5	2.0	45.5	1.82	10.5
90	H_2O	R1	51.0	2.13	49.7	1.99	6.2
110	H_2O	R3	57.8	2.41	51.3	2.06	14.4
100	H_2O	R3	55.3	2.33	50.7	2.04	12.4
90	H_2O	R3	53.4	2.23	52.5	2.14	4.0
110	C_2H_5OH	R1	59.5	6.49	56.8	5.86	9.8
100	C_2H_5OH	R1	66.5	7.43	61.9	6.42	13.7
90	C_2H_5OH	R1	53.8	5.96	52.5	5.43	9.0
110	C_2H_5OH	R3	62.4	6.86	55.5	5.75	16.2
100	C_2H_5OH	R3	67.0	7.46	65.1	6.80	8.8
90	C_2H_5OH	R3	54.5	6.08	53.0	5.48	9.8
110	C_2H_5OH	R7	71.3	7.79	70.1	7.25	6.9
100	C_2H_5OH	R7	70.0	7.69	60.1	6.23	19.0
90	C_2H_5OH	R7	64.2	7.12	62.6	6.48	9.0

Presented in the table 2 comparison indicates that there is heat dissipation in the vapour line before the condenser. Despite application of the thermal insulation there were still some heat sinks present in the installation. The sections of the tubes, namely the vapour line, liquid line, evaporator and the condenser have been evaluated for the corresponding losses of heat to the surroundings indicating the difference of about 3W, which is the missing value from the comparison of the condenser and the evaporator.

CONCLUSION

The new facility for studies of capillary effect in the porous tube has been presented. The facility has been developed to study the possibility of designing a innovative evaporator capable of reducing the pumping power requirement for application for example in the ORC installation. The net effect obtained in the facility was approximately 22hPa in pressure difference has been obtained for studied porous material, namely the sintered tube from stainless steel powder with the average pore-size of 3 micrometers. That pressure difference can contribute to drive the fluid in the perspective heat exchanger. The topic of passive pumping will be further scrutinised with the view of testing other fluids, other pore sizes in the same material and finally by varying the vertical distance between the evaporator and condenser.

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